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لجنة الميكانيك

Polytechnic

Lecture 1

INTRODUCTION TO HYDRAULICS AND PNEUMATICS

Learning Objectives

Upon completion of this chapter, the student should be able to:

- ☐ Explain the meaning of fluid power.
- ☐ List the various applications of fluid power.
- ☐ Differentiate between fluid power and transport systems.
- ☐ List the advantages and disadvantages of fluid power.
- ☐ Explain the industrial applications of fluid power.
- ☐ List the basic components of the fluid power.
- ☐ List the basic components of the pneumatic systems.
- ☐ Differentiate between electrical, pneumatic and fluid power systems.
- ☐ Appreciate the future of fluid power in India.

1.1 Introduction

In the industry we use three methods for transmitting power from one point to another. Mechanical transmission is through shafts, gears, chains, belts, etc. Electrical transmission is through wires, transformers, etc. Fluid power is through liquids or gas in a confined space. In this chapter, we shall discuss a structure of hydraulic systems and pneumatic systems. We will also discuss the advantages and disadvantages and compare hydraulic, pneumatic, electrical and mechanical systems.

1.2 Fluid Power and Its Scope

Fluid power is the technology that deals with the generation, control and transmission of forces and movement of mechanical element or system with the use of pressurized fluids in a confined system. Both liquids and gases are considered fluids. Fluid power system includes a hydraulic system (*hydra* meaning water in Greek) and a pneumatic system (*pneuma* meaning air in Greek). Oil hydraulic employs pressurized liquid petroleum oils and synthetic oils, and pneumatic employs compressed air that is released to the atmosphere after performing the work.

Perhaps it would be in order that we clarify our thinking on one point. By the term “fluid” we refer to air or oil, for it has been shown that water has certain drawbacks in the transmission of hydraulic power in machine operation and control. Commercially, pure water contains various chemicals (some deliberately included) and also foreign matter, and unless special precautions are taken when it is used, it is nearly impossible to maintain valves and working surfaces in satisfactory condition. In the cases where the hydraulic system is closed (i.e., the one with a self-contained unit that serves one machine or one small group of machines), oil is commonly used, thus providing, in addition to power transmission, benefits of lubrication not afforded by water as well as increased life and efficiency of packings and valves. It should be mentioned that in some special cases, soluble oil diluted with water is used for safety reasons. The application of fluid power is limited only by the ingenuity of the designer, production engineer or plant engineer. If the application pertains to lifting, pushing, pulling, clamping, tilting, forcing, pressing or any other straight line (and many rotary) motions, it is possible that fluid power will meet the requirement.

Fluid power applications can be classified into two major segments:

Stationary hydraulics: Stationary hydraulic systems remain firmly fixed in one position. The characteristic feature of stationary hydraulics is that valves are mainly solenoid operated**.** The applications of stationary hydraulics are as follows:

Production and assembly of vehicles of all types.

- Machine tools and transfer lines.
- Lifting and conveying devices.
- Metal-forming presses.
- Plastic machinery such as injection-molding machines.
- Rolling machines.
- Lifts.
- Food processing machinery.
- Automatic handling equipment and robots.

Mobile hydraulics: Mobile hydraulic systems move on wheels or tracks such as a tower crane or excavator truck to operate in many different locations or while moving. A characteristic feature of mobile hydraulics is that the valves are frequently manually operated. The applications of mobile hydraulics are as follows:

□□ Automobiles, tractors, aeroplanes, missile, boats, etc.
□□ Construction machinery.
□□ Tippers, excavators and elevating platforms.
□□□ Lifting and conveying devices.
□□□ Agricultural machinery.

Hydraulics and pneumatics have almost unlimited application in the production of goods and services in nearly all sectors of the country. Several industries are dependent on the capabilities that fluid power affords. Table 1.1 summarizes few applications of fluid power.

Table 1.1 More applications of fluid power

Agriculture	Tractors; farm equipment such as mowers, ploughs, chemical and water sprayers, fertilizer spreaders, harvesters
Automation	Automated transfer lines, robotics
Automobiles	Power steering, power brakes, suspension systems, hydrostatic transmission
Aviation	Fluid power equipment such as landing wheels in aircraft. Helicopters, aircraft trolleys, aircraft test beds, luggage loading and unloading systems, ailerons, aircraft servicing, flight simulators
Construction industry/equipment	For metering and mixing of concrete rudders, excavators, lifts, bucket loaders, crawlers, post-hole diggers, road graders, road cleaners, road maintenance vehicles, tippers
Defense	Missile-launching systems, navigation controls
Entertainment	Amusement park entertainment rides such as roller coasters
Fabrication industry	Hand tools such as pneumatic drills, grinders, borers, riveting machines, nut runners
Food and beverage	All types of food processing equipment, wrapping, bottling,
Foundry	Full and semi-automatic molding machines, tilting of furnaces, die-casting machines
Glass industry	Vacuum suction cups for handling

Hazardous gaseous areas	Hydraulic fracturing technologies: It involves pumping large volumes of water and sand into a well at high pressure to fracture shale and other tight formations, allowing hazardous oil and gas to flow into the well. However, hydraulic fracturing has serious environmental and water pollution related issues.
Instrumentation	Used to create/operate complex instruments in space rockets, gas turbines, nuclear power plants, industrial labs
Jigs and fixtures	Work holding devices, clamps, stoppers, indexers
Machine tools	Automated machine tools, numerically controlled(NC) machine tools
Materials handling	Jacks, hoists, cranes, forklifts, conveyor systems
Medical	Medical equipment such as breathing assistors, heart assist devices, cardiac compression machines, dental drives and human patient simulator
Movies	Special-effect equipment use fluid power; movies such as Jurassic park, Jaws, Anaconda, Titanic
Mining	Rock drills, excavating equipment, ore conveyors, loaders
Newspapers and periodicals	Edge trimming, stapling, pressing, bundle wrapping
Oil industry	Off-shore oil rigs
Paper and packaging	Process control systems, special-purpose machines for rolling and packing
Pharmaceuticals	Process control systems such as bottle filling, tablet placement, packaging
Plastic industry	Automatic injection molding machines, raw material feeding, jaw closing, movement of slides of blow molder

Press tools	Heavy duty presses for bulk metal formation such as sheet metal, forging, bending, punching, etc.
Printing industry	For paper feeding, packaging
Robots	Fluid power operated robots, pneumatic systems
Ships	Stabilizing systems, unloading and loading unit, gyroscopic instruments, movement of flat forms, lifters, subsea inspection equipment
Textiles	Web tensioning devices, trolleys, process controllers
Transportation	Hydraulic elevators, winches, overhead trams
Under sea	Submarines, under sea research vehicles, marine drives and control of ships
Wood working	Tree shearers, handling huge logs, feeding clamping and saw operations

The following are the two types of hydraulic systems:

1. **Fluid transport systems:** Their sole objective is the delivery of a fluid from one location to another to accomplish some useful purpose. Examples include pumping stations for pumping water to homes, cross-country gas lines, etc.
2. **Fluid power systems:** These are designed to perform work. In fluid power systems, work is obtained by pressurized fluid acting directly on a fluid cylinder or a fluid motor. A cylinder produces a force resulting in linear motion, whereas a fluid motor produces a torque resulting in rotary motion.

1.3 Classification of Fluid Power Systems

The fluid power system can be categorized as follows:

1. **Based on the control system**

☐ **Open-loop system:** There is no feedback in the open system and performance is based on the characteristics of the individual components of the system. The open-loop system is not accurate and error can be reduced by proper calibration and control.

- **Closed-loop system:** This system uses feedback. The output of the system is fed back to a comparator by a measuring element. The comparator compares the actual output to the desired output and gives an error signal to the control element. The error is used to change the actual output and bring it closer to the desired value. A simple closed-loop system uses servo valves and an advanced system uses digital electronics.

2. Based on the type of control

- **Fluid logic control:** This type of system is controlled by hydraulic oil or air. The system employs fluid logic devices such as AND, NAND, OR, NOR, etc. Two types of fluid logic systems are available:
  - **(a)** *Moving part logic (MPL):* These devices are miniature fluid elements using moving parts such as diaphragms, disks and poppets to implement various logic gates.
  - **(b)** *Fluidics:* Fluid devices contain no moving parts and depend solely on interacting fluid jets to implement various logic gates.
- **Electrical control:** This type of system is controlled by electrical devices. Four basic electrical devices are used for controlling the fluid power systems: switches, relays, timers and solenoids. These devices help to control the starting, stopping, sequencing, speed, positioning, timing and reversing of actuating cylinders and fluid motors. Electrical control and fluid power work well together where remote control is essential.
- **Electronic control:** This type of system is controlled by microelectronic devices. The electronic brain is used to control the fluid power muscles for doing work. This system uses the most advanced type of electronic hardware including programmable logic control (PLC) or microprocessor (μP). In the electrical control, a change in system operation results in a cumbersome process of redoing hardware connections. The difficulty is overcome by programmable electronic control. The program can be modified or a new program can be fed to meet the change of operations. A number of such programs can be stored in these devices, which makes the systems more flexible.

1.4 Hydrostatic and Hydrodynamic Systems

A hydrostatic system uses fluid pressure to transmit power. Hydrostatics deals with the mechanics of still fluids and uses the theory of equilibrium conditions in fluid. The system creates high pressure, and through a transmission line and a control element, this pressure drives an actuator (linear or rotational). The pump used in hydrostatic systems is a positive displacement pump. The relative spatial position of this pump is arbitrary but should not be very large due to losses (must be less than 50 m). An example of pure hydrostatics is the transfer of force in hydraulics.

Hydrodynamic systems use fluid motion to transmit power. Power is transmitted by the kinetic energy of the fluid. Hydrodynamics deals with the mechanics of moving fluid and uses flow theory. The pump used in hydrodynamic systems is a non-positive displacement pump. The relative spatial position of the prime mover (e.g., turbine) is fixed. An example of pure hydrodynamics is the conversion of flow energy in turbines in hydroelectric power plants.

In oil hydraulics, we deal mostly with the fluid working in a confined system, that is, a hydrostatic system.

1.5 History of Fluid Power

Fluid power is as old as our civilization itself. Water was used for centuries to produce power by means of water wheels and air was used to turn windmills and to propel ships. Chinese used wooden valves to control water flow through bamboo pipes in 4000 BC. Ancient Egyptians have built a masonry dam across Nile, 14 miles south to present Cairo, for the control of irrigation water by canals, sluices, brick conduits and ceramic pipes. During the Roman empire, extensive water systems using aqueducts, reservoirs and valves were constructed to carry water to cities. However, these early uses of fluid power required the movement of huge quantities of fluid because of the relatively low pressures provided by nature.

Fluid power technology actually began in 1650 with the discovery of Pascal's law. Simply stated, this law says that *pressure in a fluid at rest is transmitted undiminished equally in all directions in a confined body of fluid*. Pascal found that when he rammed a cork down into a jug completely full of wine, the bottom of the jug broke and fell out. However, in order for Pascal's law to be made effective for practical use, it was necessary to make a piston that would fit exactly. Not until over 100 years later was this accomplished. It was in 1795 that

Joseph Brahmah invented the cup packing that led to the development of a workable hydraulic press. Brahmah's hydraulic press consisted of a plunger pump piped to a large cylinder and a ram. This new hydraulic press found wide use in England because it provided a more effective and economical means of applying large force to industrial applications.

In 1750, Bernoulli developed his law of conservation of energy for a fluid flowing in a pipeline. Both Pascal's and Bernoulli's laws operate at the heart of all fluid power applications and are used for analytical purposes. However, it was not until the Industrial Revolution of 1850 in Great Britain that these laws were actually applied to the industry.

The first use of a large hydraulic press for foregoing work was made in 1860 by Whitworth. In the next 20 years, many attempts were made to reduce the waste and excessive maintenance costs of the original type of accumulator. In 1872, Rigg patented a three-cylinder hydraulic engine in which provision was made to change the stroke of plungers to vary its displacement without a throttle valve. In 1873, the Brotherhood three-cylinder, constant-stroke hydraulic engine was patented and was widely used for cranes, winches, etc. Both the above-mentioned engines were driven by fluid from an accumulator.

Up to this time, electrical energy was not developed to power the machines of industry. Instead, fluid power was being used to drive hydraulic equipment such as cranes, presses, shearing machines, etc. With electricity emerging dominantly in the 19th century, it was soon found superior to fluid power for transmitting power over great distances.

The modern era in fluid power began around the turn of the century. Fluid applications were made to such installations as the main armament system of USS Virginia in 1906. In these applications, a variable-speed hydrostatic transmission was installed to drive the main guns. Since that time, marine industry has applied fluid power to cargo-handling systems, controllable pitch controllers, submarine control system, aircraft elevators, aircraft- and missile-launching system and radar/sonar-driven systems. In 1926, the United States developed the first unitized, packaged hydraulic system consisting of a pump, controls and an actuator.

Today fluid power is used extensively in practically every branch of industry. The innovative use of modern technology such as electrohydraulic closed loops, microprocessors and improved materials for component construction continues to advance the performance of fluid power systems. The military requirements kept fluid power applications and developments going at a good pace. Aviation and aerospace industry provided the impetus for many advances in fluid power technology.

1.6 Advantages of a Fluid Power System

Oil hydraulics stands out as the prime moving force in machinery and equipment designed to handle medium to heavy loads. In the early stages of industrial development, mechanical linkages were used along with prime movers such as electrical motors and engines for handling loads. But the mechanical efficiency of linkages was very low and the linkages often failed under critical loading conditions. With the advent of fluid power technology and associated electronics and control, it is used in every industry now.

The advantages of a fluid power system are as follows:

1. **Fluid power systems are simple, easy to operate and can be controlled accurately:** Fluid power gives flexibility to equipment without requiring a complex mechanism. Using fluid power, we can start, stop, accelerate, decelerate, reverse or position large forces/components with great accuracy using simple levers and push buttons. For example, in Earth-moving equipment, bucket carrying load can be raised or lowered by an operator using a lever. The landing gear of an aircraft can be retrieved to home position by the push button.

2. **Multiplication and variation of forces:** Linear or rotary force can be multiplied by a fraction of a kilogram to several hundreds of tons.
3. **Multifunction control:** A single hydraulic pump or air compressor can provide power and control for numerous machines using valve manifolds and distribution systems. The fluid power controls can be placed at a central station so that the operator has, at all times, a complete control of the entire production line, whether it be a multiple operation machine or a group of machines. Such a setup is more or less standard in the steel mill industry.
4. **Low-speed torque:** Unlike electric motors, air or hydraulic motors can produce a large amount of torque while operating at low speeds. Some hydraulic and pneumatic motors can even maintain torque at a very slow speed without overheating.
5. **Constant force or torque:** Fluid power systems can deliver constant torque or force regardless of speed changes.
6. **Economical:** Not only reduction in required manpower but also the production or elimination of operator fatigue, as a production factor, is an important element in the use of fluid power.
7. **Low weight to power ratio:** The hydraulic system has a low weight to power ratio compared to electromechanical systems. Fluid power systems are compact.

8. **Fluid power systems can be used where safety is of vital importance:** Safety is of vital importance in air and space travel, in the production and operation of motor vehicles, in mining and manufacture of delicate products. For example, hydraulic systems are responsible for the safety of takeoff, landing and flight of aeroplanes and space craft. Rapid advances in mining and tunneling are the results of the application of modern hydraulic and pneumatic systems.

1.7 Basic Components of a Hydraulic System

Hydraulic systems are power-transmitting assemblies employing pressurized liquid as a fluid for transmitting energy from an energy-generating source to an energy-using point to accomplish useful work. Figure 1.1 shows a simple circuit of a hydraulic system with basic components.

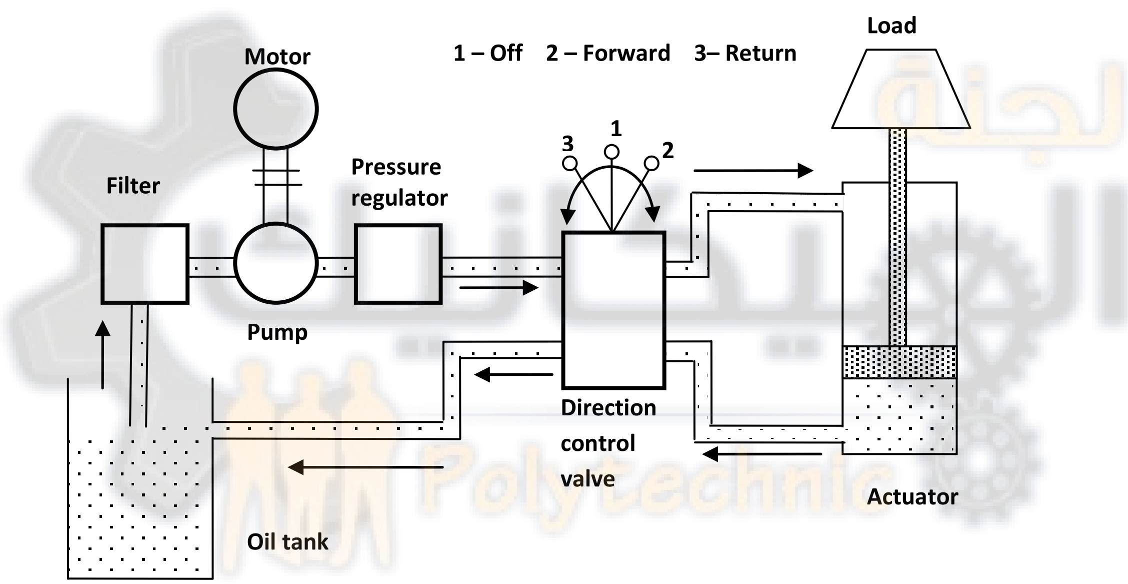


Figure 1.1 Components of a hydraulic system

Functions of the components shown in Fig. 1.1 are as follows:

1. The hydraulic actuator is a device used to convert the fluid power into mechanical power to do useful work. The actuator may be of the linear type (e.g., hydraulic cylinder) or rotary type(e.g., hydraulic motor) to provide linear or rotary motion, respectively.
2. The hydraulic pump is used to force the fluid from the reservoir to rest of the hydraulic circuit by converting mechanical energy into hydraulic energy.
3. Valves are used to control the direction, pressure and flow rate of a fluid flowing through the circuit.

4. External power supply (motor) is required to drive the pump.
5. Reservoir is used to hold the hydraulic liquid, usually hydraulic oil.
6. Piping system carries the hydraulic oil from one place to another.
7. Filters are used to remove any foreign particles so as keep the fluid system clean and efficient, as well as avoid damage to the actuator and valves.
8. Pressure regulator regulates (i.e., maintains) the required level of pressure in the hydraulic fluid.

The piping shown in Fig. 1.1 is of closed-loop type with fluid transferred from the storage tank to one side of the piston and returned back from the other side of the piston to the tank. Fluid is drawn from the tank by a pump that produces fluid flow at the required level of pressure. If the fluid pressure exceeds the required level, then the excess fluid returns back to the reservoir and remains there until the pressure acquires the required level.

Cylinder movement is controlled by a three-position change over a control valve.

1. When the piston of the valve is changed to upper position, the pipe pressure line is connected to port A and thus the load is raised.

2. When the position of the valve is changed to lower position, the pipe pressure line is connected to port B and thus the load is lowered.

3. When the valve is at center position, it locks the fluid into the cylinder(thereby holding it in position) and dead-ends the fluid line (causing all the pump output fluid to return to tank via the pressure relief).

In industry, a machine designer conveys the design of hydraulic systems using a circuit diagram. Figure 1.2 shows the components of the hydraulic system using symbols. The working fluid, which is the hydraulic oil, is stored in a reservoir. When the electric motor is switched ON, it runs a positive displacement pump that draws hydraulic oil through a filter and delivers at high pressure. The pressurized oil passes through the regulating valve and does work on actuator. Oil from the other end of the actuator goes back to the tank via return line. To and fro motion of the cylinder is controlled using directional control valve.

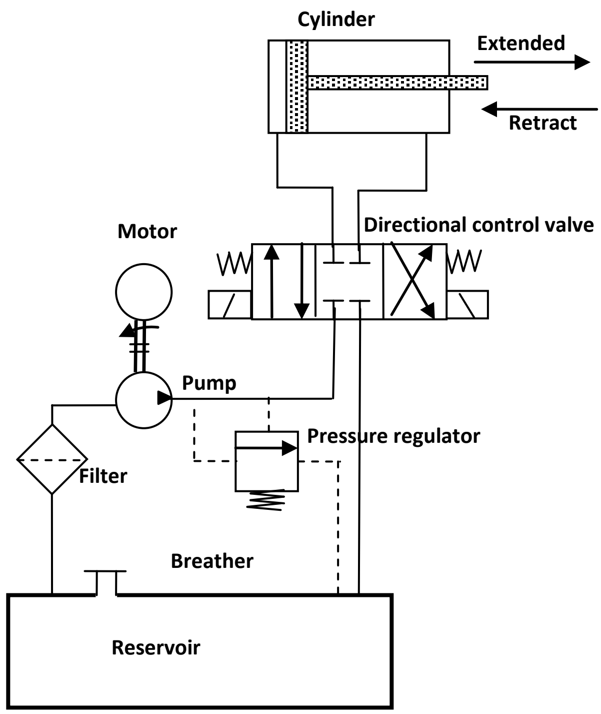


Figure 1.2 Components of a hydraulic system (shown using symbols).

The hydraulic system discussed above can be broken down into four main divisions that are analogous to the four main divisions in an electrical system.

1. The power device parallels the electrical generating station.

2. The control valves parallel the switches, resistors, timers, pressure switches, relays, etc.

3. The lines in which the fluid power flows parallel the electrical lines.

4. The fluid power motor (whether it is a rotating or a non rotating cylinder or a fluid power motor) parallels the solenoids and electrical motors.

1.8 Basic Components of a Pneumatic System

A pneumatic system carries power by employing compressed gas, generally air, as a fluid for transmitting energy from an energy-generating source to an energy-using point to accomplish useful work. Figure 1.3 shows a simple circuit of a pneumatic system with basic components.

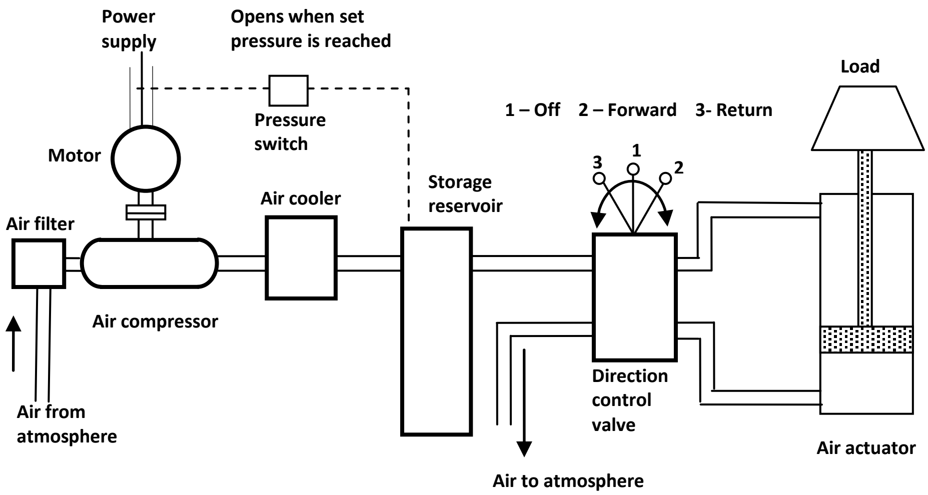


Figure 1.3 Components of a pneumatic system.

The functions of various components shown in Fig. 1.3 are as follows:

1. The pneumatic actuator converts the fluid power into mechanical power to perform useful work.
2. The compressor is used to compress the fresh air drawn from the atmosphere.
3. The storage reservoir is used to store a given volume of compressed air.
4. The valves are used to control the direction, flow rate and pressure of compressed air.
5. External power supply (motor) is used to drive the compressor.
6. The piping system carries the pressurized air from one location to another.

Air is drawn from the atmosphere through an air filter and raised to required pressure by an air compressor. As the pressure rises, the temperature also rises; hence, an air cooler is provided to cool the air with some preliminary treatment to remove the moisture. The treated pressurized air then needs to get stored to maintain the pressure. With the storage reservoir, a pressure switch is fitted to start and stop the electric motor when pressure falls and reaches the required level, respectively.

The three-position change over the valve delivering air to the cylinder operates in a way similar to its hydraulic circuit.

1.9 Comparison between Hydraulic and Pneumatic Systems

Usually hydraulic and pneumatic systems and equipment do not compete. They are so dissimilar that there are few problems in selecting any of them that cannot be readily resolved. Certainly, availability is one of the important factors of selection but this may be outweighed by other factors. In numerous instances, for example, air is preferred to meet certain unalterable conditions, that is, in“hot spots” where there is an open furnace or other potential ignition hazard or in operations where motion is required at extremely high speeds. It is often found more efficient to use a combined circuit in which oil is used in one part and air in another on the same machine or process. Table 1.2 shows a brief comparison of hydraulic and pneumatic systems.

Table 1.2 Comparison between a hydraulic and a pneumatic system</table>

S. No.	Hydraulic System	Pneumatic System
1.	It employs a pressurized liquid as a fluid	It employs a compressed gas, usually air, as a fluid
2.	An oil hydraulic system operates at pressures up to 700 bar	A pneumatic system usually operates at 5–10 bar
3.	Generally designed as closed system	Usually designed as open system
4.	The system slows down when leakage occurs	Leakage does not affect the system much
5.	Valve operations are difficult	Valve operations are easy
6.	Heavier in weight	Lighter in weight
7.	Pumps are used to provide pressurized liquids	Compressors are used to provide compressed gases
8.	The system is unsafe to fire hazards	The system is free from fire hazards
9.	Automatic lubrication is provided	Special arrangements for lubrication are needed

1.10 Comparison of Different Power Systems

There are three basic methods of transmitting power: electrical, mechanical and fluid power. Most applications actually use a combination of the three methods to obtain the most efficient overall system. To properly determine which method to use, it is important to know the salient features of each type. For example, fluid systems can transmit power more economically over greater distances than mechanical types. However, fluid systems are restricted to shorter distances compared to electrical systems. Table 1.3 lists the salient features of each type.

Table 1.3 Comparison of different power systems</table>

Property	Mechanical	Electrical	Pneumatic	Hydraulic
Input energy source	I C engines Electric motor	I C engines Water/gas turbines	I C engines Pressure tank	I C engines Electric motor Air turbine
Energy transfer element	Levers, gears, shafts	Electrical cables and magnetic field	Pipes and hoses	Pipes and hoses
Energy carrier	Rigid and elastic objects	Flow of electrons	Air	Hydraulic liquids
Power-to-weight ratio	Poor	Fair	Best	Best
Torque/inertia	Poor	Fair	Good	Best
Stiffness	Good	Poor	Fair	Best
Response speed	Fair	Best	Fair	Good
Dirt sensitivity	Best	Best	Fair	Fair
Relative cost	Best	Best	Good	Fair
Control	Fair	Best	Good	Good
Motion type	Mainly rotary	Mainly rotary	Linear or rotary	Linear or rotary

1.11 Future of Fluid Power Industry in India

The automation market in India is estimated to be 1/10ththat of China. If India has to become one of the leading economies in the world, based on manufacturing, it will have to attain higher technological standards and higher level of automation in manufacturing.

In the past 30 years, fluid power technology rose as an important industry. With increasing emphasis on automation, quality control, safety and more efficient and green energy systems, fluid power technology should continue to expand in India.

Fluid power industry is gaining a lot of importance in Indian industry. According to a recent survey, it has shown a growth of 20% over the last 10 years and the size of market is estimated to be close to 5000 crores per annum. This makes it a sizable industry segment in India. The growth rate of this industry in India is typically about twice the growth of economy.

The reasons for this are three-fold:

1. As the economy grows, this industry grows.

2. There is a lot of automation and conversion into more sophisticated manufacturing methods which increases the rate.

3. One of the interesting things happening in this industry is that India is becoming an attractive destination for manufacturing and outsourcing of some of the products.

So these three aspects together create a situation where the growth of this industry is twice the growth of GDP in India.

The fluid power sector in India consists of many sophisticated Indian industries and partnership with number of global fluid power technology leaders that include Festo, Rexroth, Vickers, Eaton, Parker Hannifin, Norgen, , Saucer Donfos, Yuken, Siemens, Shamban, Pall and Gates, , Rotex, , Janatics, Maxwell, Wipro Dynamatic Technologies and many more.

One of the major segments for hydraulic industry in India is mobile hydraulics. Because of massive programs on road construction, there is a major expansion of construction machinery industry as well. In addition to this, a trend toward the usage of more sophisticated hydraulics in tractors and farm equipment is witnessed. The manufacturing industry in India is working toward higher automation and quality of output. As Indian industry moves toward modernization to meet the productivity and to compete in the global market, an excellent potential for the pneumatic industry is expected in India.

Another area of interest for fluid power industry would be the opportunities in defense equipment. Defense is a major market segment in Indian fluid power industry and contributes to over 40% of the market demand. There is also a move toward products with miniature pneumatics, process valves, servo drives, hydraulic power steering with new controls and sophisticated PLC, microprocessor controls.

However, the key input required for the effective utilization of fluid power is education and training of users. So there is a big need for education and training in design application and maintenance of fluid power systems. Rexroth recently opened many competence centers in India to train the manpower and to create awareness about the use of fluid power in Indian industry.

Objective Type Questions

Fill in the Blanks

1. Fluid power is the technology that deals with the generation, __________and transmission of forces and movement of mechanical elements or systems.

2. The main objective of fluid transport systems is to deliver a fluid from one location to another, whereas fluid power systems are designed to perform __________.

3. There are three basic methods of transmitting power: Electrical, mechanical and __________.

4. Only _________are capable of providing constant force or torque regardless of speed changes.

5. The weight-to-power ratio of a hydraulic system is comparatively __________than that of an electromechanical system.

State True or False

1. Hydraulic lines can burst and pose serious problems.

2. Power losses and leakages are less in pneumatic systems.

3. Pneumatic system is not free from fire hazards.

4. Hydraulic power is especially useful when performing heavy work.

5. Water is a good functional hydraulic fluid.

Review Questions

1. Define the term fluid power.
2. Differentiate between fluid transport and fluid power systems.
3. Differentiate between hydraulics and pneumatics.
4. List the six basic components used in a hydraulic system.
5. List the six basic components used in a pneumatic system.
6. List 10 applications of fluid power in the automotive industry.
7. Name 10 hydraulic applications and 10 pneumatic applications.
8. List five advantages and five disadvantages of hydraulics.
9. List five advantages and five disadvantages of pneumatics.
10. List the main components of a fluid power system and their functions.
11. Discuss in detail the future of fluid power industry in India.
12. Compare different power systems used in industries.
13. What is the main difference between an open-loop and a closed-loop fluid power system?
14. List five major manufactures of fluid power equipment and systems in India.
15. List five major manufactures of fluid power equipment and systems in the world.
16. Visit any industry nearby and list the hydraulic/pneumatic parts or systems used and their purposes.
17. Why is the hydraulic power especially useful when performing heavy work?
18. Differentiate between oil hydraulics and pneumatics.
19. List any five applications of fluid power systems.

20 List the main components of a fluid power system and their functions.

Answers

Fill in the Blanks

1. Control
2. Work
3. Fluid power
4. Fluid power systems
5. Less

State True or False

1. True
2. True
3. False
4. True
5. False

Lecture 2

PROPERTIES OF FLUID

Learning Objectives

Upon completion of this chapter, the student should be able to:

- Define three states of matter: Solid, liquid and gas.
- Define mass density, specific weight and weight density.
- Understand the meaning of the term pressure.
- Explain the difference between gauge and absolute pressures.
- Understand the difference between kinematic and absolute viscosities.
- Differentiate between the ideal and real fluids.
- Define bulk modulus
- Explain the dependence of viscosity on temperature.

1.1 Introduction

Fluids, both liquids and gases, are characterized by their continuous deformation when a shear force, however small, is applied. Liquids and gases may be distinguished by their relative incompressibilities and the fact that liquid may have a free surface while a gas expands to fill its confining container. Because the liquid is the medium of transmission of power in hydraulic system, knowledge of its characteristics is essential. The purpose of this chapter is to define certain fundamental properties of fluids which will be useful to apply the basic principles of fluid mechanics to the solution of practical problems on fluid power.

1.2 Solids and Fluids (Liquids and Gases)

1.2.1 Distinction between a Solid and a Fluid

The molecules of a solid are usually closer to each other than those of a fluid. The attractive forces between the molecules of a solid are so large that a solid tends to retain its shape. This is not the case with a fluid where the attractive forces between the molecules are smaller. An ideal elastic solid deforms under load, and once the load is removed, it returns to its original state.

1.2.2 Distinction between a Gas and a Liquid

A fluid may be either a gas or a liquid. The molecules of a gas are much farther apart than those of a liquid. Hence, a gas is very compressible, and when all external pressure is removed, it tends to expand indefinitely. A liquid is relatively incompressible, and if all pressure, except its own vapor pressure, is removed, the cohesion between molecules holds them together, so that the liquid does not expand indefinitely. Therefore, a liquid may have a free surface, that is, a surface from which all pressure is removed, except its own vapor pressure.

1.3 Density, Specific Weight, Specific Volume and Specific Gravity

1.3.1 Density

Density (ρ) is defined as mass per unit volume, that is,

$$\rho = \frac{\text{Mass}}{\text{Volume}} = \frac{m}{V}\left(\frac{\text{kg}}{\text{m}^3}\right) \tag{1.1}$$

The density characteristics of typical hydraulic fluids are given in Table 1.1.

Table 1.1 Density characteristics of hydraulic fluids

Fluid	**Density (kg/m^3)**
Shell Tellus ISO 32 mineral oil	875
Shell HFB 60%oil,40%oil	933
Shell HFC 60%glycol,40%water	1084
Shell HFD phosphate ester	1125
Shell Naturelle HFE 32	918

It is well known that density increases with pressure and decreases with temperature. Figure 1.1shows such a characteristic for an ISO 32 mineral oil.

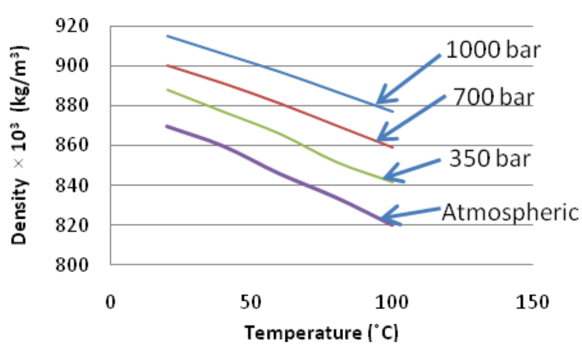


Figure 1.1 Characteristics of ISO 32 mineral oil.

1.3.2Specific Weight

Specific weight (γ) is defined as weight per unit volume, that is,

$$\gamma = \frac{\text{Weight}}{\text{Volume}}\left(\frac{\text{N}}{\text{m}^3}\right) \tag{1.2}$$

Density and specific weight are related by

$$\rho = \frac{\gamma}{g} \;\Rightarrow \gamma = \rho g$$

where g is acceleration due to gravity. Now

$$\text{Dimension of } \rho = \frac{\text{Dimension of } \gamma}{\text{Dimension of } g} = \frac{\mathrm{N/m^3}}{\mathrm{m/s^2}} = \frac{(\mathrm{kgm/s^2})/\mathrm{m^3}}{\mathrm{m/s^2}} = \frac{\mathrm{kg}}{\mathrm{m^3}}$$

Note that the density (ρ) is absolute, since it depends on mass, which is independent of location. The specific weight (γ), on the other hand, is not absolute, since it depends on the value of gravitational acceleration (g), which varies with location, primarily latitude and elevation above mean sea level. The densities and specific weights of fluids vary with temperature. Table 1.2 gives specific weight of common fluids. The specific weight of a liquid varies only slightly with pressure, depending on the bulk modulus of the liquid; it also depends on temperature and the variation may be considerable. Since the specific weight (γ) is equal to ρg, the specific weight of a fluid depends on the local value of the acceleration due to gravity in addition to the variation in temperature and pressure.

Table 1.2 Specific weight(γinkN/m^3) of common liquids at 20°C, 1013 millibar abs with g = 9.81 m/s^2

Carbon tetrachloride	15.6
Ethyl alcohol	7.76
Gasoline	6.6
Glycerin	12.3
Kerosene	7.9
Motor oil	8.5
Seawater	10.03
Water	9.79

1.3.3Specific Volume

Specific volume (SV) is the volume occupied by a unit mass of fluid. We commonly apply it to gases and usually express it in m^3/kg. Specific volume is the reciprocal of density. Thus,

$$\mathrm{SV} = \frac{1}{\rho} \qquad (1.3)$$

1.3.4Specific Gravity

Specific gravity (SG) of a given fluid is defined as the specific weight of the fluid divided by the specific weight of water, that is

$$\mathrm{SG}_{\mathrm{oil}} = \frac{\gamma_{\mathrm{oil}}}{\gamma_{\mathrm{water}}}$$

$$\Rightarrow \mathrm{SG}_{\mathrm{oil}} = \frac{\rho_{\mathrm{oil}}}{\rho_{\text{water at standard temperature}}} \tag{1.4}$$

SG of a liquid is a dimensionless ratio. Physicists use 4°C as the standard temperature, but engineers often use 15.56°C. In the metric system, the density of water at 4°C is 1.00 g/cm^3, equivalent to 1000 kg/m^3, and hence the SG (which is dimensionless) of a liquid has the same numerical value as its density expressed in g/mL or mg/m^3.

The SG of a gas is the ratio of its density to that of either hydrogen or air at some specified temperature and pressure, but there is no general agreement on these standards, and so we must explicitly state them in any given case. Since the density of a fluid varies with temperature, we must determine and specify specific gravities at a particular temperature.

Example 1.1

Air at 20°C and atmospheric pressure has a density of 1.23 kg/m^3. Find its specific gravity. What is the ratio of the specific gravity of water to the specific gravity of air at 20°C and atmospheric pressure? What is the significance of the ratio?

Solution: Given$T$ = 20°C, air density $(\rho_{\mathrm{air}}) = 1.23\,\mathrm{kg/m^3}$, water density $(\rho_{\mathrm{water}}) = 1000\,\mathrm{kg/m^3}$. Specific gravity of air is given by

$$\frac{\rho_{\mathrm{air}}}{\rho_{\mathrm{water}}} = 1.23\times10^{-3}$$

We know that specific gravity of water = 1. So

$$\frac{\text{Specific gravity of air}}{\text{Specific gravity of water}} = \frac{1}{1.23\times10^{-3}} = 813$$

Therefore, water is 813 times heavier than air.

Example1.2

The specific weight of oil mixture at ordinary pressure and temperature is 9.81 kN/m^3. The specific gravity of mercury is 13.56. Compute the density of oil mixture and the specific weight and density of mercury.

Solution: Given specific weight of oil mixture γ_{oil} = 9.81. The density of oil is

$$\rho_{\text{oil}} = \frac{\gamma_{\text{oil}}}{g} = \frac{9.81}{9.81} = 1.00\,\text{mg/m}^3 = 1.00\ \text{g / mL}$$

The specific weight and density of mercury are, respectively,

$$\gamma_{\text{mercury}} = S_{\text{mercury}}\gamma_{\text{oil}} = 13.56(9.81) = 133.0\,\text{kN / m}^3$$

$$\rho_{\text{mercury}} = S_{\text{mercury}}\rho_{\text{oil}} = 13.56\,(1.00) = 13.56\,\text{kN / m}^3$$

Example 1.3
A cylinder container has a diameter of 0.5 m and a height of 1 m. If it is to be filled with a liquid having a specific weight of 2000 N/m^3, how many kg of this liquid must be added?

Solution: Given diameter $(d) = 0.5$ m, height $(h) = 1$ m. The volume is given by

$$\text{Volume } (V) = \frac{\pi d^2 h}{4} = 0.19635\,\text{m}^3$$

Also it is given that specific weight (γ) is 2000 N/m^3. Now

$$\text{Weight} = V \times \gamma = 0.19635 \times 2000 = 392.7\ \text{N}$$

The mass is given by

$$\text{Mass} = \frac{W}{g} = 40\ \text{kg}$$

Example 1.4

One liter of SAE30 oil weighs 8.70 N. Calculate its specific weight, density and specific gravity.

Solution: Given volume $(V) = 1\ \text{liter} = 10^{-3}\ \text{m}^3$, weight $(W) = 8.7$ N. The specific weight is

$$\gamma = \frac{W}{V} = \frac{8.7}{10^{-3}} = 8700\,\text{N / m}^3$$

and mass density is

$$\rho_{\text{oil}} = \frac{\gamma}{g} = 887.755\ \text{kg / m}^3$$

Therefore,

$$\text{SG}_{\text{oil}} = \frac{\rho_{\text{oil}}}{\rho_{\text{water at standard temperature}}} = \frac{8878.75}{1000} = 0.889$$

1.4 **Pressure**

Pressure is defined as force per unit area. It is the amount of force acting over a unit area, that is

$$P = \frac{\text{Force}}{\text{Area}} = \frac{F}{A} \tag{1.5}$$

The pressure developed at the bottom of a column of any liquid is called hydrostatic pressure and is given by

$$P = \rho g h \tag{1.6}$$

where P is the hydrostatic pressure or the pressure at the bottom of liquid column in Pascal or N/m^2, ρ is the density of liquid in kg/m^3, g is the acceleration due to gravity in m/s^2 and h is the level of the column of liquid in meters.

Since

$$P = \rho g h = \frac{mgh}{V} = \frac{Wh}{V}$$

and

$$P = \gamma h$$

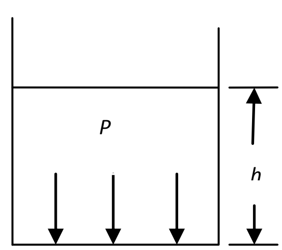
where γ is the specific weight of liquid in N/m^3, we have $P\ (N/m^2) = \gamma\ (N/m^3) \times h\ (m)$.

1.4.1 Pressure at the Bottom of a Column of Liquid

Let us now refer to Fig. 1.2, which shows the pressure head developed at the bottom due to the column of liquid. Let h be the height of the liquid column and W be the weight of the liquid. Let the liquid have a specific weight γ and volume V:

$$P = \frac{F}{A} = \frac{W}{A} = \frac{W}{A} \times \frac{V}{V} = \frac{W}{V} \times \frac{V}{A}$$

$$= \frac{(\gamma \times V)}{A} = \frac{(\gamma \times A \times H)}{A}$$



$$\Rightarrow P = \gamma H \tag{1.7}$$

Figure 1.2 Pressure developed by a column of fluid.

Observing the equation, it can be concluded that pressure does not depend on the cross-sectional area of the liquid column but only on the column height and specific weight of the liquid. Changing the cross-sectional area of the liquid column changes the weight of the liquid by a proportional amount. Hence, F/A (pressure) remains constant.

1.4.2 Atmospheric Pressure and Absolute Pressure

Atmospheric pressure is the force per unit area exerted against a surface by the weight of air above that surface in the Earth's atmosphere. In most circumstances, atmospheric pressure is closely approximated by the hydrostatic pressure caused by the weight of air above the measurement point. Low-pressure areas have less atmospheric mass above their location, whereas high-pressure areas have more atmospheric mass above their location. Similarly, as elevation increases, there is less overlying atmospheric mass, such that the pressure decreases with increasing elevation. The standard atmosphere is a unit of pressure that is equal to 101325 Pa or 101.325 kPa. The following units are equivalent, but only to the number of decimal places displayed: 760 mmHg (torr), 29.92 in Hg, 14.696 psi, 1013.25 mbar/hPa. One standard atmosphere is the standard pressure used for pneumatic fluid power (ISO R554) and in the aerospace (ISO 2533) and petroleum (ISO 5024) industries. Absolute pressure is measured relative to a perfect vacuum such as that existing in outer space.

1.4.3 **Gauge pressure and absolute pressure**

Gauge pressure is measured relative to the atmosphere, whereas absolute pressure is measured relative to a perfect vacuum such as that existing in outer space.

A chart showing the difference between gauge and absolute pressure is given in Fig. 1.3. Let us examine the two pressure levels P_1 and P_2.

1. Relative to a prefect vacuum, they are:
- $P_1 = 0.7$ bar (absolute), that is, a pressure less than an atmospheric pressure.
- $P_2 = 2$ bar (absolute), that is, a pressure greater than an atmospheric pressure.

2. Relative to atmosphere, they are:
- $P_1 = -0.3$ bar (gauge, suction or vacuum).
- $P_2 = 1$ bar (gauge).

As can be seen from Fig. 1.3, the following rule can be used in pressure conversion calculations:

$$\text{Absolute pressure} = \text{Gauge pressure} + \text{Atmospheric pressure}$$

It should be noted that vacuum or suction pressures exist in a certain location of fluid power systems (e.g., in the inlet or suction lines of pumps). Therefore, it is important to understand the meaning of pressures below atmospheric pressure. One way to generate a suction pressure is to remove some of the fluid from a closed vessel initially containing fluid at atmospheric pressure.

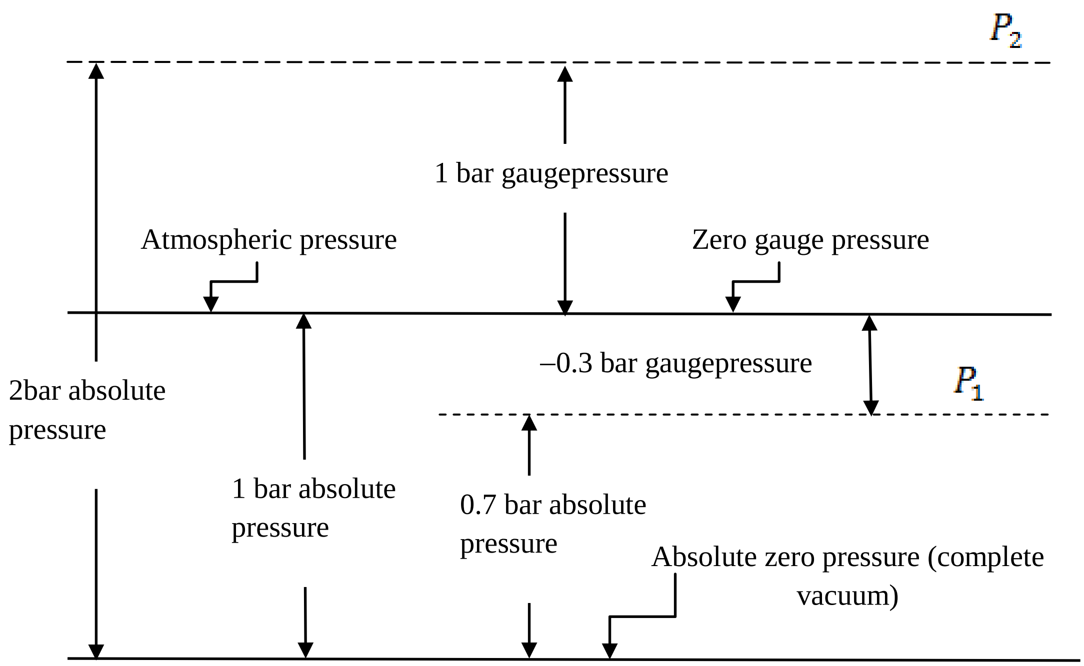


Figure 1.3 Difference between absolute and gauge pressure.

Example 1.5

For the fluid power automobile lift system of Fig. 1.4, the hydraulic piston has a 250-mm diameter. How much of oil pressure (kPa) is required to lift a 13300 N automobile?

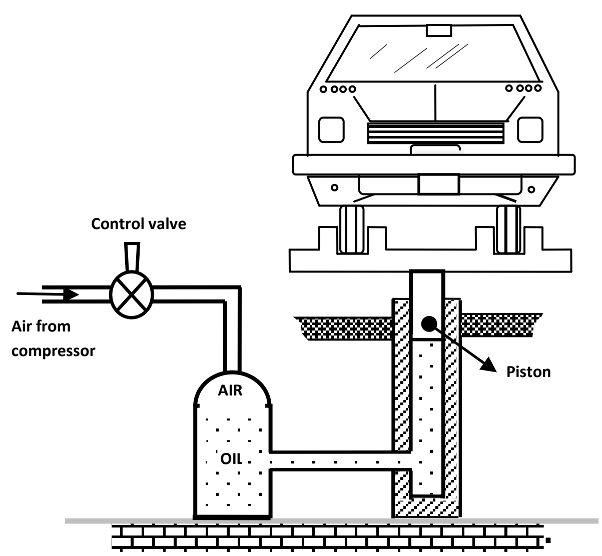


Figure 1.4

Solution: Given force F = 13300 N, diameter (d) = 0.25 m, area (A) = 0.0491 m^2.The gauge pressure is

$$\text{Pressure}\,(P) = \frac{F}{A} = \frac{13300}{250} = 270.876\ \text{kPa}$$

Example 1.6

For the fluid power automobile lift system of Fig. 1.5, the air pressure equals 550 kPagauge. If the hydraulic piston has a diameter of 250 mm, what is the maximum weight of an automobile that can be lifted? The specific gravity of oil is 0.9. What percentage error in the answer to this problem occurs by ignoring the 1-m head of oil to between the air and interface surface and bottom surface of the piston? $\text{Density}\ (\rho) = 900\ \text{kg}/\text{m}^3$.

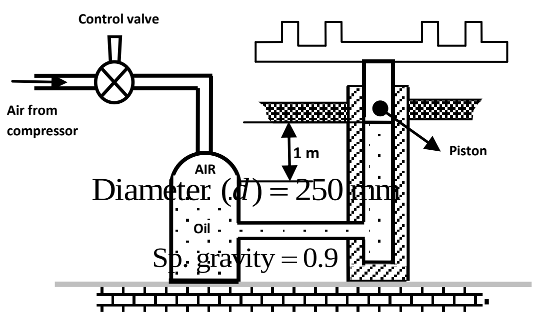


Figure 1.5

Solution: Given pressure of

air $(P_{\text{air}}) = 550\ \text{kPa}$. We know that

$$P_{\text{air}} = \text{Pressure at piston}\ (P_{\text{piston}}) + \text{Pressure due to 1 m oil head}$$

$$\Rightarrow 550 \times 10^3 = P_{\text{piston}} + \rho g h$$

$$\Rightarrow P_{\text{piston}} = 550 \times 10^3 - 900 \times 9.81 \times 1 = 541.18\ \text{kPa}$$

If we ignore 1-m oil head, then $P_{\text{piston}} = 550\ \text{kPa}$. Then, error in solution is calculated as

$$\%\text{error} = \frac{550 - 541.18}{541.18} \times 100 = 1.63\%$$

1.5 Compressible and Incompressible Fluids

Fluid power deals with both incompressible and compressible fluids, that is, with oil and air of either constant or variable density. Although there is no such thing in reality as an incompressible fluid, we use this term where the change in density with pressure is so small as to be negligible. This is usually the case with liquids. We may also consider gases to be incompressible when the pressure variation is small compared with the absolute pressure.

Ordinarily, we consider liquids to be incompressible fluids; yet sound waves, which are really pressure waves, travel through them. This provides the evidence of the elasticity of liquids.

The flow of air in a ventilating system is a case where we may treat a gas as incompressible, for the pressure variation is so small that the change in density is of no importance. However, for a gas or steam flowing at a high velocity through a long pipeline, the decrease in pressure may be so high that we cannot ignore the change in density. For an airplane flying at a speed below 100 m/s, we may consider the air to be of constant density. But as an object moving through air approaches the velocity of sound, which is of the order of 1200 km/h depending on temperature, the pressure and density of the air adjacent to the body become materially different from those of the air at some distance away.

1.6 Bulk Modulus (Volume Modulus of Elasticity)

Bulk modulus is a measure of the compressibility of a liquid and is required when it is desired to calculate oil volume changes for high pressure and large system volumes such as forging pressures or natural frequencies generally caused by the interaction of fluid compressibility and moving mass. Bulk modulus is analogous to the modulus of elasticity for solids; however, for fluids, it is defined on a volume basis rather than in terms of the familiar one-dimension stress–strain relation. The compressibility (a change in volume due to a change in pressure) of liquid is inversely proportional to its bulk modulus. For liquids, the value of bulk modulus is 1.72×10^6kPa. The volume modulus of mild steel is about 170000 MPa. Taking a typical value for the volume modulus of cold water to be 2200 MPa, we see that water is about 80 times as compressible as steel. The compressibility of liquids covers a wide range. Mercury, for example, is approximately 8% as compressible as water, whereas the compressibility of nitric acid is nearly six times that of water.

Example 1.7

A 500 cm^3 sample of oil is to be compressed in a cylinder until its pressure is increased from 1 to 50 atm. If the bulk modulus of oil equals 1750 MPa, find the percentage change in its volume.

Solution: Given volume (V) = 500 cm^3, $P_1 = 1$ atm, $P_2 = 50$ atm and $B = 1750$ MPa. We know that

$$B = \frac{-\Delta P}{\Delta V / V}$$

$$\Rightarrow \frac{\Delta V}{V} = \frac{-\Delta P}{B} = \frac{-49 \times 1.013 \times 1000000}{1750000000}$$

Reduction in volume = −0.2836 %, which implies that oil is incompressible.

Example 1.8

A positive displacement pump with a delivery of 1 L/min is fed into a pipe with a total volume of 1 L. If the end of the pipe is suddenly blocked, calculate the rise in pressure after 1 s. (The bulk modulus of the fluid being pumped may be taken as 2000 MPa; neglect any change in the volume of the pipe.)

Solution: Bulk modulus is given as

$$B=\frac{-\Delta P}{\Delta V/V}$$

where ΔP is the change in pressure, ΔV is the change in volume and V is the original volume. Now

$$\Delta V = \text{Pump flow in 1 s} = \frac{1}{60}\text{ L}$$

Hence,

$$\Delta P = B\left(\frac{\Delta V}{V}\right) = \frac{2000}{60} = 33.3\text{ MPa}$$

This rapid rise in pressure illustrates the necessity of having some form of control to limit the rise in pressure in a system should a pump be deadheaded. The control may be built into the pump or may be an external pressure-limiting device such as a relief valve.

Example 1.9

An 8 L sample of oil is compressed in a cylinder until pressure increases from 0.7 to 2.7 MPa. If the bulk modulus equals 80 MPa, find the change in the volume of oil.

Solution: Given initial volume V = 8 L = 0.008 m^3 and change in pressure ΔP =2.7 − 0.7 = 2 MPa. Bulk modulus is given by

$$B=\frac{-\Delta P}{\Delta V/V}$$

So, change in volume

$$\Delta V = -V\left(\frac{\Delta P}{B}\right) = -0.008\left(\frac{2}{80}\right) = -0.002\text{ m}^3$$

1.7 Reynolds Number

It is a dimensionless number referred to as a compressible or incompressible fluid flow. It was postulated by a British engineer Osborne Reynolds. The Reynolds number set criteria by which the fluid flow regime may be distinguished:

$$Re = \frac{\rho v D}{\mu}$$

where ρ is the density (kg/m^3), v is the velocity of fluid (m/s), D is the diameter of the pipe (m) and μ is the absolute or dynamic viscosity (Pa s or ms/m^2).

1.8 Types of Fluid Flow

Based on the range of Reynolds number, the flow of fluid is classified as laminar flow, transition flow and turbulent flow.

1. **Laminar flow:** In the laminar flow region, the flow is characterized by the smooth motion of the laminae or layers. When there is no macroscopic mixing of adjacent fluid layers for the flow in the laminar regimes, the Reynolds number is less than 2000.

2. **Turbulent flow:** In the turbulent flow region, the flow is characterized by the random motion of the fluid particles in three dimensions in addition to mean motion. There is considerable macroscopic mixing of adjacent fluid layers and significant velocity fluctuations. For the turbulent flow, the Reynolds number is greater than 4000.

3. **Transition flow:** In the transition flow region, the flow is in transition between laminar and turbulent flows. The Reynolds number lies between 2000 and 4000.

1.9 Ideal Fluid

An ideal fluid is usually defined as a fluid in which there is no friction; it is inviscid (its viscosity is zero). Thus, the internal forces at any section within it are always normal to the section, even during motion. So these forces are purely pressure forces. Although such a fluid does not exist in reality, many fluids approximate frictionless flow at a sufficient distance from solid boundaries, and so we can often conveniently analyze their behaviors by assuming an ideal fluid.

In a real fluid, either liquid or gas, tangential or shearing forces always develop whenever there is a motion relative to body, thus creating fluid friction, because these forces oppose the motion of one particle past another. These friction forces give rise to a fluid property called viscosity.

1.10 Viscosity

The viscosity of a fluid is a measure of its resistance to shear or angular deformation. Motor oil, for example, has a high viscosity and resistance to shear; it is cohesive and feels “sticky,” whereas gasoline has a low viscosity. The friction forces in a flowing fluid result from the cohesion and momentum which are interchangeable between molecules. The viscosity of typical fluids depends on temperature. Figure 1.7 indicates how the viscosities of typical fluids depend on temperature. As the temperature increases, the viscosities of all liquids decrease, whereas the viscosities of all gases increase. This is because of the force of cohesion, which diminishes with temperature, predominates with liquids, whereas with gases, the predominating factor is the interchange of molecules between the layers of different velocities. Thus, a rapidly moving gas molecule shifting into a slower moving layer tends to speed up the latter, and a slow-moving molecule entering a faster moving layer tends to slow down the faster moving layer. This molecular interchange sets up a shear or produces a friction force between adjacent layers. At higher temperatures, molecular activity increases, thereby causing the viscosity of gases to increase with temperature.

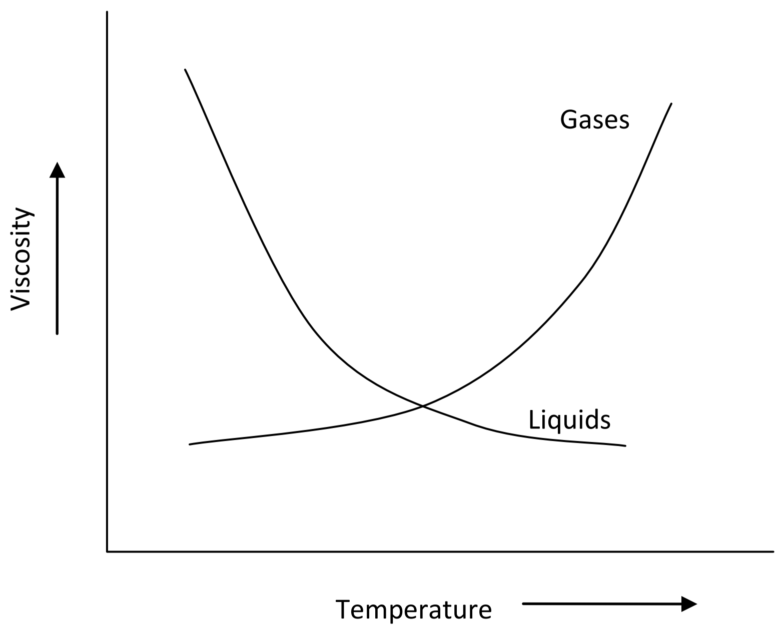
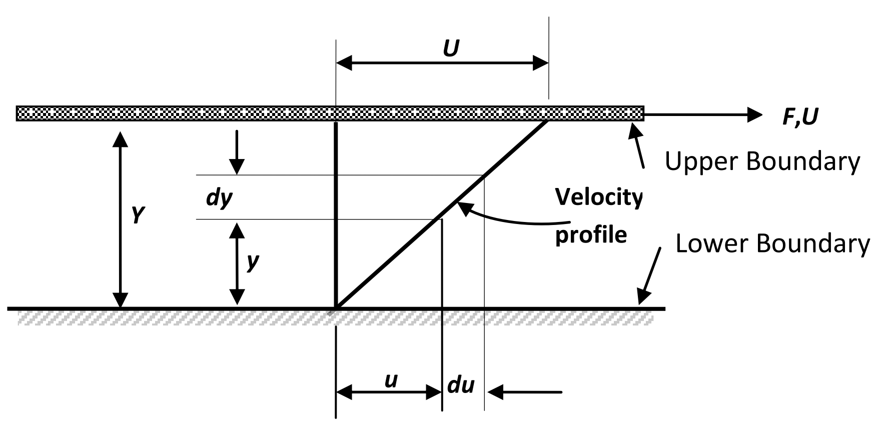
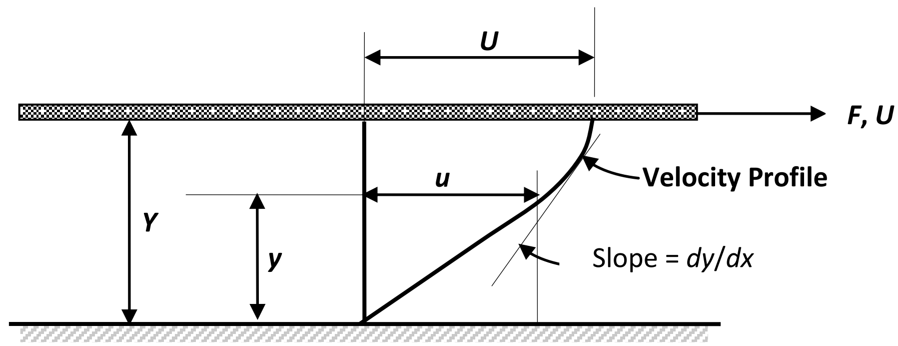


Figure 1.7 Trends in viscosity variation with temperature.

Consider a classic case of two parallel plates (Fig. 1.8), sufficiently large that we can neglect edge conditions, a small distance *Y* apart, with fluid filling the space in between. The lower plate is stationary whereas the upper one moves parallel to it with a velocity *U* due to a force *F* corresponding to some area *A* of the moving plate.



(a)



(b)

Figure 1.8 Velocity profiles across parallel plates: (a) Linear (no bulk flow) and (b) linear (bulk flow to right).

At the boundaries, the particles of a fluid adhere to the walls, so their velocities are zero relative to the walls. This is the so-called no-slip condition that occurs in all viscous fluids. Thus, in Fig. 1.8, the fluid velocities must be U when in contact with the plate at the upper boundary and zero at the lower boundary. We call the form of velocity variation with distance between these two extremes, as depicted in Fig. 1.8, a *velocity profile*. If the separation distance Y is not too large, the velocity profile is linear, as shown in Fig. 1.8(a). If, in addition, there is a small amount of bulk fluid transport between the plates, which could result from pressure-fed lubrication, for example, the velocity profile becomes the sum of the previous linear profile and a parabolic profile as shown in Fig. 1.8(b); the parabolic additions to the linear profile are zero at the walls and maximum at the centerline. The behavior of the fluid is much as if it is consisted of a series of thin layers, each of which slips a little relative to the next.

For a large class of fluids under the conditions of Fig. 1.8(a), experiments have shown that

$$F \propto \frac{AU}{Y}$$

We see from similar triangles that we can replace U/Y by the velocity gradient du/dy. If we now introduce a constant of proportionality μ, we can express the shearing stress τ between any two thin sheets of fluid by

$$\tau = \frac{F}{A} = \mu \frac{U}{Y} = \mu \frac{du}{dy} \tag{1.8}$$

This equation is called Newton's equation of viscosity, since Sir Isaac Newton (1642–1727) first suggested it. Although better known for this formulation of the fundamental laws of motion and gravity for the development of differential calculus, Newton, an English mathematician and natural philosopher, also performed many pioneering studies in fluid mechanics. In transposed form, Eq. (1.8) defines the proportionality constant

$$\mu = \frac{\tau}{du / dy}$$

where μ is known as the coefficient of viscosity, the dynamic viscosity or simply the viscosity of the fluid. "Absolute viscosity" shall be used to help differentiate it from another viscosity that will be discussed shortly.

In the beginning of this chapter, we noted that the distinction between a solid and a fluid lies in the manner in which each can resist shearing stress. A further distinction among various kinds of fluids and solids will be clarified by referring to Fig. 1.9. In the case of a solid, shear stress depends on the magnitude of the deformation, but Eq. (1.8) shows that in many fluids, shear stress is proportional to the time rate of deformation.

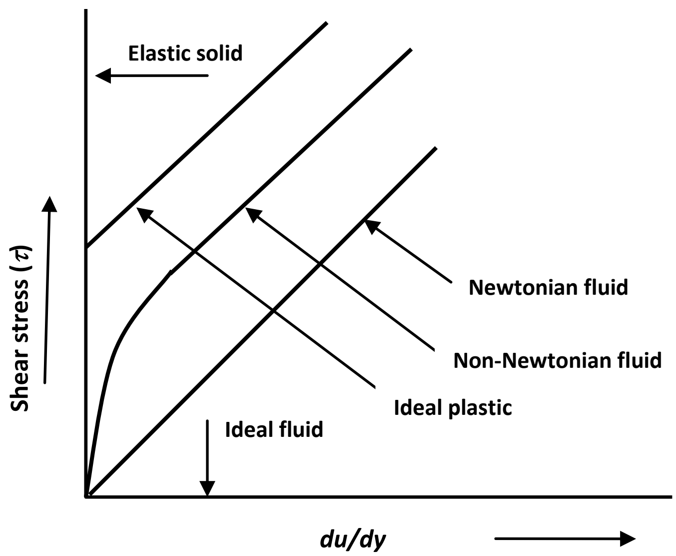


Figure 1.9 Velocity profiles – linear (bulk flow to right).

A fluid for which the constant of proportionality does not change with the rate of deformation is called a Newtonian fluid, and this is plotted as a straight line in Fig. 1.9. The slope of this line is the absolute viscosity μ. The ideal fluid with no viscosity falls on the horizontal axis, whereas a true elastic solid is plotted on the vertical axis. A plastic that sustains a certain amount of stress before suffering a plastic flow corresponds to a straight line intersecting the vertical axis at the yield stress. There are certain non-Newtonian fluids in which μ varies with the rate of deformation. These are relatively uncommon in engineering usage, so we restrict the remainder of this text to the common fluids that under normal conditions obey Newton's equation of viscosity. Note that

$$\text{Dimension of } \mu = \frac{\text{Dimension of } \tau}{\text{Dimension of } (du/dy)} = \frac{(\text{N}/\text{m}^2)}{\text{s}^{-1}} = \frac{\text{N s}}{\text{m}^2}$$

A widely used unit of viscosity in the metric system is the poise (P) named after Jean Louis Poiseuille (1799–1869). Poiseuille, a French anatomist, was one of the first investigators of viscosity. A poise = 0.10 Ns/m^2. A centipoise (=0.01 P) is frequently a more convenient unit. It has a further advantage in that the viscosity of water at 20.2°C is 1 cP. Thus, the value of viscosity in centipoise is an indication of the viscosity of a fluid relative to that of water at 20.2°C.

In many problems involving viscosity, the absolute viscosity is divided by density. This ratio defines the kinematic viscosity, ν <COMP: Greek letter nu>, so called because force is not involved, the only dimensions being length and time, as in kinematics. Thus,

$$\nu = \frac{\mu}{\rho} \tag{1.9}$$

Usually, the kinematic viscosity (ν) is measured in m^2/s. Previously, in the metric system, the common units were cm^2/s, also called the stoke (St), after Sir George Stokes (1819–1903), an English physicist and pioneering investigator of viscosity. Many physicists found centistoke (cSt) a more convenient unit to work with.

A comparison of kinematic viscosities of fluids is shown in Fig. 1.10(a) for low operating pressures. A more detailed characteristic of an ISO 32 mineral oil is shown in Fig. 1.10(b), illustrating the important effect of both temperature and pressure. The data shown in Fig. 1.10 make it clear that experimental testing must specify both pressure and temperature so that related studies may be compared with at least a minimum of confidence. It is common that computer dynamic simulations of hydraulic systems usually assume a mean temperature in a sense that the temperature does not vary significantly during the milliseconds to second of transient behavior. However, it may be necessary to model the effect of pressure on viscosity if large fluctuations in pressure are expected, although its effect may well be of secondary significance.

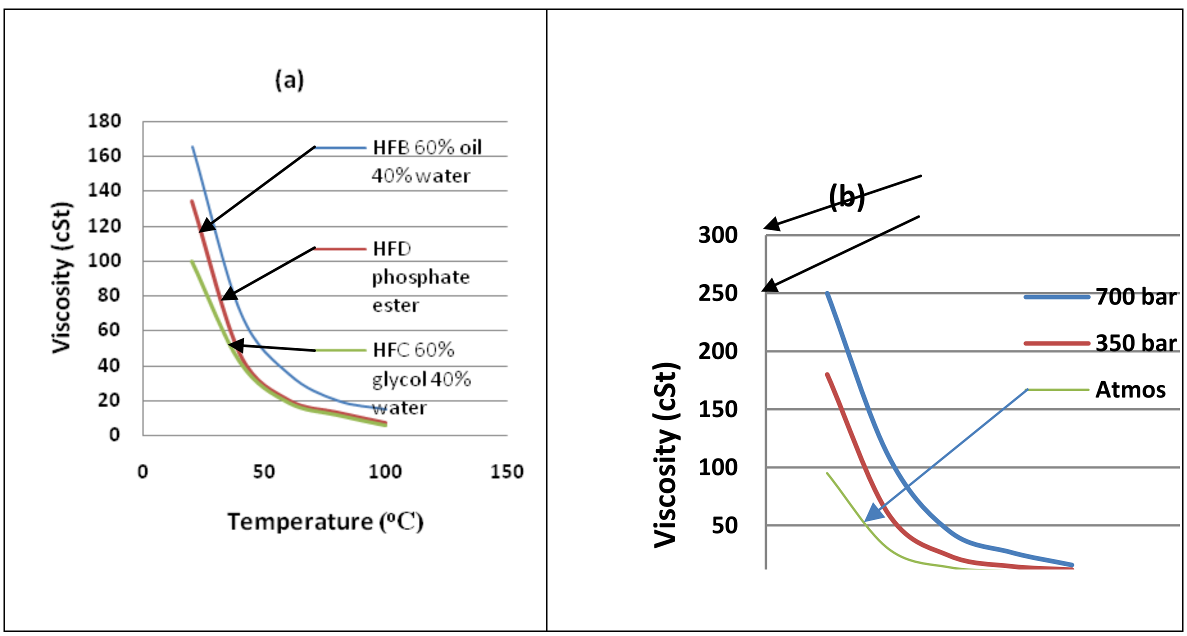


Figure 1.10 Typical kinematic viscosities for a range of fluids. (a) Some fire-resistant fluids. (b) An ISO 32 mineral oil.

Example 1.11

A 4.5 N force moves a piston inside a cylinder at a velocity of 3 m/s as shown in Fig. 1.11. The piston of 10.16 cm diameter is centrally located in the cylinder having an internal diameter of 10.17 cm. An oil film separates the piston from the cylinder. Find the absolute viscosity of the oil.

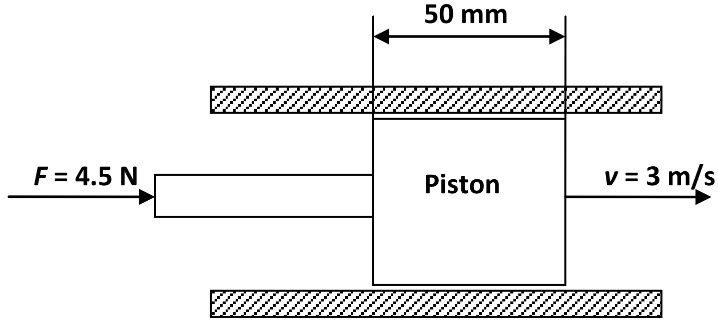


Figure 1.11

Solution: Given force on the piston = 4.5 N, piston diameter (d) =10.16 cm, velocity (v) = 3 m/s, cylinder diameter (D) = 10.17 cm and length (L) = 5 cm. Using Newton's law of viscosity we have

$$F = \mu \frac{du}{dy}$$

For small gaps,

$$F = \mu \frac{Av}{y} \tag{1.10}$$

Now we have

$$A = \pi DL = \pi \times 10.16 \times 5 = 159.6\,\text{cm}^2$$

$$v = 3\,\text{m/s} = 300\,\text{cm/s}$$

$$y = \frac{D-d}{2} = \frac{10.17 - 10.16}{2} = 0.005\,\text{cm}$$

Substituting in Eq. (1.10) we get

$$4.5 = \frac{\mu \times 159.6 \times 300}{0.005}$$

$$\Rightarrow \mu = \frac{4.7\times10^{-7}\,\text{N s}}{\text{cm}^2} = \frac{4.7\times10^{-7}\,\text{N s}}{10^{-4}\,\text{m}^2} = 4.7\times10^{-3}\,\text{N s / m}^2$$

Since 1 N = 10^5 dynes, we get

$$\mu = \frac{4.7\times10^{-7}\,\text{N s}}{\text{cm}^2} = \frac{4.7\times10^{-7}\times10^5\,\text{dynes s}}{\text{cm}^2}$$
$$= 0.047 \text{ poise} = 4.7 \text{ centipoise}$$

Now

Absolute viscosity (μ) in cP = Specific gravity (0.89) × Kinetic viscosity (ν) in cS

$\Rightarrow$ ν = 4.7/0.89 = 5.28 cS

Example 1.12

In Fig. 1.12, oil of the absolute viscosity μ fills the small gap of thickness Y.

(a) Neglecting fluid stress exerted on the circular underside, obtain an expression for the torque T required to rotate the cone at a constant speed ω.

(b) What is the rate of heat generation, in J/s, if the absolute viscosity of oil is 0.20 Ns/m^2, α = 45^o, a = 45 mm, b = 60 mm, Y = 0.2 mm and the speed of rotation is 90 rpm?

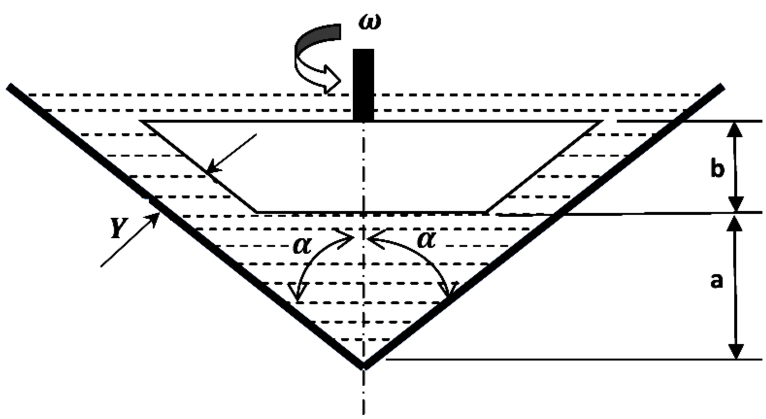


Figure 1.12

Solution: We have

(a) We have $U = \omega r$. For small gap Y we have

$$\frac{du}{dy}=\frac{U}{Y}=\frac{\omega r}{Y}$$

$$\tau=\frac{\mu\, du}{dy}=\frac{\mu\omega r}{Y}$$

$$dA=2\pi r ds=\frac{2\pi r dy}{\cos\alpha}$$

$$dF=\tau dA=-\frac{\mu\omega r\,(2\pi r dy)}{Y\,(\cos\alpha)}$$

$$dT=r\,dF=\frac{2\pi\mu\omega}{Y\,\cos\alpha}r^3 dy \quad (\text{where } r=\tan\alpha)$$

$$\Rightarrow dT=\frac{2\pi\mu\omega\tan^3\alpha}{Y\,\cos\alpha}\,y^3\,dy$$

Integrating we get

$$T=\frac{2\pi\mu\omega\,\tan^3\alpha}{Y\,\cos\alpha}\int_a^{a+b}y^3 dy$$

$$\Rightarrow T=\frac{2\pi\mu\omega\,\tan^3\alpha}{4Y\,\cos\alpha}[(a+b)^4-a^4]$$

(b) We have

$$[(a+b)^4-a^4]=(0.105\text{ m})^4-(0.045\text{ m})^4=0.0001175\text{ m}^4$$

ω = (90 rev/min)(2π radians/rev)(1min/60s) = 3π rad/s

Heat generation rate = Power =$T\omega$

$$P=\frac{2\pi\mu\,\omega^2\,\tan^3\alpha}{4Y\,\cos\alpha}[(a+b)^4-a^4]$$
$$=\frac{2\pi(0.20\text{ Ns/m}^2)(3\pi\text{ s}^{-1})^2(1)^3(0.0001175\;m^4)}{4(2\times10^{-4}\text{m})\cos 45°}$$

= 23.2 Nm/s = 23 .2 J/s

1.11 Viscosity Index

The viscosity of hydraulic oils decreases with increase in temperature. Hence, the viscosity of the given oil must be represented at a special temperature. The variation of viscosity with respect to temperature is different for different oils.

The viscosity index (VI) is a relative measure of the change in the viscosity of an oil with respect to a change in temperature. An oil having a low VI is one that exhibits a large change in viscosity with a small change in temperature. A high VI oil does not change appreciably with a change in temperature.

The VI of any hydraulic oil can be calculated as follows:

$$\mathrm{VI} = \frac{L-U}{L-H} \times 100 \tag{1.11}$$

where L is the viscosity in SUS (Saybolt universal viscosity)of a 0 VI oil at 100°F, U is the viscosity in SUS of an unknown VI oil at 100°F and H is the viscosity in SUS of a 100 VI oil at 100 °F. The VI of an unknown oil is determined from tests. A reference oil of 0 VI and a reference oil of 100 VI are selected. The viscosities of three oils (L, U and H) are then measured at 100°F.This is shown schematically in Fig. 1.13.

A high VI oil is a good all-weather-type oil for use in outdoor machines operating in extreme temperature swings. For a hydraulic system, the oil temperature does not change appreciably; hence, the VI of the oil is not crucial.

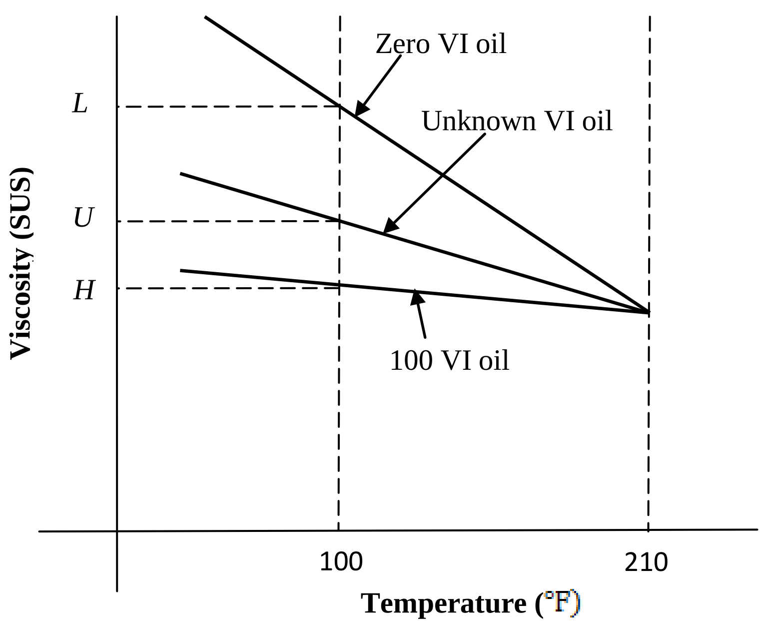


Figure 1.13 Viscosity index.

Example 1.13

A sample of oil with viscosity index of 70 is tested with a 0 VI oil and a 100 VI oil whose viscosity values at 100°F are 375 and 125 SUS, respectively. What is the viscosity of the sample oil at 100°F in units of SUS?

Solution: We know that

$$\mathrm{VI} = \frac{L - U}{L - H} \times 100$$

where L is the viscosity of 0 VI oil at 100°F, H is the viscosity of 100 VI oil at 100°F, and U is the required viscosity in SUS So,

$$70 = \frac{375 - U}{375 - 125} \times 100$$

$$\Rightarrow U = 200\,\mathrm{SUS}$$

Example 1.14

A sample oil is tested with a 0 VI oil and a 100 VI oil whose viscosity values at 38°C are 400 and 150 SUS, respectively. If the viscosity of the sample oil at 38°C is 200 SSU, what is the viscosity index of the sample oil?

Solution: We know that

$$\text{VI of the sample oil} = \frac{L - U}{L - H} \times 100 = \frac{400 - 200}{400 - 150} \times 100$$

$$= \frac{200}{250} \times 100 = 80\ \mathrm{VI}$$

Objective Type Questions

Fill in the Blanks

1. The molecules of a solid are usually ______ to each other than those of a fluid.

2. The molecules of a gas are much ______ than those of a liquid.

3. Absolute pressures are measured relative to ______ such as that existing in outer space.

4. An ideal fluid is usually defined as a fluid in which there is ______.

5. The viscosity of a fluid is a measure of its resistance to ______.

State True or False

1. Gauge pressures are measured relative to atmosphere.

2. The bulk modulus is analogous to the modulus of elasticity for solids.

3. At any given temperature, the bulk modulus of water does vary a great deal for a moderate range in pressure.

4.Hydraulic power is especially useful when performing a heavy work.

5. The viscosity of hydraulic oils increases with increase in temperature.

6. Mineral oil has a high bulk modulus compared to phosphate esters and water glycol fluids.

Review Questions

1. What are the differences between a liquid and a gas?

2. Define the terms specific density, specific weight and specific gravity.

3. Differentiate between absolute and gauge pressures.

4. What is meant by bulk modulus? Give its typical value for liquid and gas.

5. Differentiate between viscosity and viscosity index.

6. Define viscosity index and also suggest an optimum range.

7. Name two undesirable results when using an oil with a viscosity that is too high.

8. Name two undesirable results when using an oil with a viscosity that is too low.

9.What is meant by the term pressure head?

10.Under what conditions is viscosity index important?

11.It is desired to select an oil of a viscosity index suitable for a hydraulic powered front-end loader what will operate a year round. What viscosity index characteristics should be specified? Why?

Answers

Fill in the Blanks

1. Closer

2. Farther apart

3. Perfect vacuum

4. No friction

5. Shear or angular deformation

State True or False

1. True

2. True

3. True

4. True

5. False

6.False

Lecture 3

FLUIDS FOR HYDRAULIC SYSTEMS

Learning Objectives

Upon completion of this chapter, the student should be able to:

- ☐ List the various functions of hydraulic fluid.
- ☐ Explain the desirable properties of fluid.
- ☐ Differentiate between thick and boundary lubrication.
- ☐ Define neutralization number and demulsibility.
- ☐ Define fire point and flash point.
- ☐ Explain the various hydraulic fluids.
- ☐ Explain the various fire-resistant fluids.

1.1 Introduction

Although mineral oils were readily available in the beginning of 20th century, they were not practically used in hydraulic system until the 1920. In 1940s, additives were first used to improve the physical and chemical properties of hydraulic mineral oils. The first additives were developed to counter rust and oxidation. However, mineral oils are highly flammable, and fire risk increases when operating at high temperatures. This led to the development of fire-resistant fluids that are mainly water based, with limitations on the operating conditions. The need for extremes of operating temperatures and pressures led to the development of synthetic fluids. Personnel who operate, service, or design fluid power should have knowledge of the individual characteristics of hydraulic fluid and this chapter deals with functions of hydraulic fluid and their effect on system's performance.

1.2 Functions of Hydraulic Fluids

A hydraulic fluid is the transmitting medium of a hydraulic system. It is, therefore, an essential part of the system and we must know enough about it to ensure that the hydraulic system works efficiently. The most common liquid that is used as a medium in fluid power systems is petroleum-based mineral oil.
In fluid power systems, a hydraulic fluid has to perform various functions such as the following:

1. **Power transmission:** To transmit power, which is the primary function.
2. **Lubrication:** To lubricate various parts, so as to avoid metal-to-metal contact and reduce friction, wear and heat generation.
3. **Sealing:** To seal the moving elements to avoid leakage.
4. **Cooling:** To carry away the heat generated in the system and to dissipate the heat through a reservoir or a heat exchanger.
5. **Contaminant removal:** To carry along the contaminations to the tank, where they can be removed through filters.

For a fluid to perform efficiently, it must possess certain properties. The various properties required for an ideal hydraulic fluid are as follows:

1. Ideal viscosity.
2. Good lubrication capability.
3. Demulsibility.
4. Good chemical and environmental stability.
5. Incompressibility.
6. Fire resistance.
7. Low flammability.
8. Foam resistance.
9. Low volatility.
10. Good heat dissipation.
11. Low density.
12. System compatibility.

It is almost impossible to achieve all these properties in a hydraulic fluid. Although we can select a good fluid with desirable properties, some of the characteristics of a fluid change with usage. For example, it is common for the temperature of a fluid to rise due to friction in the system, which reduces the viscosity of the fluid, which in turn increases leakage and reduces lubrication ability. A fluid gets oxidized and becomes acidic with usage. Certain additives are added to preserve the desirable properties and to make the fluid more stable. Some of the desirable properties and their influence on a hydraulic fluid are discussed briefly in the following sub-sections.

1.2.1 Ideal Viscosity

The most basic desirable property of a hydraulic fluid is optimum viscosity. It is a measure of a fluid's resistance to flow. When viscosity is low, the fluid flows easily. On the other hand, when viscosity is high, the fluid flows with difficulty. A low viscous fluid is thin and can flow easily, whereas a high viscous fluid is thick and cannot flow easily. The viscosity of a fluid should be high enough to seal the working gap between the parts and prevent leakage but should be low enough to cause easy flow throughout the system. A high-viscosity fluid requires high energy to overcome the internal friction, resulting in excess heat generation. On the other hand, a low-viscosity fluid flows easily but causes leakages and reduces the volumetric and overall efficiency. Therefore the hydraulic fluid should have an optimum viscosity.

1. **High viscosity:**
   - High resistance to flow.
   - Increased power consumption due to frictional loss.
   - High temperature caused by friction.
   - Increased pressure drop because of the resistance.
   - Possibility of sluggish or slow operation.
   - Difficulty in separating air from oil in a reservoir.
   - Greater vacuum at the pump inlet, causing cavitation.
   - Higher system noise level.

2. **Low viscosity:**
    - Increased internal leakage.
    - Excessive water.
    - Possibility of decreased pump efficiency, causing slower operation of the actuator.
    - Increased temperature resulting from leakage losses.

There are two basic methods of specifying the viscosity of fluids: absolute and kinematic viscosity. Viscosity index is an arbitrary measure of a fluid resistance to viscosity change with temperature changes. Thus, viscosity is affected by temperature changes. As temperature increases, the viscosity of a fluid decreases. A fluid that has a relatively stable viscosity at temperature extremes has a high viscosity index. A fluid that is very thick while cold and very thin while hot has a low viscosity index.

1.2.1.1 *Specification of Oil as Per ISO*

Standardization of hydraulic oils has been done by the International Organization for Standardization. Table 1.1 lists ISO VG for engine oils and Table 1.2 lists ISO VG for industrial oil. Indian Oil Corporation markets oil as per ISO designation.

Table 1.1 ISO VG for engine oils

ISO Viscosity Grade	**Kinematic Viscosity** (cS @ 40 □C)	
	Minimum	**Maximum**
ISO VG 2	1.98	2.42
ISO VG 3	2.88	3.52
ISO VG 5	4.14	5.06
ISO VG 7	6.12	7.48
ISO VG 10	9	11.0
ISO VG 15	13.5	16.5
ISO VG 22	19.8	24.2
ISO VG 32	28.8	35.2
ISO VG 46	41.4	50.6
ISO VG 68	61.2	74.8
ISO VG 100	90	110
ISO VG 150	135	165
ISO VG 220	198	242
ISO VG 320	288	352
ISO VG 460	414	506
ISO VG 680	612	748
ISO VG 1000	900	1100
ISO VG 1500	1350	1650

Table 1.2 ISO VG for industrial oil

Old Designation	**New Designation**
Servo system 311	Servo system 32
Servo system 314	Servo system 46
Servo system 317	Servo system 57[a]
	Servo system 68
Servo system 321	Servo system 81[a]
	Servo system 100
Servo system 526	Servo system 121
	Servo system 150
Servo system 533	Servo system 176
	Servo system 220
Servo system 563	Servo system 320
	Servo system 460
Servo system 711	Servo system A 32
Servo system 733	Servo system A 176[a]
Servocirol 11	Servocirol 32
Servocirol 14	Servocirol 46
Servocirol 17	Servocirol 57[a]
	Servocirol 68
Servocirol 21	Servocirol 81[a]
	Servocirol 100
Servocirol 26	Servocirol 121[a]
	Servocirol 100
Servohydex 14	Servohydex 32
Servohydex 21	Servohydex 57[a]

[a]Non-standard ISO VG.

1.2.2Lubrication Capability

Hydraulic fluids must have *good* lubricity to prevent friction and wear between the closely fitted working parts such as vanes of pumps, valve spools, piston rings and bearings. Wear is the removal of surface material due to the frictional force between two metal-to-metal contact of surfaces. This can result in a change in dimensional tolerances, which can lead to improper functioning and failure of the components. Hydraulic oil should have a good lubricating property. That is, the film so formed should be strong enough that it is not wiped out by the moving parts.

There are two main kinds of lubrication mechanisms: thick film and boundary film. In low-pressure hydraulic systems such as hand-operated pumps and cylinders, a fluid providing thick-film lubrication is sufficient. A thick film is about 10 times the surface roughness. Under such conditions, there is no metal-to-metal contact as shown in Fig. 1.1, and therefore there is no wear (the coefficient of friction is as low as 0.01–0.02).

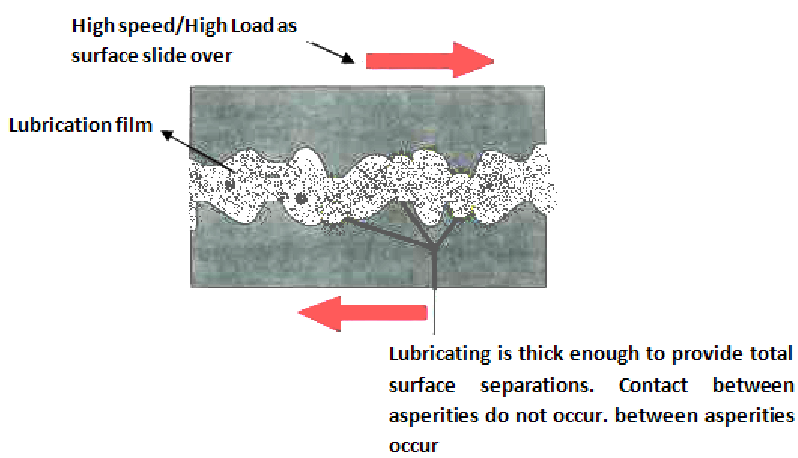


Figure 1.2 Thick film lubrication

However as the speed of the moving part increases like in high speed motor, actuators, valves the film thickness reduces to about three to five times the surface roughness. This increases metal-to-metal contact as shown in Fig. 1.2 and also increases the coefficient friction and wear rate. In such situations, additives are added to the fluid to increase the load- carrying capacity of the film. In this case, the coefficient of friction is quite high and the wear rates are higher than the thick-film lubrication condition.

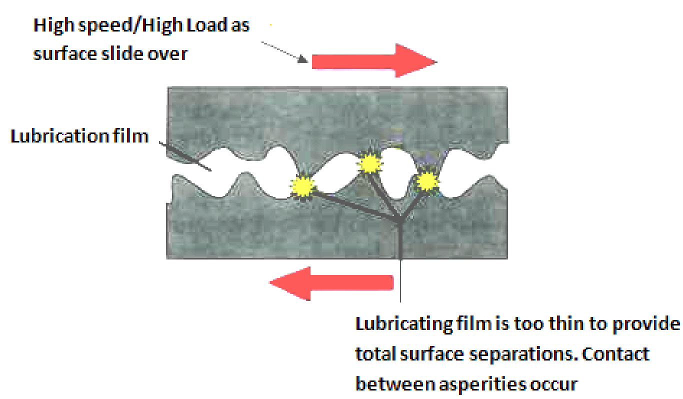


Figure 1.2 Boundary film lubrication

The friction force (F) is the force parallel to the two mating surfaces that slide relative to each other. This friction force actually opposes the sliding movement between the two surfaces. The greater the frictional force, the greater the wear and heat generated. This, in turn, results in power losses and reduced life, which, in turn, increases maintenance cost.

It has been determined that the frictional force (F) is proportional to the normal force N that forces the two surfaces together. The proportionality constant is called the coefficient of friction (μ)

$$F = \mu N \qquad (1.1)$$

Thus, the greater the values of the coefficient of friction and normal force, the greater the frictional force and hence wear. The magnitude of the normal force depends upon the amount of power and forces being transmitted and thus is independent of hydraulic fluid properties. However, the coefficient of friction depends on the ability of the fluid to prevent metal–metal contact of the closely fitting mating parts.

Equation (1.1) can be rewritten to solve for the coefficient of friction, which is a dimensionless parameter:

$$\mu = \frac{F}{N} \qquad (1.2)$$

It can be seen now that μcan be experimentally determined to give an indication of the anti-wear properties of a fluid F and N can be measured.

1.2.3 Demulsibility

The ability of a hydraulic fluid to separate rapidly from moisture and successfully resist emulsification is known as "demulsibility." If an oil emulsifies with water, the emulsion promotes the destruction of lubricating and sealant properties. Highly refined oils are basically water resistant by nature.

1.2.4 Good Chemical and Environmental Stability (Oxidation and Corrosion Resistance)

For a good hydraulic fluid, a good chemical and environmental stability is desirable. Most fluids are vulnerable to oxidation, as they come in contact with oxygen in air. Mineral oils or petroleum-based oils (widely used in hydraulic systems) contain carbon and hydrogen molecules, which easily react with oxygen. The oxidation products are highly soluble in oil and being acidic in nature they can easily corrode metallic parts. The soluble acidic products cause corrosion, whereas insoluble products make the operation sluggish. Oxidation leads to deterioration in the chemical nature of fluid, which may form some chemical sledges, gum or varnish at low velocity or stagnation points in the system.

Many factors influence the rate of oxidation, such as temperature, pressure, moisture and so on. Temperature is the most affecting one, as the rate of oxidation increases severely with rise in temperature.

The moisture entering the hydraulic system with air causes the parts made of ferrous materials to rust. Rust is a chemical reaction between iron or steel and oxygen. Corrosion, on the other hand, is the chemical reaction between a metal and an acid. The result of rusting and corrosion is the "eating away" of the metal surfaces of the hydraulic components. Rust and corrosion cause excessive leakage between moving parts. Rust and corrosion can be prevented by incorporating additives that plate on the metal surface to prevent chemical reaction.

1.2.5 Neutralization Numbers

Neutralization number is a measure of the acidity or alkalinity of hydraulic oil. This is referred to as the pH value of the oil. High acidity causes the oxidation rate in oil to increase rapidly.

1.2.6 Incompressibility

Though we consider hydraulic fluids as incompressible, in practice, they are relatively compressible. Most mineral oils undergo reduction in the volume of about 0.7% for every 100 bar rise in pressure. In fact, the compressibility of a fluid is greatly influenced by temperature and pressure.

The incompressibility of a fluid is a measure of its stiffness and is given by its bulk modulus. The bulk modulus (B) of a fluid is the ratio of volumetric stress to volumetric strain and is given by the relation

$$B = -\frac{\Delta P}{\Delta V / V}$$

where B is the bulk modulus (Pa), ΔP is the change in pressure (Pa), ΔV is the change in volume (m^3) and V is the original volume.The compressibility of a fluid has an influence on the system response and makes it susceptible to shock waves. In normal hydraulic systems, its effect on system response is not considered, whereas decompression valves are used to avoid shock wave problems.

1.2.7 FireResistance

There are many hazardous applications where human safety requires the use of a fire-resistant fluid. Examples include coal mines, hot metal processing equipment, aircraft and marine fluid power systems. A fire-resisting fluid is one that can be ignited but does not support combustion when the ignition source is removed. Flammability is defined as the ease of ignition and ability to propagate the flame. The following are the usual characteristics tested in order to determine the flammability of hydraulic fluids:

1. **Flash point:** The temperature at which an oil surface gives off sufficient vapors to ignite when a flame is passed over the surface.
2. **Fire point:** The temperature at which an oil releases sufficient vapors to support combustion continuously for 5 s when a flame is passed over the surface.
3. **Autogenously ignition temperature:** The temperature at which ignition occurs spontaneously.

The fire-resistant fluids are designated as follows:

1. **HFA:** A high-water-content fluid or HWCF (80% or more), for example, water–oil emulsions.
2. **HFB:** This is water–oil emulsion containing petroleum oil and water.
3. **HFC:** This is a solution of water and glycol.
4. **HFD:** This is a synthetic fluid, for example, phosphates or phosphate–petroleum blends.

The commonly used hydraulic liquids are petroleum derivatives; consequently, they burn vigorously once they reach a fire point. For critical applications, artificial or synthetic hydraulic fluids are used that have fire resistance.

Fire-resistant fluids have been developed to reduce fire hazards. There are basically four different types of fire-resistant hydraulic fluids in common use:

1. **Water–glycol solution:** This type consists of an actual solution of 40% water and 60% glycol. These solutions have high-viscosity-index values, but as the viscosity rises, the water evaporates. The operating temperature ranges run from −20°C to about 85°C.
2. **Water-in-oil emulsions:** This type consists of about 40% water completely dispersed in a special oil base. It is characterized by small droplets of water completely surrounded by oil. The operating temperature range runs from −30°C to about 80°C. As is the case with water–glycol solutions, it is necessary to replenish evaporated water to maintain proper viscosity.
3. **Straight synthetics**: This type is chemically formulated to inhibit combustion and in general has the highest fire-resistant temperature. The disadvantages of straight synthetics include low viscosity index, incompatibility with most natural or synthetic rubber seals and high cost.
4. **High-water-content fluids (HWCFs):** This type consists of about 90% water and 10% concentrate (designated as 90/10). The concentrate consists of fluid additives that improve viscosity, lubrication, rust protection and protection against bacteria growth.The maximum operating temperature should be held to 50°C to minimize evaporation.

The advantages of HWCF are as follows:

1. Fire resistance due to a high flash point of about 150°C.
2. Lower system operating temperature due to good heat dissipation.
3. Biodegradable and environmental-friendly additives.
4. High viscosity index.
5. Cleaner operation of the system.
6. Low cost of concentrate and storage.

The disadvantages of HWCF are as follows:

1. Greater contamination due to higher densities of fluids.
2. High evaporation loss.
3. Faster corrosion due to oxidation.
4. pH value to be maintained between 7.5 and 9.0.
5. Promotion of bacterial growth and filtration difficulties due to the acidic nature of fluids.

Performance of HWCFs can be improved using additives such as

1. Anti-wear additives.
2. Anti-foaming additives.
3. Corrosion inhibitors.
4. Biocides to kill water-borne bacteria.
5. Emulsifying agents.
6. Flocculation promoters.
7. Deionization agents.
8. Oxidation inhibitors.
9. Anti-vaporizing agents.

1.2.8Low Flammability

A fire-resistant fluid is one that can get ignited in the presence of an ignition source but does not support combustion when the source is removed. This characteristic is defined as flammability. It refers to the ease with which a fluid gets ignited and propagates the flame. Hence, it is desirable to have a low flammability for a hydraulic fluid.

1.2.9 Foam Resistance
Air can be present in a hydraulic fluid in two forms: dissolved and entrained. For example, if the return line to the reservoir is not submerged, the jet of oil entering the liquid surface will carry air with it. This causes air bubbles to form in the oil. If these bubbles rise to the surface too slowly, they will be drawn into the pump intake. This can cause pump damage due to cavitation. Another adverse effect of entrained and dissolve air is a great reduction in the bulk modulus of the hydraulic fluid.

1.2.10 Low Volatility
A fluid should possess low vapor pressure or high boiling point. The vapor pressure of a fluid varies with temperature and hence the operating temperature range of the system is important in determining the stability of the fluid.

1.2.11 Good Heat Dissipation
A hydraulic fluid should have a high heat dissipation capability. The temperature of a fluid shoots up if its heat dissipation characteristics are poor. Too high fluid temperature can cause a system to malfunction. If the fluid overheats, it may cause the following:

1. Give off vapor and cause cavitation of the pump.
2. Increase the rate of oxidation causing its rapid deterioration by producing sledges, varnishes, etc., thus shortening its useful life.
3. Reduce viscosity of the fluid resulting in increased leakage, both internal and external.
4. Cause thermal distortion in components.
5. Damage seals and packaging owing to embrittlement.

Hydraulic systems should be designed so that a heat balance occurs at a satisfactory operating temperature.

1.2.12 Low Density
The relative density of a mineral oil is 0.9 (the exact value depends on the base oil and the additive used). Synthetic fluids can have a relative density greater than 1. The relative density is important when designing the layout of pumps and reservoir.

1.2.13 System Compatibility
A hydraulic fluid should be inert to materials used in or near the hydraulicequipment. If the fluid in anyway attacks, destroys, dissolves or changes the parts of hydraulic system, the system may lose its functional efficiency and may start malfunction.

1.3 Additives in Hydraulic Fluids

Some of the commonly used additives and their purposes are as follows:

1. Pour point depressant: A pour point is the temperature at which a fluid ceases to flow. The minimum operating temperature in a hydraulic system should be at least 10°C above the pour point. Pour point depressants inhibit the formation of wax crystals in the mineral oils and hence enhance the pour points.There is a range of pour point depressant additives of different chemical species, important ones are polymethacrylates, polyacrylates and alkalated naphthalene.

2. Viscosity index improvers: These additives are long-chain polymers that stay in a coiled form in the hydraulic fluid. At a low operating temperature, they have no effect on viscosity. But when the temperature rises, these polymers uncoil and intermesh causing a thickness effect in the fluid, thereby not allowing the viscosity to drop down.

3. Defoamers(anti-foam additives): Certain additives, such as silicon polymer, act as defoamers. They cause a rapid breakdown of the foam by removing the entrained air bubbles. Foaming occurs in oil as a surface phenomenon. Bubbles of air are encircled by an oil film and cannot escape. These bubbles under pressure become very hot and can be the cause of system overheating.Foam usually forms in the reservoir and, if drawn into the pump suction, can cause noisy pump operation and may even damage pump parts. Control response is spongy and unreliable.Although all fluids are susceptible to foaming, the amount of foam in a system can be reduced to a minimum by the addition of chemical depressants.

4. Oxidation inhibitors: Oxidation causes the chemical reaction and formation of acidic products that leads to corrosion problems. The oxidation rate increases with temperature. Certain additives having greater affinity for oxygen are added so that they easily react with them than with oil.

5. Corrosion inhibitors: These additives form a thin film on the metal surface and shield it from coming in direct contact with the chemicals/acids in the fluid, thereby preventing corrosion problems.

6. Anti-wear additives: These are either long-chain polymer or extreme pressure (EP) additives. The long-chain polymers are adsorbed on the metal surfaces, causing a high local temperature and polish the surface. This helps in reducing the surface roughness, hence the wear problem.

7. Load-carrying capacity: The load-carrying capacity of a hydraulic fluid is a measure of the oil's capability to maintain a film of lubricant between two metal surfaces under extremes of load or pressure. All hydraulic oils have a natural load-carrying capacity that can be enhanced by special additives known as EP additives. These additives help reduce wear especially in hydraulic pumps and motors by providing lubrication when almost all the oil film has been squeezed out under heavy load conditions.

1.4 Types of Hydraulic Fluids

There are different types of hydraulic fluids that have the required properties. In general, while selecting a suitable oil, a few important factors are considered. First, its compatibility with seals, bearing and components is seen; second, its viscosity and other parameters such as fix resistance and environmental stability are also considered. There are five major types of hydraulic flow fluids which meet various needs of the system. These are briefly discussed as follows:

1. Petroleum-based fluids: Mineral oils are the petroleum-based oils that are the most commonly used hydraulic fluids. Basically, they possess most of the desirable characteristics: they are easily available and are economical. In addition, they offer the best lubrication

ability, least corrosion problems and are compatible with most seal materials. The only major disadvantage of these fluids is their flammability. They pose fire hazards, mainly from the leakages, in high-temperature environments such as steel industries, etc.
Mineral oils are good for operating temperatures below 50°C, At higher temperatures, these oils lose their chemical stability and form acids, varnishes, etc. All these lead to the loss of lubrication characteristics, increased wear and tear, corrosion and related problems. Fortunately, additives are available that improve chemical stability, reduce oxidation, foam formation and other problems.

A petroleum oil is still by far the most highly used base for hydraulic fluids. In general, petroleum oil has the following properties:

- ☐ Excellent lubricity.
- ☐ Higher demulsibility.
- ☐ More oxidation resistance.
- ☐ Higher viscosity index.
- ☐ Protection against rust.
- ☐ Good sealing characteristics.
- ☐ Easy dissipation of heat.
- ☐ Easy cleaning by filtration.
- ☐ Most of the desirable properties of the fluid, if not already present in the crude oil, can be incorporated through refining or adding additives.

A principal disadvantage of petroleum oil is that it burns easily. For applications where fire could be a hazard, such as heat treating, hydroelectric welding, die casting, forging and many others, there are several types of fire-resistant fluids available.

2. **Emulsions:** Emulsions are a mixture of two fluids that do not chemically react with others. Emulsions of petroleum-based oil and water are commonly used. An emulsifier is normally added to the emulsion, which keeps liquid as small droplets and remains suspended in the other liquid. Two types of emulsions are in use:

- ☐ **Oil-in-water emulsions:** This emulsion has water as the main phase, while small droplets of oil are dispersed in it. Generally, the oil dilution is limited, about 5%; hence, it exhibits the characteristics of water. Its limitations are poor viscosity, leading to leakage problems, loss in volumetric efficiency and poor lubrication properties. These problems can be overcome to a greater extent by using certain additives. Such emulsions are used in high-displacement, low-speed pumps (such as in mining applications).
- ☐ **Water-in-oil emulsions:** Water-in-oil emulsions, also called inverse emulsions, are basically oil based in which small droplets of water are dispersed throughout the oil phase. They are most popular fire-resistant hydraulic fluids. They exhibit more of an oil-like characteristic; hence, they have good viscosity and lubrication properties. The commonly used emulsion has a dilution of 60% oil and 40% water. These emulsions are good for operations at 25°C, as at a higher temperature, water evaporates and leads to the loss of fire-resistant properties.

3. Water glycol: Water glycol is another nonflammable fluid commonly used in aircraft hydraulic systems. It generally has a low lubrication ability as compared to mineral oils and is not suitable for high-temperature applications. It has water and glycol in the ratio of 1:1. Because of its aqueous nature and presence of air, it is prone to oxidation and related problems. It needs to be added with oxidation inhibitors. Enough care is essential in using this fluid as it is toxic and corrosive toward certain metals such as zinc, magnesium and aluminum. Again, it is not suitable for high-temperature operations as the water may evaporate. However, it is very good for low-temperature applications as it possesses high antifreeze characteristics.

4. Synthetic fluids: Synthetic fluid, based on phosphate ester, is another popular fire-resistant fluid. It is suitable for high-temperature applications, since it exhibits good viscosity and lubrication characteristics. It is not suitable for low-temperature applications. It is not compatible with common sealing materials such as nitrile. Basically being expensive, it requires expensive sealing materials (viton). In addition, phosphate ester is not an environmental-friendly fluid. It also attacks aluminum and paints.

5. Vegetable oils: The increase in the global pollution has led to the use of more environmental-friendly fluids. Vegetable-based oils are biodegradable and are environmental safe. They have good lubrication properties, moderate viscosity and are less expensive. They can be formulated to have good fire resistance characteristics with certain additives. Vegetable oils have a tendency to easily oxidize and absorb moisture. The acidity, sludge formation and corrosion problems are more severe in vegetable oils than in mineral oils. Hence, vegetable oils need good inhibitors to minimize oxidation problems.

6. Biodegradable hydraulic fluids: As more and more organizations are understanding their social responsibility and are turning toward eco-friendly machinery and work regime, a biodegradable hydraulic fluid is too becoming a sought after product in the dawn of an environmentalist era. Biodegradable hydraulic fluids, alternatively known as bio-based hydraulic fluids, Bio-based hydraulic fluids use sunflower, rapeseed, soybean, etc., as the base oil and hence cause less pollution in the case of oil leaks or hydraulic hose failures. These fluids carry similar properties as that of a mineral oil–based anti-wear hydraulic fluid,

Hypothetically, if a company plans to introduce bio-based fluids into the hydraulic components of the machinery and the permissible operating pressure of hydraulic components is reduced to 80%, then it would inversely lead to a 20% reduction in breaking-out force owing to the 20% reduction in excavator's operating pressure. It is so because a reduction in the operating pressure of a system leads to a reduction in actuator force.

Besides, the transformation would not only include the cost of fluid and flushing of machinery to transcend from a mineral oil to vegetable oil repeatedly but also include the derating costs of machinery.

1.5 Factors Influencing the Selection of a Fluid

The selection of a hydraulic fluid for a given system is governed by the following factors:

1. Operating pressure of the system.
2. Operating temperature of the system and its variation.
3. Material of the system and its compatibility with oil used.
4. Speed of operation.
5. Availability of replacement fluid.
6. Cost of transmission lines.
7. Contamination possibilities.
8. Environmental condition (fire proneness, extreme atmosphere like in mining, etc.).
9. Lubricity.
10. Safety to operator.
11. Expected service life.

Objective-Type Questions

Fill in the Blanks

1. The ability of a hydraulic fluid to separate rapidly from moisture and successfully resist emulsification is known as _______.
2. The neutralization number is a measure of the _______of hydraulic oil.
3. Emulsions are a mixture of two fluids which _______chemically react with others.
4. Rust is a chemical reaction between iron or steel and _______.
5. The incompressibility of a fluid is a measure of its _______.

State True or False

1. Synthetic fluids can have a relative density greater than 1.
2. Viscosity index improvers are short–chain polymers.
3. Viscosity index is the arbitrary measure of a fluid resistance to viscosity change with temperature changes.
4. Mineral oils or petroleum-based oils easily react with oxygen.
5. Rust is a chemical reaction between a metal and an acid.

Review Questions

1. What is a fluid? What are the functions and characteristics of hydraulic fluids?

2. List 10 properties that a hydraulic oil should possess.

3. Discuss the role of additives used in fluid power systems.

4. Differentiate between absolute and kinematic viscosities. Also write their units in CGS and SI systems of units.

5. List some fire-resistant fluids used in hydraulic industry.

6. Why is water not used as a medium in fluid power systems?

7. What are the advantages of high-water-based fluids?

8. How are fire-resistant fluids designated?

9. What is the neutralization number of a hydraulic fluid?Give its importance.

10. What are the functions of a hydraulic fluid?

11. What are the types of liquids used in a hydraulic system?

12. List the types of additives used in a hydraulic fluid.

13. Give the specification of two typical mineral-based hydraulic oils as per ISO.

14. What are the main disadvantages of biodegradable oils?

Answers

Fill in the Blanks

1.Demulsibility
2. Acidity or alkalinity
3. Does not
4. Oxygen
5. Stiffness

State True or False

1. True
2. False
3. True
4. True
5. False

Lecture 4
GOVERNING PRINCIPLES AND LAWS

Learning Objectives

Upon completion of this chapter, the student should be able to:

- State Pascal's law.
- Write force power and force displacement relations.
- State practical applications of Pascal's law and evaluate the parameters.
- Explain the wworking of pressure booster and evaluate the parameters.
- Explain law of conservation of energy.
- Derive ccontinuity and Bernoulli's equation.
- Modify Bernoulli's equation to energy equation.
- State Torricelli's theorem and workout related problems.
- State siphon principleand workout problems.

1.1 Introduction

Fluid power systems are designed using all the principles learned in fluid mechanics. It is appropriate to briefly review these principles before proceeding with our study of the applications. One of underlying postulates of fluid mechanics is that, for a particular position within a fluid at rest, the pressure is the same in all directions. This follows directly from Pascal's Law. A second postulate states that fluids can support shear forces only when in motion. These two postulates define the characteristics of fluid media used to transmit power and control motion. This chapter deals with fundamental laws and equations which govern the fluid flow which is essential for the rational design of fluid power components and systems. Traditional concept such as continuity, Bernoulli's equation and Torricelli's theorem are presented after a brief review on mechanics.

1.2 Brief review of Mechanics

Fluid power deals with conversion Hydraulic power to mechanical power. Therefore it is essential to understand the concept of energy and power.

1.2.1 Energy

Energy is defined as the ability to perform work. If a force acts on a body and moves the body through a specified distance in the direction of its application, a work has been done on the body. The amount of this work equals the product of the force and distance where both the force and distance are measured in the same direction. Mathematically we can write

$$W_D = Fd$$

where F is the force (N), d is the distance (m) and W_D is the work done (J or Nm).In the SI system, a joule (J) is the work done when a force of 1 N acts through a distance of 1 m. Since work equals force times distance, we have

$$1\text{ J} = 1\text{ N} \times 1\text{ m} = 1\text{ Nm}$$

Thus, we have

$$\text{Energy (J)} = F\text{ (N)} \times d\text{ (m)}$$

The transfer of energy is an important consideration in the operation of fluid power systems. Energy from a prime mover is transferred to a pump via a rotating motor shaft and couplings. The pump converts this mechanical energy into hydraulic energy by increasing the fluid pressure. The pressurized fluid does work on hydraulic actuators. An actuator converts the hydraulic energy into mechanical energy and moves the external load. Not all the input mechanical energy is converted into useful work. There are frictional losses through valves, fittings and other system control components.

These losses show up as heat loss that is lost to the atmosphere with the increase in the fluid temperature.

1.2.2 Power

Itis defined as the rate of doing work. Thus, the power input to the hydraulic system is the rate at which an actuator delivers energy to the external load. Similarly, the rate at which an actuator delivers energy to the external load is equal to the power output of a hydraulic system. The power output is determined by the requirements of the external load.

A hydraulic system is used because of its versatility in transferring power. The versatility includes the advantages of variable speed, reversibility, overload protection, high power-to-weight ratio and immunity to damage under a stalled condition:

$$\text{Power} = P = \frac{Fd}{t}$$

or

$$P = Fv$$

whereF is the force (N), v is the velocity (m/s) and P is the power (N m/s or W). In the SI system, 1 watt (W) of power is the rate at which 1 J of work is done per second:

$$\text{Power} = \frac{\text{Work}}{\text{Time}}$$

In SI units we have

$$1\,\text{W} = \frac{1\,\text{Joule}}{\text{s}} = 1\,\text{N m / s}$$

Thus, we have

$$\text{Power (W)} = \frac{\text{Work (N m)}}{\text{Time(s)}}$$

Balancing the units, we can write

$$\text{Hydraulic power}\,(W) = \text{Pressure} \times \text{Flow}$$

$$= p\ (\text{N/m}^2) \times Q\ (\text{m}^3/\text{s})$$

$$= p \times Q\ (\text{N m/s}) = p \times Q\ (\text{W})$$

It is usual to express flow rate in liters/minute (LPM) and pressure in bars. To calculate hydraulic power using these units, a conversion has to be made. Thus,

$$Q\ (\text{L/min}) = Q/60\ (\text{L/s})$$

$$\Rightarrow Q\left(\frac{\text{L}}{\text{min}}\right) = \frac{Q}{60}\left(\frac{\text{L}}{\text{s}}\right) = \frac{Q}{60\times 10^3}\left(\frac{\text{m}^3}{\text{s}}\right)$$

$$\Rightarrow p(\text{bar}) = p \times 10^5\ \frac{\text{N}}{\text{m}^2}$$

Hydraulic power is

$$Q\left(\frac{\text{l}}{\text{min}}\right) \times p(\text{bar}) \times \frac{1\times 10^5}{60\times 10^3}\left(\frac{\text{m}^3}{\text{s}} \times \frac{\text{N}}{\text{m}^2}\right)$$

$$\Rightarrow Q \times p(\text{bar}) \times \frac{1\times 10^3}{600}\ (\text{W}) = \frac{Q(\text{LPM}) \times p(\text{bar})}{600}\ (\text{kW})$$

Thus, hydraulic power (kW) is

$$\frac{\text{Flow (LPM)} \times \text{Pressure (bar)}}{600}$$

In the SI metric system, all forms of power are expressed in watt. The pump head Hp in units of meters can be related to pump power in units of watt by using $p = \gamma h$. So

$$H_p = \frac{\text{Pump hydraulic power (W)}}{\gamma\left(\frac{\text{N}}{\text{m}^3}\right)\times Q\left(\frac{\text{m}^3}{\text{s}}\right)}$$

The above equation can also be used to find a motor head where H_p is replaced by H_m. The hydraulic power is replaced by the motor hydraulic power and Q represents the motor flow rate.

The mechanical power output (brake power or torque power) delivered by a hydraulic motor can be found by the following equation

$$\text{Power (kW)} = \frac{T\,(\text{N m})\times\omega\ (\text{rad/s})}{1000} = \frac{T\ (\text{N m})\times N\ (\text{rpm})}{9550}$$

Where$T$ is the torque and ω or N is the angular speed.

1.3 Pascal's Law

Pascal's law states that the pressure exerted on a confined fluid is transmitted undiminished in all directions and acts with equal force on equal areas and at right angles to the containing surfaces. In Fig. 1.1, a force is being applied to a piston, which in turn exerts a pressure on the confined fluid. The pressure is equal everywhere and acts at right angles to the containing surfaces. Pressure is defined as the force acting per unit area and is expressed as

$$\text{Pressure} = \frac{F}{A}$$

where F is the force acting on the piston, A is the area of the piston and p is the pressure on the fluid.

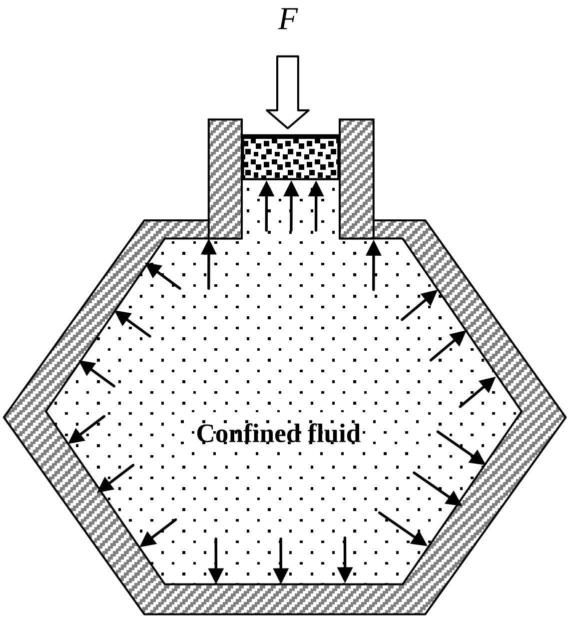


Figure 1.1 Illustration of Pascal's law

1.3.1 *Multiplication of Force*

The most useful feature of fluid power is the ease with which it is able to multiply force. This is accomplished by using an output piston that is larger than the input piston. Such a system is shown in Fig. 1.2.

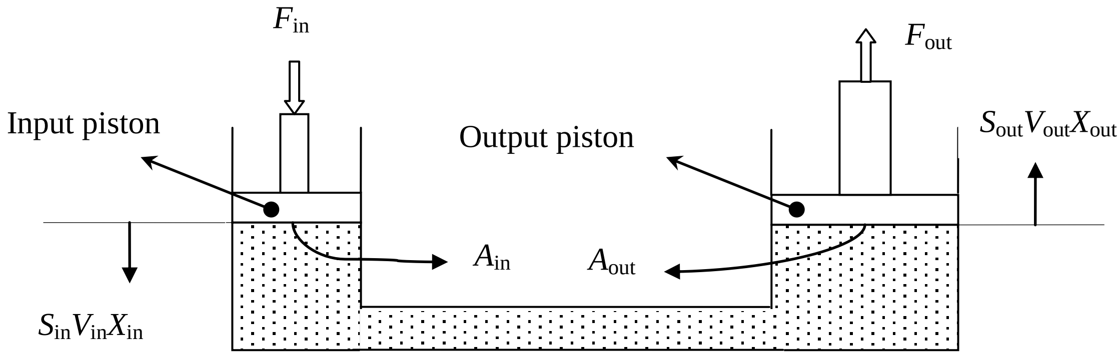


Figure 1.2 Multiplication of force

This system consists of an input cylinder on the left and an output cylinder on the right that is filled with oil. When the input force is F_{in} on the input piston, the pressure in the system is given by

$$P = \frac{F_{out}}{A_{out}}$$

$$\Rightarrow F_{out} = PA_{out} = \frac{F_{in}}{A_{in}} A_{out} = \frac{A_{out}}{A_{in}} F_{in}$$

Here to obtain the output force, the input force is multiplied by a factor that is equal to the ratio of the output piston area to the input piston area. If the output piston area is x times the input piston area, then the output force is x times the input force. Generally, the cross-sectional area of the piston is circular. The area is given by

$$A = \pi d^2 / 4$$

Hence, the above equation can be written as

$$F_{out} = \frac{d_{out}^2}{d_{in}^2} F_{in}$$

$$\Rightarrow \frac{F_{out}}{F_{in}} = \frac{d_{out}^2}{d_{in}^2}$$

The conservation of energy is very fundamental principle. Itstates that energy can neither be created nor destroyed. At first sight, multiplication of force as depicted in Fig.1.2 may give the impression that something small is turned into something big. But this is wrong, since the large piston on the right is only moved by the fluid displaced by the small piston on left. Therefore, what has been gained in force must be sacrificed in piston travel displacement. Now we shall mathematically derive force displacement relation and force power relation.

1. **Force displacement relation**: A hydraulic oil is assumed to be incompressible; hence, the volume displaced by the piston is equal to the volume displaced at the output piston:

$$V_{in} = V_{out}$$

Since the volume of a cylinder equals the product of its cross-sectional area and its height, we have

$$A_{in}S_{in} = A_{out}S_{out}$$

whereS_{in} is the downward displacement of the input piston and $S_{out}$is the upward displacement of the output piston:

$$\frac{S_{in}}{S_{out}} = \frac{A_{out}}{A_{in}}$$

Comparing

$$\frac{F_{out}}{F_{in}} = \frac{A_{out}}{A_{in}} = \frac{S_{in}}{S_{out}} \qquad (1.1)$$

2. **Force power relation:** A hydraulic oil is assumed to be incompressible; hence, the quantity of oil displaced by the input piston is equal to the quantity of oil gained and displaced at the output piston:
   Flow rate is the product of area and volume of fluid displaced in a specified time

$$\begin{aligned} Q_{in} &= Q_{out} \\ \Rightarrow A_{in}V_{in} &= A_{out}V_{out} \\ \Rightarrow \frac{A_{out}}{A_{in}} &= \frac{V_{in}}{V_{out}} \end{aligned} \qquad (1.2)$$

Comparing Equations. (1.1) and (1.2) we get

$$\frac{A_{out}}{A_{in}} = \frac{V_{in}}{V_{out}} = \frac{F_{out}}{F_{in}} = \frac{S_{in}}{S_{out}}$$

From the above equation, we get

$$F_{in}S_{in} = F_{out}S_{out}$$

or (Work done)$_{in}$ = (Work done)$_{out}$

We know that

Power = Force x Velocity

$$\Rightarrow F_{in}v_{in} = F_{out}v_{out}$$

or (Power)$_{in}$ = (Power)$_{out}$

Example 1.1

An input cylinder with a diameter of 30 mm is connected to an output cylinder with a diameter of 80 mm (Fig. 1.3). A force of 1000 N is applied to the input cylinder.

(a) What is the output force?

(b) How far do we need to move the input cylinder to move the output cylinder 100 mm?

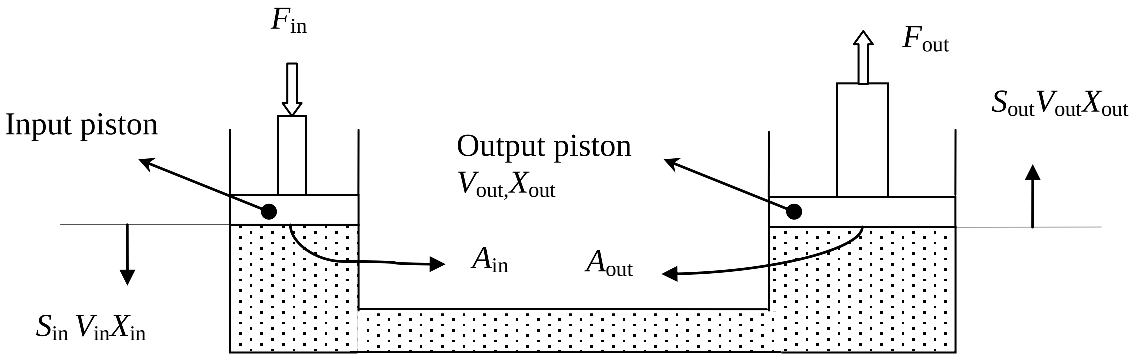


Figure 1.3

Solution: Since the volume of a cylinder equals the product of its cross-sectional area and its height, we have

$$A_{in}X_{in} = A_{out}X_{out}$$

whereX_{in} is the downward movement of the input piston and X_{out} is the upward movement of the output piston. Hence we get

$$\frac{X_{out}}{X_{in}} = \frac{A_{in}}{A_{out}}$$

The piston stroke ratio X_{out}/X_{in} equals the piston area ratio A_{in}/A_{out}. For a piston area of 10, the output force F_{out} increases by a factor of 10, but the output motion decreases by a factor of 10.

Thus, the output force is greater than the input force, but the output movement is less than the input force and the output movement is less than the input movement. Hence, we can write by combining equations

$$A_{in}X_{in} = A_{out}X_{out} \text{ and } \frac{X_{out}}{X_{in}} = \frac{A_{in}}{A_{out}}$$

that

$$\frac{F_{out}}{F_{in}} = \frac{X_{in}}{X_{out}}$$

$$\Rightarrow W_{in} = W_{out}$$

Hence, the input work equals the output work.
Given F_{in} = 1000 N, A_1 = 0.7854 × 30^2 mm^2 and A_2 = 0.7854 × 80^2 mm^2, S_{out} = 1000 mm.
To calculate S_{in}andF_2.

(a) **Force on the large piston F_2:**By Pascal's law, we have

$$\frac{F_1}{F_2} = \frac{A_1}{A_2}$$

$$\Rightarrow F_2 = \frac{A_2 \times F_1}{A_1} = \frac{1000\,\text{N}}{0.7854 \times 30^2} \times 0.7854 \times 80^2$$

$$\Rightarrow F_2 = 7111.1\text{N}$$

(b) **Distance moved by the large piston S_{out}:**We also know by the conversation of energy that

$$\frac{F_1}{F_2} = \frac{S_{out}}{S_{in}}$$

$$\Rightarrow S_{in} = \frac{S_{out} \times F_2}{F_1} = \frac{1000 \times 7111.1}{1000}$$

$$\Rightarrow S_{in} = 7111.11 \text{ mm}$$

Example 1.2

A force of P = 850 N is applied to the smaller cylinder of a hydraulic jack (Fig.1.4). The area a of the small piston is 15 cm^2 and the area A of the larger piston is 150 cm^2. What load W can be lifted on the larger piston (a) if the pistons are at the same level, (b) if the large piston is 0.75 m below the smaller one? The mass density ρ of the liquid in the jack is 103 kg /m^3.

Solution: A diagram of a hydraulic jack is shown in Fig. 1.4.A force F is applied to the piston of the small cylinder which forces oil or water into the large cylinder thus raising the piston supporting the load W. The force F acting on the area a produces a pressure p_1 that is transmitted equally in all directions through the liquid. If the pistons are at the same level, the pressure p_2 acting on the larger piston must equal $p_1$as per Pascal's law

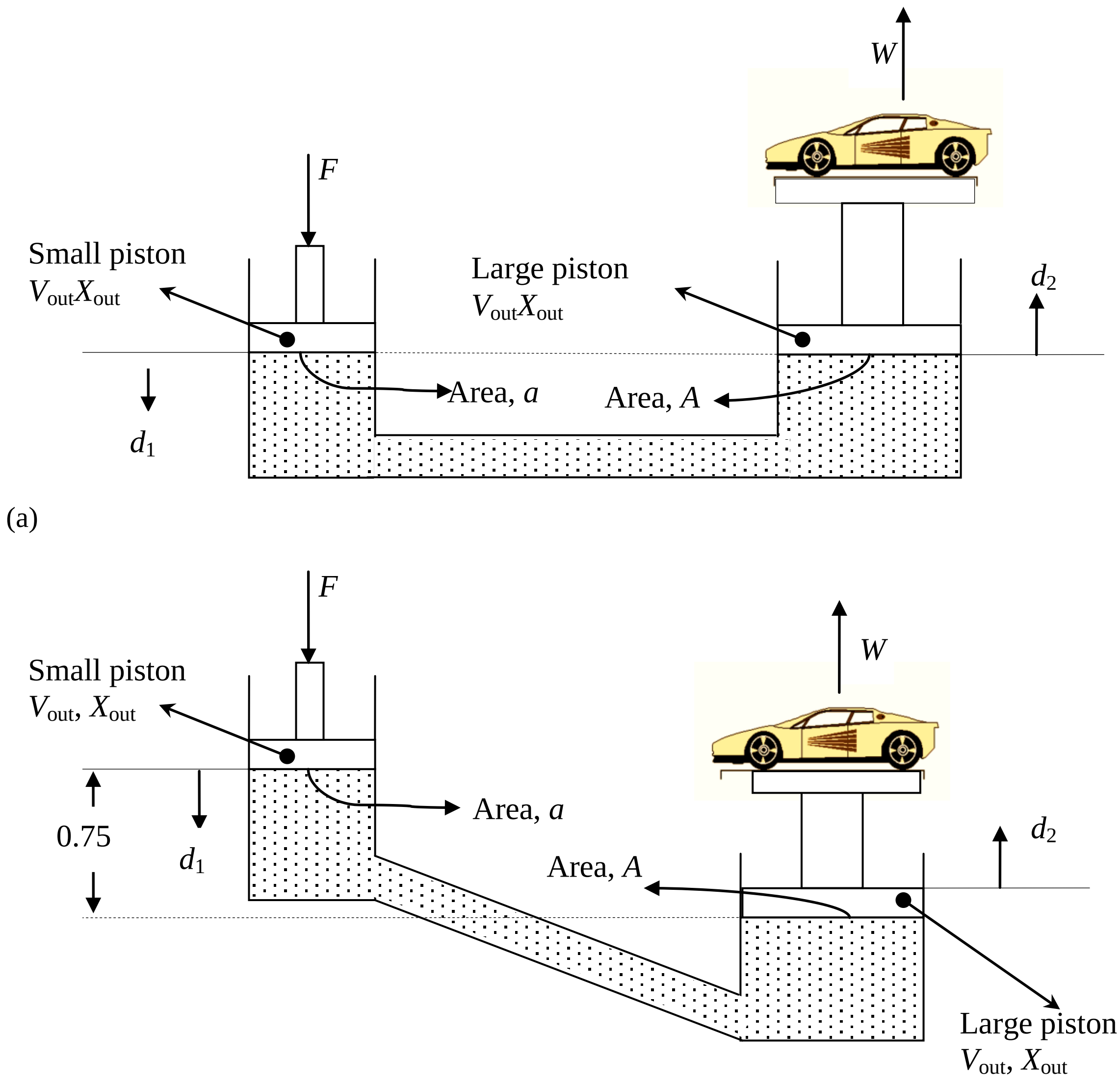


Figure 1.4 (a) Pistons are at same level. (b) Pistons are at different level.

We know that

$$p_1 = \frac{F}{a} \text{ and } p_2 = \frac{W}{A}$$

If $p_1 = p_2$, a small force can raise a larger load W. The jack has a mechanical advantage of A/a.

(a) Now $P = 850\,\text{N}, a = 15/1000\ \text{m}^2, A = 150/10000\ \text{m}^2$. Using Pascal's law we can write

$$p_1 = p_2$$
$$\Rightarrow \frac{F}{a} = \frac{W}{A}$$
$$\Rightarrow W = \frac{F \times A}{a} = \frac{850 \times 1.5}{0.15} = 8500\,\text{N}$$

Now

$$\text{Mass lifted} = \frac{W}{g} = \frac{8500}{9.81} = 868\,\text{kg}$$

(b) If the larger piston is a distance h below the smaller, the pressure p_2 is greater than p_1, due to the head h, by an amount ρg where ρ is the mass density of the liquid:

$$p_2 = p_1 + \rho gh$$

Now

$$p_1 = \frac{F}{a} = \frac{850}{15 \times 10^{-4}} = 56.7 \times 10^4 \text{ N/m}^2$$

$$\rho = 103 \text{kg/m}^3$$

$$h = 0.75 \text{m}$$

So

$$p_2 = 56.7 \times 104 + (103 \times 9.81) \times 0.75$$

$$= 57.44 \times 104 \text{N/m}^2$$

Now

$$W = p_2 A = 57.44 \times 104 \times 150 \times 10^{-4} = 8650 \text{N}$$

Therefore

$$\text{Mass lifted} = \frac{W}{g} = 883 \text{kg}$$

Example 1.3

Two hydraulic cylinders are connected at their piston ends (cap ends rather than rod ends) by a single pipe (Fig. 1.5). Cylinder A has a diameter of 50 mm and cylinder B has a diameter of 100 mm. A retraction force of 2222 N is applied to the piston rod of cylinder A. Determine the following:

(a) Pressure at cylinder A.
(b) Pressure at cylinder B.
(c) Pressure in the connection pipe.
(d) Output force of cylinder B.

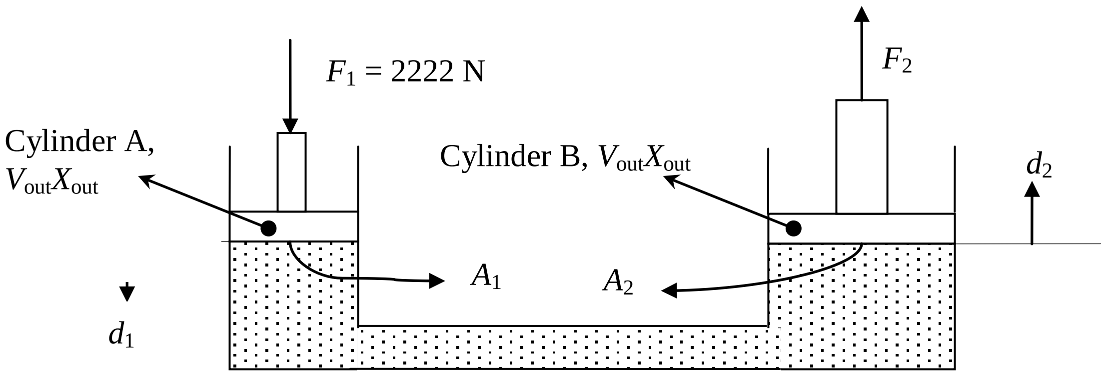


Figure 1.5

Area of the piston of cylinder A is

$$\frac{\pi}{4}(50)^2 = 1963.5 \text{mm}^2$$

Area of the piston of cylinder B is

$$\frac{\pi}{4}(100)^2 = 7853.8 \text{mm}^2$$

(a) Pressure in cylinder A is given by

$$\frac{\text{Force}}{\text{Area}} = \frac{2222}{1963.5} = 1.132\frac{\text{N}}{\text{mm}^2} = 1.132\,\text{MPa}$$

(b) By Pascal's law, pressure in cylinder A = pressure in cylinder B = 1.132 MPa

(c) By Pascal's law, pressure in cylinder A = pressure in cylinder B = pressure in the pipe line = 1.132 MPa

(d) Force on the large piston (cylinder B) F_2: By Pascal's law, we have

$$\frac{F_1}{F_2} = \frac{A_1}{A_2}$$

$$\Rightarrow F_2 = \frac{A_2 \times F_1}{A_1} == \frac{2222\,\text{N}}{1963.5\,\text{mm}^2} \times 7853.8\,\text{mm}^2 = 8888\ \text{N}$$

Example 1.4

A pump delivers oil to a cylindrical storage tank, as shown in Fig. 1.6. A faulty pressure switch, which controls the electric motor driving the pump, allows the pump to fill the tank completely. This causes the pressure p_1 near the base of the tank to build to 103.4 kPa.

(a) What force is exerted on the top of the tank?

(b) What does the pressure difference between the tank top and point 1 say about Pascal's law?

(c) What must be true about the magnitude of system pressure if the changes in pressure due to elevation changes can be ignored in a fluid power system (assume the specific gravity of oil to be 0.9).

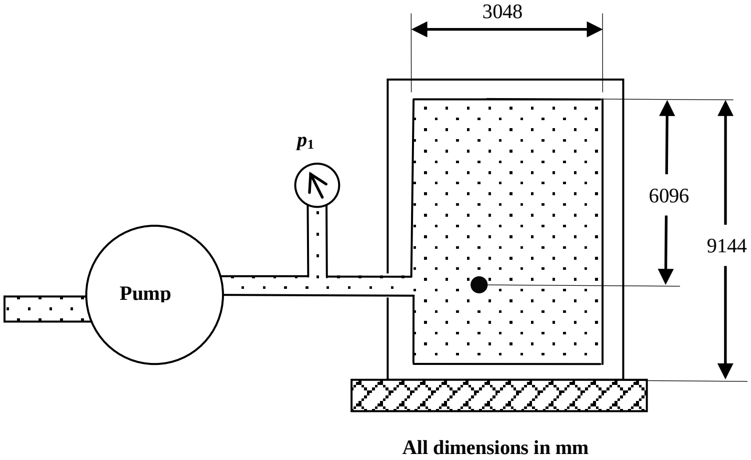


Figure 1.6

(a) We know that

$$\begin{aligned}\Delta p &= \gamma(\Delta H)\\ &= 900 \times 9.81\frac{\text{N}}{\text{m}^3} \times (6.096\ \text{m})\\ &= 53821.6\ \text{Pa}\\ &= 53.822\ \text{kPa}\end{aligned}$$

Thus,

$$F_{\text{top of tank}} = 103.4 - 53.82 = 49.58\ \text{kPa}$$

(b) Now

$$F = \text{Pressure} \times \text{Area}$$
$$= 49.58 \times 1000 \times \frac{\pi}{4}(3.048)^2$$
$$= 361755\,\text{N}$$
$$= 361.76\ \text{kN}$$

Pascal's law states that pressure in a static body of fluid is transmitted equally only at the same elevation level. Pressure increases with depth and vice versa in accordance with the following equation: $\Delta p = \gamma(\Delta H)$.

(c) Changes in pressure due to elevation changes can be ignored in a fluid power system as long as they are small compared to the magnitude of the system pressure produced at the pump discharge port.

Example 1.5

The hydraulic jack, shown in Fig. 1.7, is filled with oil. The large and small pistons have diameters of 75 and 25 mm, respectively. What force on the handle is required to support a load of 8896 N? If the force moves down by 125 mm, how far is the weight lifted?

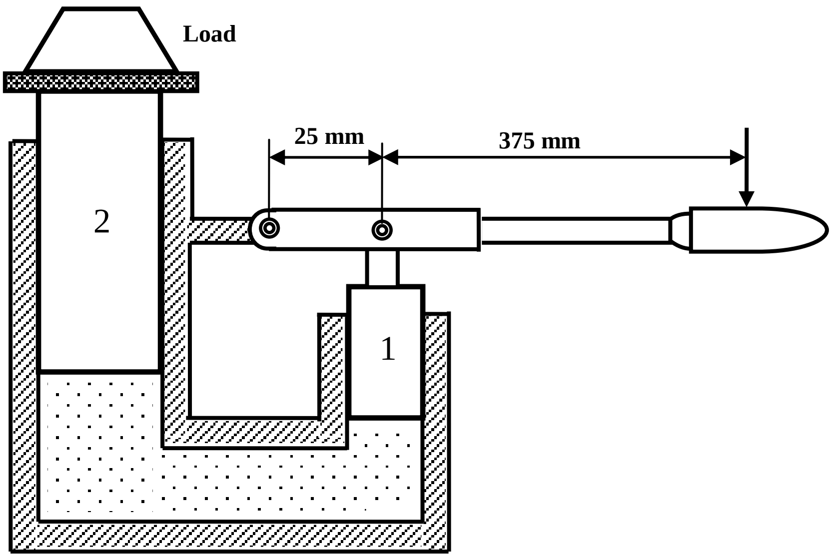


Figure 1.7

Solution: The relation for the lever force system gives

$$F \times 400 = F_1 \times 25$$
$$\Rightarrow F = \frac{F_1}{16}$$

Now since the oil pressure must remain the same everywhere, we have $p_1 = p_2$. Therefore

$$\frac{F_1}{A_1} = \frac{F_2}{A_2}$$

$$\Rightarrow \frac{F_1}{\pi \times 0.025^2 / 4} = \frac{8896}{\pi \times 0.075^2 / 4}$$

$$\Rightarrow F_1 = 988.44\,\text{N}$$

From the relation obtained above, we get

$$F = \frac{F_1}{16} = \frac{988.44}{16} = 61.78\,\text{N}$$

The force moves by 125 mm. The force displacement diagram is shown in Fig. 1.8

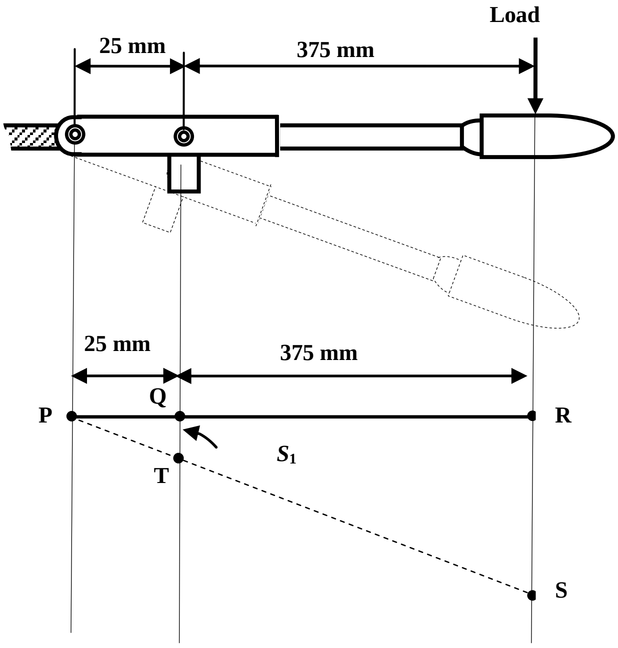


Figure 1.8 Force displacement diagram

From Fig. 1.8 we have

$$\frac{\text{RS}}{\text{QT}} = \frac{\text{RP}}{\text{PQ}}$$

$$\Rightarrow \frac{\text{RS}}{S_1} = \frac{\text{RP}}{\text{PQ}}$$

$$\Rightarrow S_1 = \frac{\text{RS} \times \text{PQ}}{\text{RP}} = \frac{150 \times 25}{400} = 9.375\,\text{mm}$$

Now

$$A_1 S_1 = A_2 S_2$$

$$\Rightarrow \pi \times \frac{25^2}{4} \times 9.375 = \pi \times \frac{75^2}{4} \times S_2$$

$$\Rightarrow S_2 = \frac{1}{9} \times 9.375 \cong 1\,\text{mm}$$

Hence, 150 mm stroke length of lever moves the load of 8896 N by only 1 mm. In other words, mechanical advantage is obtained at the expense of distance traveled by the load.

Example 1.6

In the hydraulic device shown in Fig.1.9, calculate the output torque T_2, if the input torque T_1 =10 N-cm. Use the following data: radius R_1 = 2 cm, diameter d_1 = 8 cm, radius R_2 = 4 cm, diameter d_2 = 24 cm.

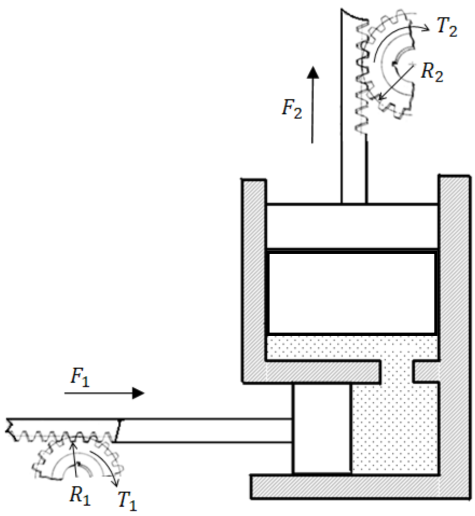


Figure 1.9

Solution: We can use Pascal's law and write

$$p_1 = p_2$$
$$\Rightarrow \frac{F_1}{F_2} = \frac{A_1}{A_2}$$
$$\Rightarrow \frac{F_1}{d_1^2} = \frac{F_2}{d_2^2}$$

Since torque $T = F \cdot R$ which implies $F = T / R,$ we can also write

$$\frac{T_1}{R_1 d_1^2} = \frac{T_2}{R_2 d_2^2}$$

$$\Rightarrow T_2 = \frac{T_1 \times R_2 d_2^2}{R_1 d_1^2}$$
$$\Rightarrow T_2 = \frac{10 \times 4 \times 24^2}{2 \times 8^2} = 180 \text{ N cm}$$

Alternate method: We know that

$$\text{Torque } T = \text{Force} \times \text{Radius of the gear}$$

Consider gear 1. We have

$$T_1 = F_1 R_1$$
$$\Rightarrow F_1 = \frac{T_1}{R_1} = \frac{10}{2} = 5 \text{ N}$$

Now

$$p_1 = \frac{F_1}{A_1}$$

whereA_1 is the area of the horizontal piston given by

$$\frac{\pi}{4}(d^2) = \frac{\pi}{4}(8^2) = 49.84 \text{ cm}^2$$

So

$$p_1 = \frac{5}{49.84} = 0.100 \text{ N / cm}^2$$

According to Pascal's law, $p_1 = p_2 = 0.100 \text{ N / cm}^2$. Here

$$p_2 = \frac{F_2}{A_2}$$

whereA_2 is the area of the horizontal piston given by

$$\frac{\pi}{4}(d^2) = \frac{\pi}{4}(24^2) = 452.16 \text{ cm}^2$$

So

$$p_2 = \frac{F_2}{A_2}$$

$$\Rightarrow 0.100 = \frac{F_2}{452.16}$$

$$\Rightarrow F_2 = 45.216 \text{ N}$$

Now

$$T_2 = F_2 R_2 = 45.216 \times 4 = 180.8 \text{ N cm}$$

Example 1.7

A hydraulic system has 380 L reservoir mounted above the pump to produce a positive pressure (above atmospheric) at the pump inlet, as shown in Fig. 1.10. The purpose of the positive pressure is to prevent the pump from cavitating, when operating, especially at start up. If the pressure at the pump inlet is to be 0.35 bar prior to turning the pump ON and the oil has a specific gravity of 0.9, what should the oil level be above the pump inlet?

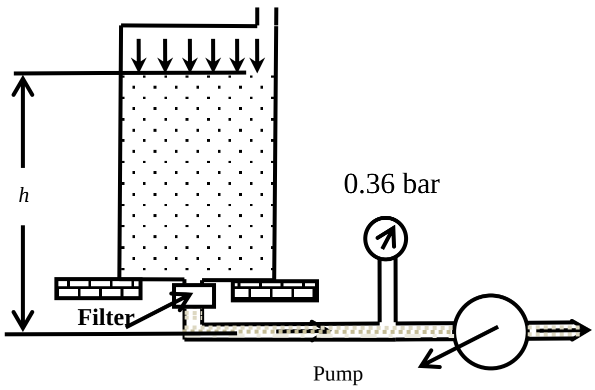


Figure 1.10

We know that

$$p_{\text{oil}} = \gamma_{\text{oil}} H_{\text{oil}}$$

$$\Rightarrow H_{\text{oil}} = \frac{p_{\text{oil}}}{\gamma_{\text{oil}}} = \frac{0.35 \times 10^5 \text{ N/m}^2}{0.90 \times 9797 \text{ N/m}^3} = 3.96 \text{ m}$$

Thus, oil level should be 3.96 m above the pump inlet.

Example 1.8

For the hydraulic pressure shown in Fig. 1.11, what would be the pressure at the pump inlet if the reservoir were located below the pump so that the oil level would be 1.22 m below the pump inlet? The specific gravity of oil is 0.90. Ignore frictional losses and changes in kinetic energy on the pressure at the pump inlet. Would this increase or decrease the chances for having pump cavitation? If yes, why?

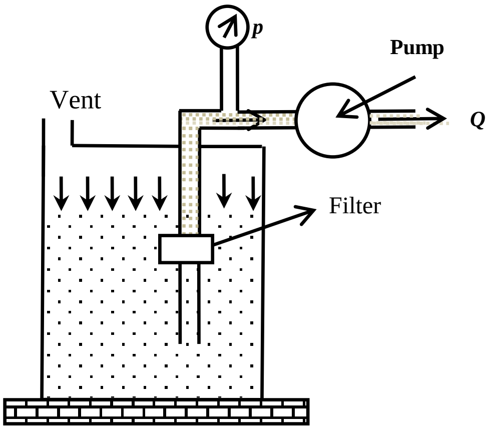


Figure 1.11

Solution: We know that

$$p_{\text{oil}} = -\gamma_{\text{oil}} H_{\text{oil}}$$
$$= -0.90 \times 9797 \frac{\text{N}}{\text{m}^3} \times 1.22\,\text{m} = -10757\,\text{Pa}$$
$$= -0.10757 \text{ bar (gauge)}$$

Frictional losses and changes in kinetic energy would cause the pressure at the pump inlet to increase negatively (greater suction pressure) because pressure energy decreases as per Bernoulli's equation. This would increase the chances for having the pump cavitation because the pump inlet pressure more closely approaches the vapor pressure of the fluid (usually about 0.34 bar suction) or −0.34 bar (gauge), allowing for the formation and collapse of vapor bubbles

Example 1.9

A hydraulic cylinder is to compress a body down to bale size in 10 s. The operation requires a 3 m stroke and a 40000 N force. If a 10 MPa pump has been selected, assuming the cylinder to be 100% efficient, find

(a) The required piston area.
(b) The necessary pump flow rate.
(c) The hydraulic power delivered to the cylinder.
(d) The output power delivered to the load.
(e) Also solve parts (a)–(d) assuming a 400 N friction force and a leakage of 1 LPM. What is the efficiency of the cylinder with the given friction force and leakage?

Solution:

(a) Since the fluid pressure is undiminished, we have $p_1 = p_2 = 10$ MPa . Now

$$p_2 = \frac{F_2}{A_2} \Rightarrow A_2 = \frac{F_2}{p_2} = \frac{40000}{10 \times 10^6} = 0.004\,\text{m}^2$$

which is the required piston area.

(b) Stroke length $l = 3$ m , time for stroke $t = 10$ s , piston area $A_2 = 0.004\,\text{m}^2$. Flow rate is

$$Q = \frac{A_2 l}{t} = \frac{0.004 \times 3}{10} = 12 \times 10^{-4}\,(\mathrm{m^3/s}) = 72\,\mathrm{LPM}$$

(c) Power delivered to the cylinder

$$\text{Power} = \text{Pressure} \times \text{Flow rate}$$
$$= (10 \times 10^6) \times (12 \times 10^{-4})$$
$$= 12000\,\mathrm{W} = 12\,\mathrm{kW}$$

(d) Power delivered to load is

$$\text{Power} = \frac{F \times l}{t} = \frac{40000 \times 3}{10} = 12000\,\mathrm{W} = 12\,\mathrm{kW}$$

Since efficiency is assumed to be 100%, both powers are the same.

(e) With a friction force of $f = 400$ N and 1LPM leakage, piston area is

$$A_2' = \frac{F_2 + f}{p_2} = \frac{40000 + 400}{10 \times 10^6} = 0.00404\ \mathrm{m^2}$$

Now pump flow rate is

$$Q' = \frac{A_2' l}{t} = \frac{0.00404 \times 3}{10}$$
$$= 12.12 \times 10^{-4}\,\mathrm{m^3/s} = 72.72\,\mathrm{LPM}$$

So

$$\text{Total flow} = Q' + \text{Leakage}$$
$$= 72.72 + 1$$
$$= 73.72\,\mathrm{LPM}$$
$$= 12.287 \times 10^{-4}\,\mathrm{m^3/s}$$

Power delivered to the cylinder is given by

$$p \times Q = (10 \times 10^6) \times (12.287 \times 10^{-4})$$
$$= 12287\,\mathrm{W} = 12.287\,\mathrm{kW}$$

Power delivered to the load is

$$\text{Power} = \frac{F \times l}{t} = \frac{40000 \times 3}{10} = 12000\,\mathrm{W} = 12\,\mathrm{kW}$$

It will remain the same as without losses. The efficiency of the cylinder

$$\eta = \frac{\text{Power delivered to load}}{\text{Power delivered to cylinder}}$$
$$= \frac{12}{12.287} \times 100 = 97.66\%$$

Example 1.10

An automobile lift raises a 15600 N car 2.13 m above the ground floor level. If the hydraulic cylinder contains a piston of diameter 20.32 cm and a rod of diameter 10.16 cm, determine the

(a) Work necessary to lift the car.
(b) Required pressure.
(c) Power if the lift raises the car in 10 s.
(d) Descending speed of the lift for 0.000629 $\mathrm{m^3/s}$ flow rate.
(f) Flow rate for the auto to descend in 10 s.

Solution:

(a) We have

Work necessary to lift the car = Force × Distance

= 15600 × 2.13 m = 33200 N m

(b) We have

$$\text{Piston area} = \frac{\pi(0.2032)^2 \text{ m}^2}{4} = 0.0324\,\text{m}^2$$

So required pressure is

$$\text{Pressure} = \frac{\text{Force}}{\text{Area}} = \frac{15600}{0.0324}$$
$$= 481000\,\text{N/m}^2 = 481 \text{ kPa}$$

(c) We have

$$\text{Power} = \frac{\text{Work done}}{\text{Time}} = \frac{33200}{10}$$
$$= 3320\text{N m/s} = 3320 \text{ W} = 3.32 \text{ kW}$$

(d) Q = 0.000629 m^3/s.

$$\text{Annulus area} = \frac{\pi(0.2032)^2 - \pi(0.1016)^2}{4} = 0.0243\,\text{m}^2$$

So

$$\text{Decending speed of the lift} = \frac{\text{Flow rate}}{\text{Annulus area}} = \frac{0.000629}{0.0243} = 0.0259 \text{ m/s}$$

(e) Flow rate for the auto to descend in 10 s is

$$\text{Flow rate} = \text{Annulus area} \times \frac{\text{Distance}}{\text{Time}}$$
$$= 0.0243 \times \frac{2.13}{10} = 0.00518 \text{ m}^3/\text{s}$$

1.3.2 *Practical Applications of Pascal's Law*

The practical applications of Pascal's law are numerous. In this section, two applications of Pascal's law are presented: (a) The hand-operated hydraulic jack and (b) the air-to-hydraulic pressure booster.

1.3.2.1 *Hand-Operated Hydraulic Jack*

This system uses a piston-type hand pump to power a hydraulic load cylinder for lifting loads, as illustrated in Fig. 1.12. The operation is as follows:

1. A hand force is applied at point A of handle ABC which is pivoted at point C. The piston rod of the pump cylinder is pinned to the input handle of the pump piston at point B.
2. The pump cylinder contains a small-diameter piston that is free to move up and down. The piston and rod are rigidly connected together. When the handle is pulled up, the piston rises and creates a vacuum in the space below it. As a result, the atmospheric pressure forces the oil to leave the oil tank and flow through check valve 1 to fill the void created below the pump piston. This is the suction process.
3. A check valve allows flow to pass in only one direction, as indicated by the arrow. When the handle is pushed down, oil is ejected from the small-diameter pump cylinder and it flows through check valve 2 and enters the bottom end of the large-diameter load cylinder.

4. The load cylinder is similar in construction to the pump cylinder and contains a piston connected to a rod. Pressure builds up below the load piston and equals the pressure generated by the pump piston. The pressure generated by the pump piston equals the force applied to the pump piston rod divided by the area of the pump piston.
5. The load that can be lifted equals the product of the pressure and the area of the load piston. Also, each time when the input handle is cycled up and down, a specified volume of oil is ejected from the pump to raise the load cylinder a given distance.
6. The bleed valve is a hand-operated valve, which, when opened, allows the load to be lowered by bleeding oil from the load cylinder back to the oil tank.

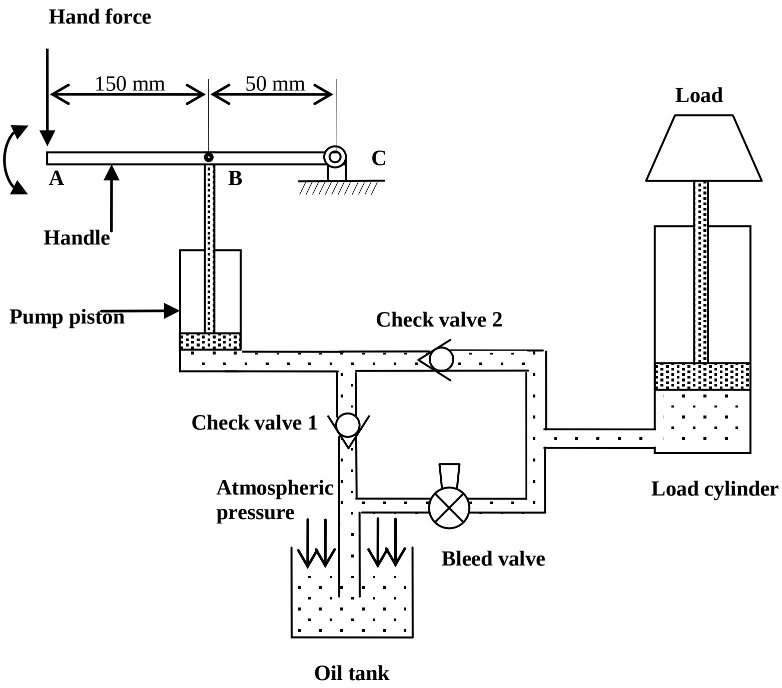


Figure 1.12 Application of Pascal's law: Hand-operated hydraulic jack

1.3.2.2 Air-to-Hydraulic Pressure Booster

This device is used for converting shop air into higher hydraulic pressure needed for operating hydraulic cylinders requiring small to medium volumes of higher pressure oil. It consists of a cylinder containing a large-diameter air piston driving a small-diameter hydraulic piston that is actually a long rod connected to the piston. Any shop equipped with an airline can obtain smooth, efficient hydraulic power from an air-to-hydraulic pressure booster hooked into the air line. The alternative would be a complete hydraulic system including expensive pumps and high-pressure valves. Other benefits include space savings and low operating and maintenance costs.

Figure 1.13 shows an application where an air-to-hydraulic pressure booster supplies high-pressure oil to a hydraulic cylinder whose short stroke piston is used to clamp a workpiece to a machine tool table. Since shop air pressure normally operates at 100 psi, a pneumatically operated clamp would require an excessively large cylinder to rigidly hold the workpiece while it is being machined.

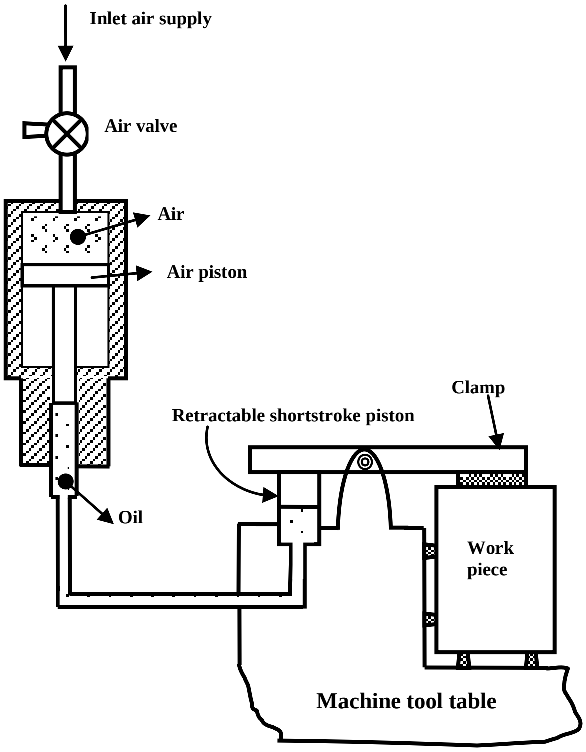


Figure 1.13 Application of Pascal's law: Air-to-hydraulic pressure booster

The air-to-hydraulic pressure booster operates as follows. Let us assume that the air piston has 10 cm^2 area and is subjected to a 10 bar air pressure. This produces a 1000 N force on the booster's hydraulic piston. Thus, if the area of the booster's hydraulic piston is 1 cm^2, the hydraulic oil pressure is 100 bar. As per Pascal's law, this produces 100 bar oil at the short stroke piston of the hydraulic clamping cylinder mounted on the machine tool table.

The pressure ratio of an air-to-hydraulic pressure booster can be found by using the following equation:

$$\text{Pressure ratio} = \frac{\text{Output oil pressure}}{\text{Input oil pressure}}$$

$$= \frac{\text{Area of air piston}}{\text{Area of hydraulic piston}}$$

Substituting into the above equation for the earlier mentioned pressure booster, we have

$$\text{Pressure ratio} = \frac{10000\text{ kPa}}{1000\text{ kPa}} = \frac{10\,\text{cm}^2}{1\,\text{cm}^2}$$

For a clamping cylinder piston area of 0.5 cm^2, the clamping force equals 1000 N/cm^2 × 0.5 cm^2 or 500 N. To provide the same clamping force of 500 N without booster requires a clamping cylinder piston area of 5 cm^2, assuming 10 bar air pressure. Air-to-hydraulic pressure boosters are available in a wide range of pressure ratios and can provide hydraulic pressures up to 1000 bar using approximately 7 bar shop air.

Example 1.11

An operator makes 15 complete cycles in 15 s interval using the hand pump shown in Fig. 1.14. Each complete cycle consists of two pump strokes (intake and power). The pump has a piston of diameter 30 mm and the load cylinder has a piston of diameter 150 mm. The average hand force is 100 N during each power stroke.

(a) How much load can be lifted?

(b) How many cycles are required to lift the load by 500 mm, assuming no oil leakage? The pump piston has 20 mm stroke.

(c) What is the output power assuming 80% efficiency?

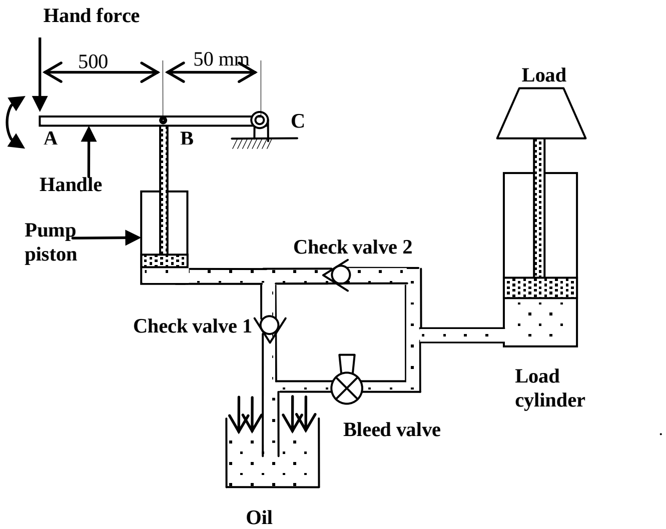


Figure 1.14

Solution: Given:pump diameter $d = 30\,\text{mm}$, load cylinder diameter $D = 150\,\text{mm}$, hand force $f = 100\,\text{N}$, number of cycles $n = 15$ strokes/s, pump piston force

$$F_1 = \frac{100 \times 550}{50} = 1100\ \text{N}$$

(a) **Load capacity:** Now since the pressure remains undiminished throughout, we have $p_1 = p_2$. Therefore,

$$\frac{F_1}{A_1} = \frac{F_2}{A_2}$$

$$\Rightarrow F_2 = \frac{\pi D^2/4}{\pi d^2/4} F_1 = \frac{150^2}{30^2} \times 1100 = 27500\,\text{N} = 27.5\,\text{kN}$$

(b) **Number of cycles:**Stroke length l = 20 mm. Let the number of strokes be *N*.Then assuming no leakage, we get

$$Q_1 = Q_2$$

where

Q_1= Total volume of fluid displaced by pump piston
= (Area ×Stroke)×Number of strokes = $N \times A_1 l$
Q_2= Flow rate of load cylinder= (Area × Stroke of load cylinder)
=$A_2 \times 500$

So we get

$$N \times A_1 l = A_2 \times 500$$

$$\Rightarrow N = \frac{150^2}{20 \times 30^2} \times 500 = 625$$

Hence, the number of cycles required is 625.

(c) **Output power:**
Input power = $F_1 \times l \times n$
Output power = $\eta \times F_1 \times l \times n = 0.8 \times 1100 \times 0.02 \times 15 = 264\,\text{W}$

Example 1.12
For the pressure booster of Fig. 1.15, the following data are given:
Inlet oil pressure (p_1) = 1 MPa
Air piston area (A_1) = 0.02 m^2
Oil piston area (A_2) = 0.001 m^2
Load carrying capacity = 300000 N
Find the load required on load piston area A_3.

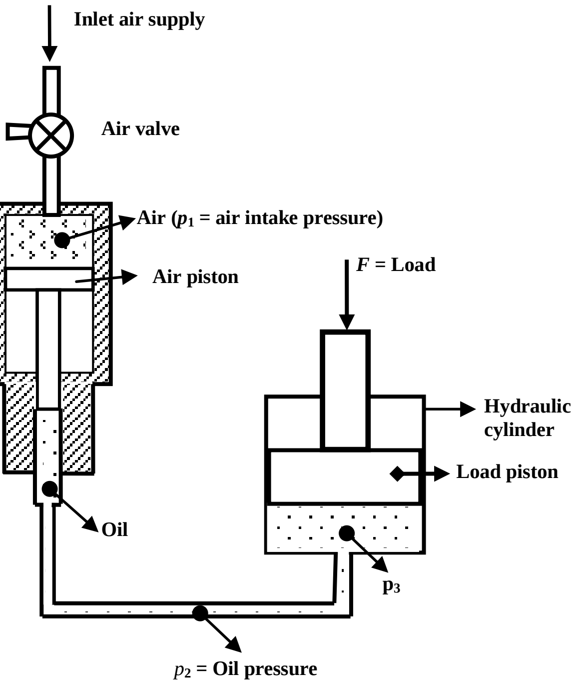


Figure 1.15

Solution: We know that

$$p_2 = \frac{p_1 A_1}{A_2} = \frac{1\,\text{MPa} \times 0.02\,\text{m}^2}{0.001\text{m}^2} = 20\,\text{MPa}$$

Also $p_3 = p_2 = 20\,\text{MPa}$. So

$$A_3 = \frac{F}{p_3} = \frac{300000\,\text{N}}{20 \times 10^6\ \text{N}/\text{m}^2} = 0.015\,\text{m}^2$$

1.4 Conservation of Energy

The first law of thermodynamics states that energy can neither be created nor be destroyed. Moreover, all forms of energy are equivalent. The various forms of energy present in fluid flow are briefly discussed. The total energy includes potential energy due to elevation and pressure and also kinetic energy due to velocity. Let us discuss all these in detail.

1. Kinetic energy of a flowing fluid:A body of mass m moving with velocity $v$possesses a kinetic energy (KE), that is,

$$\mathrm{KE} = \frac{mv^2}{2}$$

Thus, if a fluid were flowing with all particles moving at the same velocity, its kinetic energy would also be $(1/2)(m)v^2$; this can be written as

$$\frac{\mathrm{KE}}{\mathrm{Weight}} = \frac{\frac{1}{2}mv^2}{(\gamma)\mathrm{Volume}} = \frac{\frac{1}{2}[(\gamma)\mathrm{Volume}]v^2}{(\gamma)\mathrm{Volume}} = \frac{v^2}{2g}$$

where$g$ is the acceleration due to gravity. In SI units,$v^2/2g$ is expressed as Nm/N = m..

2. Potential energy due to elevation (*z*): Consider a unit weight of fluid as shown in Fig. 1.16.The potential energy of a particle of a fluid depends on its elevation above any arbitrary plane. We are usually interested only in the differences of elevation, and therefore the location of the datum plane is determined solely by consideration of convenience. A fluid particle of weight *W* situated at a distance *Z* above datum possesses a potential energy *Wz*. Thus, in SI units, its potential energy per unit weight is expressed asNm/N = m.

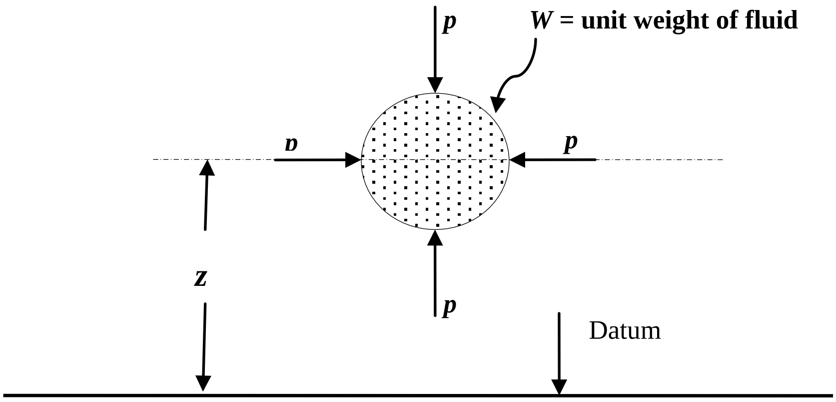


Figure 1.16 Potential energy due to elevation

3. Potential energy due to pressure (PE): This term represents the energy possessed by a fluid per unit weight of fluid by virtue of the pressure under which the fluid exists:

$$\mathrm{PE} = \frac{p}{\gamma}$$

whereγ is the specific weight of the fluid. PE has the unit of meter. The total energy possessed by the weight of fluid remains constant (unless energy is added to the fluid via pumps or removed from the fluid via hydraulic motors or friction) as the weight *W* flows through a pipeline of a hydraulic system. Mathematically, we have

$$E_{\mathrm{Total}} = z + \frac{p}{\gamma} + \frac{v^2}{2g}$$

Energy can be changed from one form to another. For example, the chunk of fluid may lose elevation as it flows through a hydraulic system and thus has less potential energy. This, however, would result in an equal increase in either the fluid's pressure energy or its kinetic energy. The energy equation takes into account the fact that energy is added to the fluid via pumps and that energy is removed from the fluid via hydraulic motors and friction as the fluid flows through actual hydraulic systems.

Example 1.13

Oil with specific gravity 0.9 enters a tee, as shown in Fig. 1.18, with velocity v_1 =5 m/s. The diameter at section 1 is 10 cm, the diameter at section 2 is 7 cm and the diameter at section 3 is 6 cm. If equal flow rates are to occur at sections 2 and 3, find the velocities v_2 and v_3.

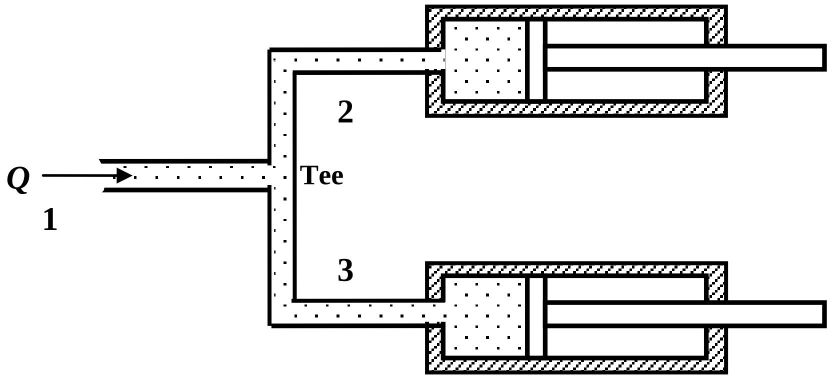


Figure 1.18

Solution: Assuming no leakage

$$Q_1 = Q_2 + Q_3$$

Also,

$$Q_2 = Q_3 = \frac{1}{2}Q_1 = \frac{1}{2}A_1 v_1$$

$$= \frac{1}{2} \times \frac{\pi d_1^2}{4} \times v_1 = \frac{1}{2} \times \frac{\pi \times 0.1^2}{4} \times 5 = 19.63 \times 10^{-3}\,\mathrm{m^3/s}$$

Therefore,

$$v_2 = \frac{Q_2}{A_2} = \frac{19.63 \times 10^{-3}}{\pi \times 0.07^2 / 4} = 5.1\ \mathrm{m/s}$$

$$v_3 = \frac{Q_3}{A_3} = \frac{19.63 \times 10^{-3}}{\pi \times 0.06^2 / 4} = 6.942\,\mathrm{m/s}$$

Example 1.14

A double-rod cylinder is one in which a rod extends out of the cylinder at both ends (Fig. 1.19). Such a cylinder with a piston of diameter 75 mm and a rod of diameter 50 mm cycles through 254 mm stroke at 60 cycles/min. What LPM size pump is required?

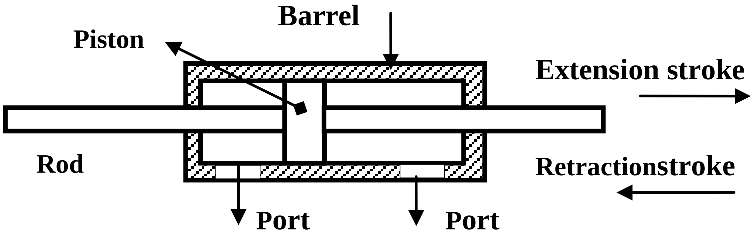


Figure 1.19

Solution: The annulus area is

$$A_{\text{Annulus}} = \frac{\pi(75^2 - 50^2)}{4} = 2454 \text{ mm}^2$$

Volume of oil displaced per minute (m^3/s) is

Area × Stroke length × No. of cycles per second

Now

$$Q = \frac{\pi(75^2 - 50^2)\text{mm}^2}{4} \times 10^{-6} \text{ m}^2 \times \left\{\frac{254}{1000} \times 2\right\} \text{(m)} \times \frac{60}{60}\text{(s)}$$

$$= 0.001296 \text{ m}^3/\text{s} = 77.8 \text{ LPM}$$

We can select 80 LPM pump.

Example 1.15

A cylinder with a piston of diameter 8 cm and a rod of diameter 3 cm receives fluid at 30 LPM. If the cylinder has a stroke of 35 cm, what is the maximum cycle rate that can be accomplished?

Solution: We know that

Volume of oil displaced per minute (m^3/min) = Area × Stroke length × No. of cycles per minute

So

$$Q = \frac{\pi(0.08^2)\text{m}^2}{4} \times \left\{\frac{35}{100}\right\}\text{(m)} \times N\text{(cycles/min)} + \frac{\pi(0.08^2 - 0.03^2)\text{ m}^2}{4} \times \left\{\frac{35}{100}\right\}\text{ (m)} \times N\text{(cycles/min)}$$

$$= 0.03\text{m}^3/\text{min}$$

$$\Rightarrow 0.030 = 0.00176 + 0.0015 \times N$$

$$\Rightarrow N = 9.2 \text{ cycles/min}$$

Example 1.16

A hydraulic pump delivers a fluid at 50 LPM and 10000 kPa. How much hydraulic power does the pump produce?

Solution: We have

$$Q = 50 \text{ LPM} = \frac{50}{60 \times 10^3} = 0.833 \times 10^{-3} \text{m}^3/\text{s}$$

Now

$$1 \text{ L} = 1000 \text{ cc} = 1000 \times 10^{-6} \text{ m}^3 = 10^{-3} \text{m}^3$$

So

Power (kW) = p (kPa) ×Q (m^3/s)

$= 10000 \times 0.833 \times 10^{-3}$

= 8.33 kW = 8330 W

1.7 The Energy Equation

The Bernoulli equation discussed above can be modified to account for fractional losses (H_L) between stations 1 and 2. Here $H_L$represents the energy loss due to friction of 1 kg of fluid moving from station 1 to station 2. As discussed earlier, H_p represents the energy head put into the flow by the pump. If there exists a hydraulic motor or turbine between stations 1 and 2, then it removes energy from the fluid. If H_m (motor head) represents the energy per kg of fluid removed by a hydraulic motor, the modified Bernoulli equation (also called the energy equation) is stated as follows for a fluid flowing in a pipeline from station 1 to station 2: The total energy possessed by 1 kg of fluid at station 1 plus the energy added to it by a pump minus the energy removed from it by a hydraulic motor minus the energy it loses due to friction equals the total energy possessed by 1 kg of fluid when it arrives at station 2. The energy equation is as follows, where each term represents a head and thus has the unit of length:

$$z_1 + \frac{p_1}{\gamma} + \frac{v_1^2}{2g} + H_p - H_m - H_L = z_2 + \frac{p_2}{\gamma} + \frac{v_2^2}{2g}$$

1.9 Elements of Hydraulic Systems and the Corresponding Bernoulli's Equation

The main elements of hydraulic systems are pump, motor, pipes, valves and fittings. Let us write the energy flow from point1 to point 2 as shown in Fig. 1.22.After the fluid leaves point 1, it enters the pump where energy is added. A prime mover, such as an electric motor, drives the pump and the impeller of the pump transfers the energy to the fluid. Then the fluid flows through a piping system composed of a valve, elbows and the lengths of pipe in which energy is dissipated from the fluid and is lost. Before reaching point 2, the fluid flows through a fluid motor that removes some of the energy to drive an external device. The general energy equation accounts for all these energies.

In a particular problem, it is possible that not all of the terms in the general energy equation are required. For example, if there is no mechanical device between the sections of interest, the terms H_p and $H_m$will be zero and can be left out of the equation. If energy losses are so small that they can be neglected, the term H_L can be left out. If both these conditions exist, it can be seen that the energy equation reduces to Bernoulli's equation.

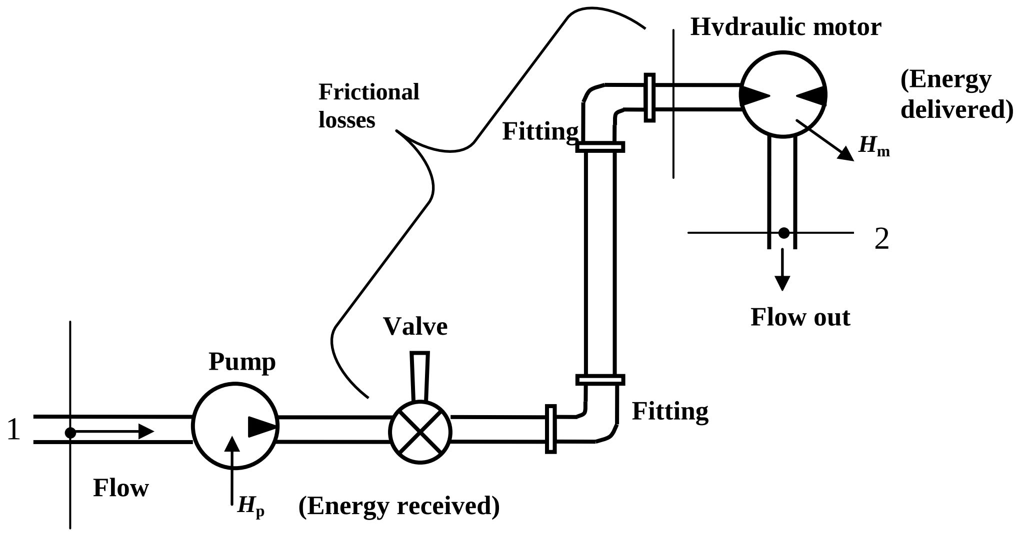


Figure1.22 Elements of a hydraulic system

Example 1.17

(a) Calculate the work required for a pump to pump water from a well to ground level 125 m above the bottom of the well (see Fig.1.23). At the inlet to the pump, the pressure is 96.5 kPa, and at the system outlet, it is 103.4 kPa. Assume the constant pipe diameter. Use γ = 9810 N/m^3, and assume it to be constant. Neglect any flow losses in the system.

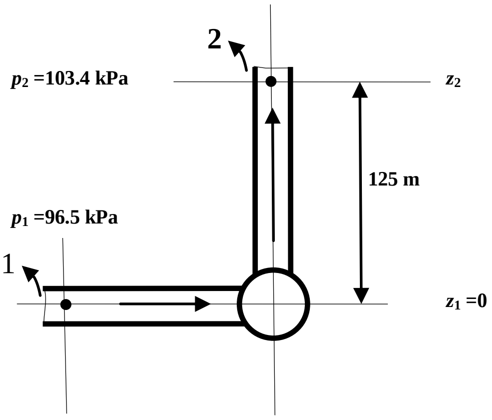


Figure 1.23

Given $z_1 = 0,\ z_2 = 125\,\text{m},\ p_1 = 96.5\,\text{kPa},\ p_2 = 103.4\,\text{kPa},\ H_L = 0,\ D_1 = D_2$. Find H_p.

Assumptions: Steady incompressible flow, no losses

Basic equations:

Continuity: $A_1v_1 = A_2v_2$

Energy equation: $\frac{p_1}{\gamma} + \frac{v_1^2}{2g} + z_1 + H_p = \frac{p_2}{\gamma} + \frac{v_2^2}{2g} + z_2 + H_L$

(b) Solve the above problem if there is friction in the system whose total head loss equals 12.5 m. Given $z_1 = 0,\ z_2 = 125\,\text{m},\ p_1 = 96.5\,\text{kPa},\ p_2 = 103.4\,\text{kPa},\ H_L = 12.5\,\text{m},\ D_1 = D_2$. Find H_p.

Assumptions: Steady incompressible flow, no losses

Basic equations:

Continuity: $A_1v_1 = A_2v_2$

Energy equation: $\frac{p_1}{\gamma} + \frac{v_1^2}{2g} + z_1 + H_p = \frac{p_2}{\gamma} + \frac{v_2^2}{2g} + z_2 + H_L$

Solution:

(a) Write the energy equation

$$\frac{p_1}{\gamma} + \frac{v_1^2}{2g} + z_1 + H_p = \frac{p_2}{\gamma} + \frac{v_2^2}{2g} + z_2 + H_L$$

Note that $v_1 = v_2$ and $H_L = 0$. Thus,

$$H_p = \frac{p_2}{\gamma} - \frac{p_1}{\gamma} + z_2 - z_1$$

With $z_1 = 0$ we get

$$H_p = \frac{103.4\,\text{kPa}}{9.81\,\text{kN/m}^3} - \frac{96.5\,\text{kPa}}{9.81\,\text{kN/m}^3} + 125 = 125.7\,\text{m}$$

(b) Write the energy equation

$$\frac{p_1}{\gamma} + \frac{v_1^2}{2g} + z_1 + H_p = \frac{p_2}{\gamma} + \frac{v_2^2}{2g} + z_2 + H_L$$

As before, $v_1 = v_2$ but $H_L = 12.5$ m.Therefore,

$$H_p = \frac{p_2}{\gamma} - \frac{p_1}{\gamma} + z_2 - z_1 + H_L$$

With $z_1 = 0$ we get

$$H_p = \frac{103.4\,\text{kPa}}{9.81\,\text{kN/m}^3} - \frac{96.5\,\text{kPa}}{9.81\,\text{kN/m}^3} + 137.5 = 138.2\,\text{m}$$

Note that the pump is required to overcome the additional friction head loss, and for the same flow, this requires more pump work. The additional pump work is equal to the head loss.

Example 1.18
A hydraulic turbine is connected as shown in Fig. 1.24. How much power will it develop? Use 1000 kg/ m^3 for the density of water. Neglect the flow losses in the system.

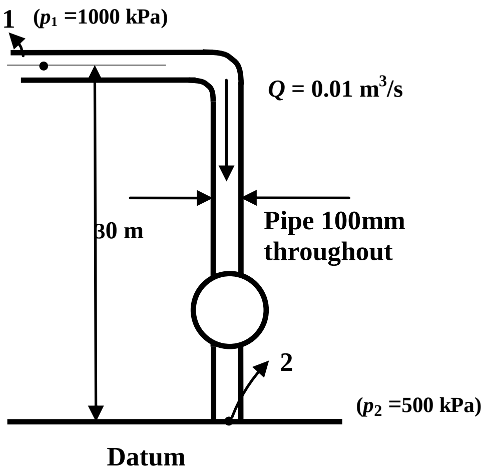


Figure 1.24

Given $z_1 = 30\,\text{m},\ z_2 = 0,\ p_1 = 1000\,\text{kPa},\ p_2 = 500\,\text{kPa},\ H_L = 0,\ D_1 = D_2 = 100\,\text{mm}, Q = 0.01\,\text{m}^3/\text{s}.$
Find turbine power.
Assumptions: Steady incompressible flow, no losses
Basic equations:
Continuity: $A_1 v_1 = A_2 v_2$

Energy: $\frac{p_1}{\gamma} + \frac{v_1^2}{2g} + z_1 + H_p = \frac{p_2}{\gamma} + \frac{v_2^2}{2g} + z_2 + H_L$

Power: $P = H_T \times Q \times \gamma$

Solution: Again let us write the energy equation, but this time for a turbine:

$$\frac{p_1}{\gamma} + \frac{v_1^2}{2g} + z_1 + H_p = \frac{p_2}{\gamma} + \frac{v_2^2}{2g} + z_2 + H_L$$

Since there are no losses in the pipe and the pipe diameter is constant, $v_1 = v_2,\ z_2 = 0,\ z_1 = 30\,\text{m}.$

Therefore, H_T is found as

$$H_T = \frac{p_1 - p_2}{\gamma} + (z_1 - z_2)$$

Using the data given we get

$$H_T = \frac{(1000-500)1000}{1000 \times 9.81} + (30-0) = 80.97\,\text{m}$$

Horsepower is given by

$$\text{Horsepower} = H_T \times Q \times \gamma$$
$$= 80.97 \times 0.01 \times 1000 \times 9.81$$
$$= 7941\ \text{W} = 7.941\ \text{kW}$$

Example 1.19

For the hydraulic system shown in Fig.1.25, the following data are given:
The pump is adding 4 kW to the fluid (i.e., the hydraulic power of the pump).
The pump flow is 0.002 m^3/s.
The pipe has an inside diameter of 25 mm.
The specific gravity of oil is 0.9.
Point 2 is at an elevation of 0.6 m above the oil level, that is, point 1.
The head loss due to friction in the line between points 1 and 2 is 10.
Determine the fluid pressure at point 2, the inlet to the hydraulic motor. Neglect the pressure drop at the strainer. The oil tank is vented to atmosphere.

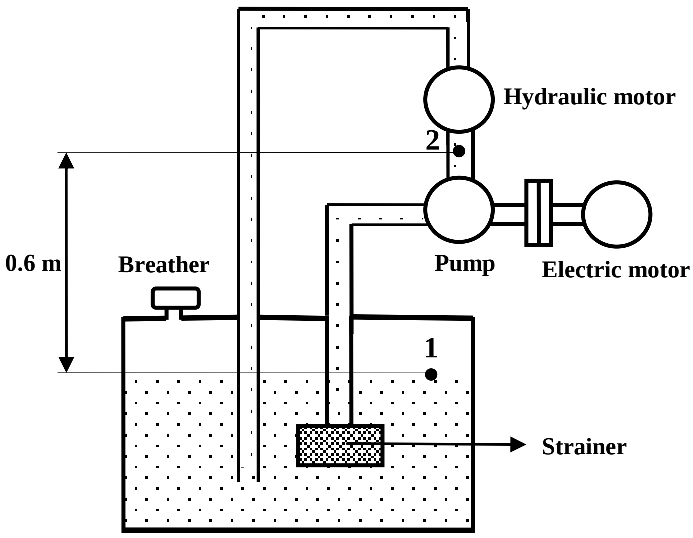


Figure 1.25

Solution:Givenp1 = 0 (as the tank is vented to the atmosphere)

P (kW) = 4 (kW) = 4 × 10^3 W
Q = 0.002 m^3/s
D_p= 25 mm = 0.025 m
SG = 0.9
$z_2 - z_1 =$ 6 m
H_L = 10 m
H_m = 0 (there is no motor between 1 and 2)

The problem can be solved by using the energy equation (Bernoulli's equation):

$$z_1 + \frac{p_1}{\gamma} + \frac{v_1^2}{2g} + H_p - H_m - H_L = z_2 + \frac{p_2}{\gamma} + \frac{v_2^2}{2g}$$

We can take v1 = 0 since the tank cross-section is large. Let us compute some of the unknown terms in the equation. The pump head is given by

$$H_{\mathrm{p}} = \frac{P\ (\mathrm{W})}{\gamma\ (\mathrm{N/m^3}) \times Q\ (\mathrm{m^3/s})}$$

$$= \frac{4\times 10^3}{0.9\times 9800\times 0.002} = 226.7\ \mathrm{m}$$

The velocity head is

$$v_2 = \frac{Q\ (\mathrm{m^3/s})}{A\ (\mathrm{m^2})} = \frac{0.002}{\frac{\pi}{4}(0.025^2)} = 4.07\,\mathrm{m/s}$$

The velocity head is

$$\frac{v_2^2}{2g} = \frac{4.07^2}{2\times 9.81} = 0.85\,\mathrm{m}$$

Substituting the values into the energy equation and rearranging, we can write

$$z_1 + 0 + 0 + 266.7 - 0 - 10 = z_2 + \frac{p_2}{\gamma} + 0.85$$

$$\Rightarrow \frac{p_2}{\gamma} = (z_1 - z_2) + 266.7 - 10 - 0.85 = -6 + 266.7 - 10 - 0.85$$

$$\Rightarrow \frac{p_2}{\gamma} = 209.85\,\mathrm{m}$$

$$\Rightarrow p_2 = 209.85\times 0.9\times 9800 = 1850877\,\mathrm{Pa} = 1850.9\ \mathrm{kPa}$$

Example 1.20

The oil tank for the hydraulic system shown in Fig.1.26is pressurized at 68 kPa gauge pressure. The inlet to the pump is 3 m below the oil level. The pump flow rate is 0.001896 $\mathrm{m^3/s}$. Find the pressure at station 2. The specific gravity of oil is 0.9 and kinematic viscosity of oil is 100 cS. Assume the pressure drop across the strainer to be 6.9 kPa. Also given the pipe diameter is 38 mm and the total length of the pipe is 6 m.

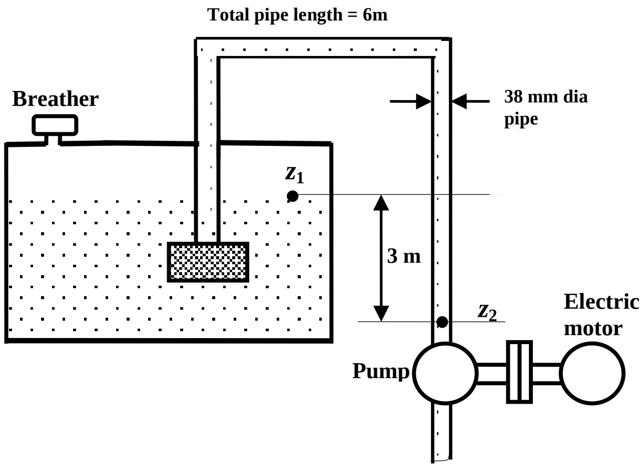


Figure 1.26

Solution: We have p_1 = 68 kPa, $z_1 - z_2$ = 3 m, Q = 0.001896 $\mathrm{m^3/s}$, p_s= 6.9 kPa, SG = 0.9, D_p= 38 mm, v= 100 cS, L_p= 6 m. To calculate p_2.

By the application of Bernoulli's (energy) equation, we can write

$$z_1 + \frac{p_1}{\gamma} + \frac{v_1^2}{2g} + H_{\mathrm{p}} - H_{\mathrm{m}} - H_{\mathrm{L}} = z_2 + \frac{p_2}{\gamma} + \frac{v_2^2}{2g}$$

Now $z_1 - z_2 = 3\,\text{m}$, $H_\text{m} = 0$ (because there is no fluid motor between points 1 and 2), $v_1 = 0$ (assuming the oil tank area to be large).The velocity at point 2, v_2, is

$$v_2 = \frac{Q\ (\text{m}^3/\text{s})}{A\ (\text{m}^2)} = \frac{0.001896}{\pi(0.038^2)/4} = 1.67\,\text{m/s}$$

Equivalent velocity head is

$$\frac{v_2^2}{2g} = \frac{1.67^2}{2 \times 9.81} = 0.142\ \text{m}$$

The head loss is

$$H_\text{L} = \frac{f \cdot L_\text{p}}{D_\text{p}} \times \frac{v^2}{2g}$$

Here

$$L_\text{p} = \text{Total length of pipe} = 6\,\text{m}$$

$$D_\text{p} = \text{Diameter of pipe} = 0.38\,\text{m}$$

Value of *f* (friction factor) depends on the value of Reynolds number.

$$\text{Re} = \frac{vD_\text{p}}{\nu} = \frac{1.67 \times 0.038}{100 \times 10^{-6}}$$
$$= 634.6 < 2000, \text{flow is laminar}$$

Now

$$f = \frac{64}{\text{Re}} = \frac{64}{634.6} = 0.1$$

So head loss due to friction is

$$H_\text{L} = \frac{0.1 \times 6}{0.038} \times 0.142 = 2.24\,\text{m}$$

Case 1: Point 2 is before the pump

When point 2 is before the pump, the pump head is zero, that is, $H_\text{p} = 0$. Rearranging the energy equation to solve for the present head, we can write

$$\frac{p_2}{\gamma} = (z_1 - z_2) + \frac{p_1}{\gamma} + H_\text{p} - H_\text{L} - \frac{v_2^2}{2g}$$
$$= 3 + \frac{68000}{0.9 \times 9800} + 0 - 2.24 - 0.142 = 8.33\ \text{m}$$

$$\Rightarrow p_2 = 8.33 \times 0.9 \times 9800 = 73470\ \text{Pa} = 73.5\ \text{kPa}$$

This valve ofp_2 is without considering the pressure drop across the strainer. The pressure drop is 6.9 kPa across the strainer. Therefore, the pressure at point 2 is

$$p_{2-\text{actual}} = 73.5 - 6.9\ \text{kPa} = 66.6\ \text{kPa}$$

Which is less than 1 atmospheric pressure (101 kPa)?

Case 2: Point 2 is after the pump

When point 2 is after the pump, the pump head must be taken into account

$$H_\text{p} = \frac{P(\text{W})}{\gamma\,(\text{N/m}^3) \times Q(\text{m}^3/\text{s})}$$

Now

$$\gamma = \text{SG} \times \gamma_\text{water} = 0.9 \times 9800\,\text{N/m}^3$$

Also

$$\text{Power} = p_1 \times Q$$

$$P = 68\left(\frac{\text{kN}}{\text{m}^3}\right) \times 0.001896\left(\frac{\text{m}^3}{\text{s}}\right) = 0.13 \text{ kW}$$

So

$$H_\text{p} = \frac{0.13 \times 10^3 \text{ W}}{0.9 \times 9800\left(\frac{\text{N}}{\text{m}^3}\right) \times 0.001896\left(\frac{\text{m}^3}{\text{s}}\right)} = 7.7 \text{ m}$$

Rearranging the energy equation to solve for the present head, we can write

$$\begin{aligned}\frac{p_2}{\gamma} &= (z_1 - z_2) + \frac{p_1}{\gamma} + H_\text{p} - \frac{v_2^2}{2g} \\ &= 3 + \frac{68000}{0.9 \times 9800} + 7.7 - 2.24 - 0.142 \\ &= 16.028 \text{ m}\end{aligned}$$

This valve of p_2 is without considering the pressure drop across the strainer. The pressure drop is 6.9 kPa across the strainer. Therefore, the pressure at point 2 is

$$p_{2\text{-actual}} = p_2 - 6.9 \text{ kPa} = 141.4 - 6 = 134.5 \text{ kPa}$$

which isgreater than 1 atmospheric pressure (101 kPa).

Example 1.21

The volume flow rate through the pump shown in Fig.1.27is 7.8 m^3/s. The fluid being pumped is oil with specific gravity 0.86. Calculate the energy delivered by the pump to the oil per unit weight of oil flowing in the system. Energy losses in the system are caused by the check valve and friction losses as the fluid flows through the piping. The magnitude of such losses has been determined to be 1.86 N m/N.

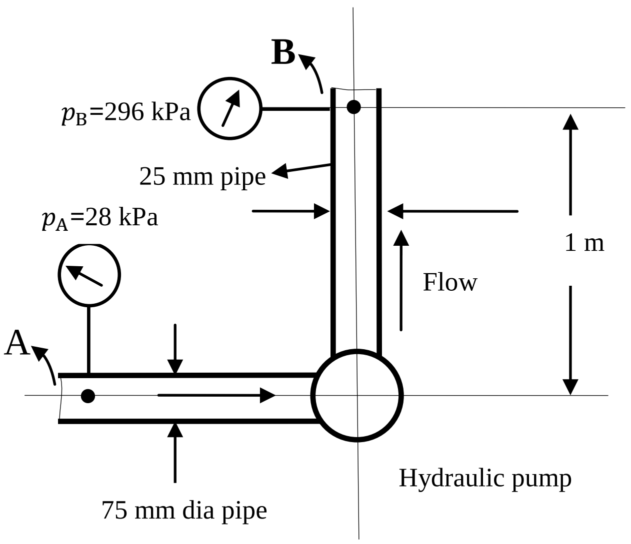


Figure 1.27

Solution: Using the section where pressure gauges are located as the section of interest, write the energy equation for the system, including only the necessary terms:

$$\frac{p_A}{\gamma} + z_A + \frac{v_A^2}{2g} + H_{\text{Added}} - H_{\text{Removed}} - H_{\text{Losses}} = \frac{p_B}{\gamma} + z_B + \frac{v_B^2}{2g}$$

or $$H_{\text{Added}} = \frac{p_B - p_A}{\gamma} + z_B - z_A + \frac{v_B^2 - v_A^2}{2g} + H_{\text{Losses}}$$

In this case, the specific gravity of oil is

$$\gamma = (\text{SG})(\gamma_{\text{water}}) = (0.86)(9.81) = 8.44 \text{ kN/m}^3$$

Since $p_B = 296$ kPa and $p_A = -28$ kPa we get

$$\frac{p_B - p_A}{\gamma} = \frac{[296 - (-28)] \text{ kN}}{\text{m}^2} \times \frac{\text{m}^3}{8.44 \text{ kN}} = 38.4 \text{ m}$$

Now $z_B - z_A = 1$ m as B is at a higher elevation than A.The volume flow rate and continuity equation are used to determine the velocity. Now

$$Q = Av = A_A v_A = A_B v_B$$

$$\Rightarrow v_A = \frac{Q}{A_A} = \left(\frac{0.014 \text{ m}^3}{\text{s}}\right)(4.768 \times 10^{-3})\text{m}^2 = 2.94 \text{ m/s}$$

and $$v_B = \frac{Q}{A_B} = \left(\frac{0.014 \text{ m}^3}{\text{s}}\right)(2.168 \times 10^{-3})\text{m}^2 = 6.46 \text{ m/s}$$

So,

$$\frac{v_B^2 - v_A^2}{2g} = \frac{(6.46^2 - 2.94^2) \text{ m}^2/\text{s}^2}{2\left(\frac{9.81 \text{ m}}{\text{s}^2}\right)} = 1.69 \text{ m}$$

Given $H_{\text{Losses}} = 1.86$ m. Therefore,

$$H_{\text{Added}} = 38.4 \text{ m} + 1.0 \text{ m} + 1.69 \text{ m} + 1.86 \text{ m} = 42.9 \text{ m} \text{ or } 42.9 \text{ N m/N}$$

That is, the pump delivers 42.9 N m of energy to each newton of oil flowing through it.

Example 1.22

For the hydraulic system of Fig.1.28, the following data are given:

1. Pump flow is 0.001896 m^3/s.
2. The air pressure at station 1 in the hydraulic tank is 68.97 kPa gauge pressure.
3. The inlet line to the pump is 3.048 m below the oil level.
4. The pipe has an inside diameter of 0.0381 m.

Find the pressure at station 2 if

(a) There is no head loss between stations 1 and 2.

(b) There is 7.622 m head loss between stations 1 and 2.

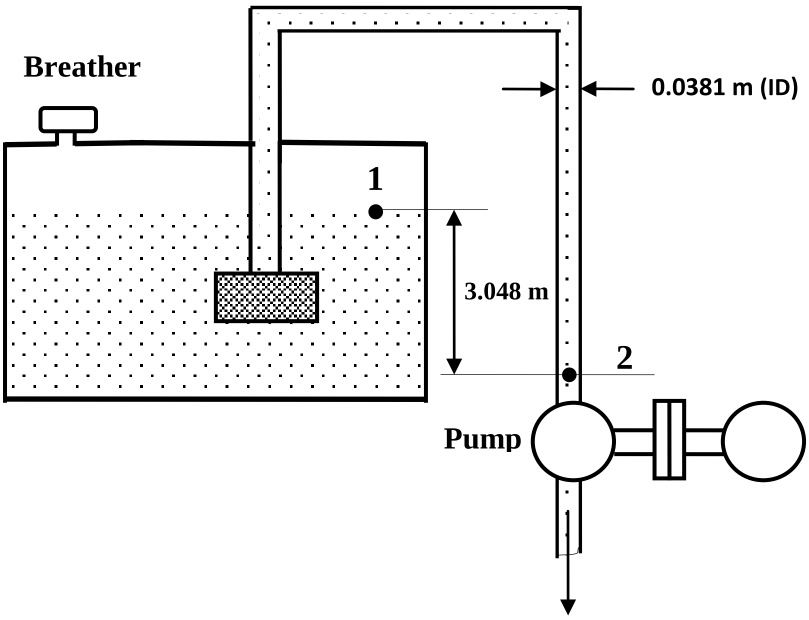


Figure 1.28

Solution: We use Bernoulli's equation

$$z_1+\frac{p_1}{\gamma}+\frac{v_1^2}{2g}+H_\mathrm{p}-H_\mathrm{m}-H_\mathrm{L}=z_2+\frac{p_2}{\gamma}+\frac{v_2^2}{2g}$$

Case (a): There is no head loss between stations 1 and 2

Here $z_1-z_2=3.0489\text{ m},\ H_\mathrm{L}=0, p_1$ =68.97 kPa. Now

$$v_2=\frac{\text{Flow}}{\text{Area}}=\frac{0.001896}{\frac{\pi}{4}(0.0381^2)}=1.66\text{ m/s}$$

Since there is no pump between 1 and 2, $H_\mathrm{p}=0.$

Since there is no motor between 1 and 2, $H_\mathrm{m}=0.$

Assume v_1 = 0 (assuming that area of cross-section is large).
Simplification gives for no head loss,

$$\frac{p_1}{\rho g}-\frac{p_2}{\rho g}=z_2-z_1+\frac{v_2^2}{2g}$$

Assuming $\rho g=8817\text{ N/m}^3$ we get

$$\frac{68970\text{ N/m}^2}{8817\text{ N/m}^3}+0+z_1+0-0-0=\frac{p_2}{\rho g}+\frac{(1.66\text{ m/s})^2}{2\times9.81\text{ m/s}^2}+z_2$$

Knowing that $z_1-z_2=3.048\text{ m}$ we get

$$\frac{p_2}{\rho g}=3.048+\frac{68970}{8817}-\frac{(1.66)^2}{2\times9.81}=3.048+7.82-0.142=10.73\text{ m}$$

$$p_2=10.73\,(\text{m})\times8817\,(\text{N/m}^3)=94610\,\text{Pa}$$
$$=94.6\,\text{kPa}$$

Case (b): There is 7.622 m head loss between stations 1 and 2

$$\frac{p_2}{\rho g} = 10.73\,\text{m} - 7.622 = 3.11\,\text{m}$$

So

$$p_2 = 3.11\,(\text{m}) \times 8817\,(\text{N/m}^3) = 27400\,\text{Pa}$$
$$= 27.4\,\text{kPa}$$

Example 1.23

For the pump in Fig.1.29, Q_{out} =0.00190 m^3/s of oil having a specific gravity of 0.9. What is Q_{in}? Find the pressure difference between A and B if

(a) The pump is turned OFF.

(b) The input power to the pump is 1494 W.

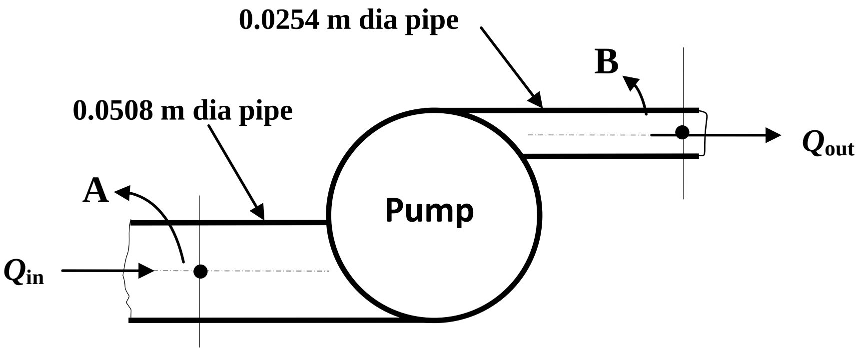


Figure 1.29

Solution:

(a) The pump is turned OFF:

As per Bernoulli's equation, $p_{\text{B}} - p_{\text{A}} = 0$

(b) The input power to the pump is 1494 W

We use Bernoulli's equation:

$$\frac{p_{\text{A}}}{\rho g} + \frac{v_{\text{A}}^2}{2g} + z_{\text{A}} + H_{\text{p}} - H_{\text{m}} - H_{\text{L}} = \frac{p_{\text{B}}}{\rho g} + \frac{v_{\text{B}}^2}{2g} + z_{\text{B}}$$

Here

$$H_{\text{p}} = \frac{\text{Pump power (W)}}{\gamma\,(\text{N/m}^3) \times Q(\text{m}^3/\text{s})}$$
$$= \frac{1494}{0.9 \times 9800 \times 0.00190} = 89.2\text{ m}$$

$$v_{\text{A}} = \frac{\text{Flow}}{\text{Area}} = \frac{0.00190}{\frac{\pi}{4}(0.0508^2)} = 0.937\,\text{m/s}$$

$$v_{\mathrm{B}} = \frac{\text{Flow}}{\text{Area}} = \frac{0.00190}{\frac{\pi}{4}(0.0254^2)} = 3.75 \text{ m/s}$$

Substituting values, we have

$$\frac{(p_{\mathrm{A}} - p_{\mathrm{B}})}{\gamma} = H_{\mathrm{p}} - \frac{(v_{\mathrm{B}}^2 - v_{\mathrm{A}}^2)}{2g}$$

$$= 89.2 - \frac{(3.75^2 - 0.937^2)}{2 \times 9.81}$$

$$= 88.5 \text{ m}$$

So

$$p_{\mathrm{B}} = 88.5\,(\mathrm{m}) \times 9800\,(\mathrm{N/m^3}) \times 0.9$$

$$= 781000\,\mathrm{Pa}$$

$$= 781\mathrm{kPa}$$

1.10 Torricelli's Theorem

Torricelli's theorem is Bernoulli's equation with certain assumptions made. Torricelli's theorem states that the velocity of the water jet of liquid is directly proportional to the square root of the head of the liquid producing it. This deals with the setup where there is a large tank with a narrow opening allowing the liquid to flow out (Fig. 1.30). Both the tank and the narrow opening (nozzle) are open to the atmosphere:

$$z_1 + \frac{p_1}{\gamma} + \frac{v_1^2}{2g} + H_{\mathrm{p}} - H_{\mathrm{m}} - H_{\mathrm{L}} = z_2 + \frac{p_2}{\gamma} + \frac{v_2^2}{2g}$$

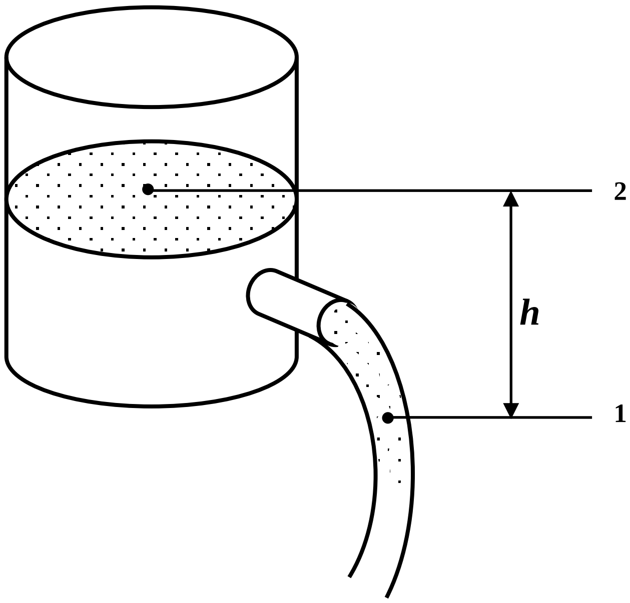


Figure 1.30 Tank with a narrow opening (nozzle)

In this setup, certain assumptions are made:

1. Pressure is the same because the tank and the nozzle are open to the atmosphere, that is, $p_1 = p_2$.

2. Also, let $z_2 - z_1 = h$.

3. The fluid velocity of the tank (water level) is very much slower than the fluid velocity of the nozzle as the area of the liquid surface is much larger than that of the cross section of nozzle, that is, $v_2 \ll v_1$.

4. There is no pump or motor, that is, $H_p = H_m = 0$.

5. There are no frictional losses, that is, $H_L = 0$.

Keeping all these assumptions in mind, Bernoulli's equation gets reduced to

$$v_2 = \sqrt{2gh}$$

where v_2 is the jet velocity (m/s), g is the acceleration due to gravity (m/s^2) and h is the pressure head (m). Now if we do not consider an ideal fluid, then the friction head will be present (H_L). In that case

$$v_2 = \sqrt{2g(h - H_L)}$$

This shows that the velocity of jet decreases if the friction losses are taken into account.

1.11 Siphon

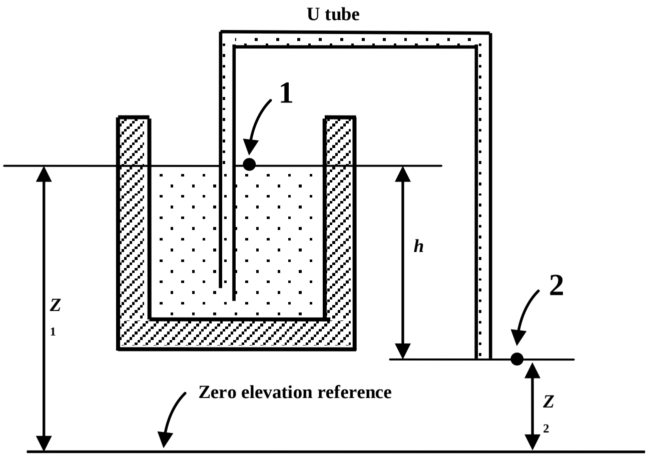


Figure 1.31 The siphon principle

A siphon is a familiar hydraulic device (Fig. 1.31). It is commonly used to cause a liquid to flow from one container in an upward direction over an obstacle to a second lower container in a downward direction. As shown in Fig. 1.31, a siphon consists of a U-tube with one end submerged below the level of the liquid surface, and the free end lying below it on the outside of the container. For the fluid to flow out of the free end, two conditions must be met:

1. The elevation of the free end must be lower than the elevation of the liquid surface inside the container.
2. The fluid must initially be forced to flow up from the container into the center portion of the U-tube. This is normally done by temporarily providing a suction pressure at the free end of the siphon. For example, when siphoning gasoline from an automobile gas tank, a person can develop this suction by momentarily sucking the free end of the hose. This allows atmospheric pressure in the tank to push the gasoline up the U-tube hose, as required. For continuous flow operation, the free end of the U-tube hose must lie below the gasoline level in the tank.

We can analyze the flow through a siphon by applying the energy equation between points 1 and 2 as shown in Fig. 1.31:

$$\frac{p_1}{\gamma}+\frac{v_1^2}{2g}+z_1+H_\mathrm{p}-H_\mathrm{m}-H_\mathrm{L}=\frac{p_2}{\gamma}+\frac{v_2^2}{2g}+z_2$$

The following conditions apply for a siphon:

1. $p_1 = p_2$ = atmospheric pressure.
2. The area of the surface of the liquid in the container is large so that the velocity v_1 equals essentially 0.

Example 1.24
For the siphon system shown in Fig.1.32, the following data are given: z_1 = 4 m, z_2 = 0.2 m, H_L = 0.5 m. If the inside diameter of the siphon pipe is 30 mm, determine the velocity of the fluid and the flow rate (in LPM) through the siphon. Apply the energy equation and solve the problem.

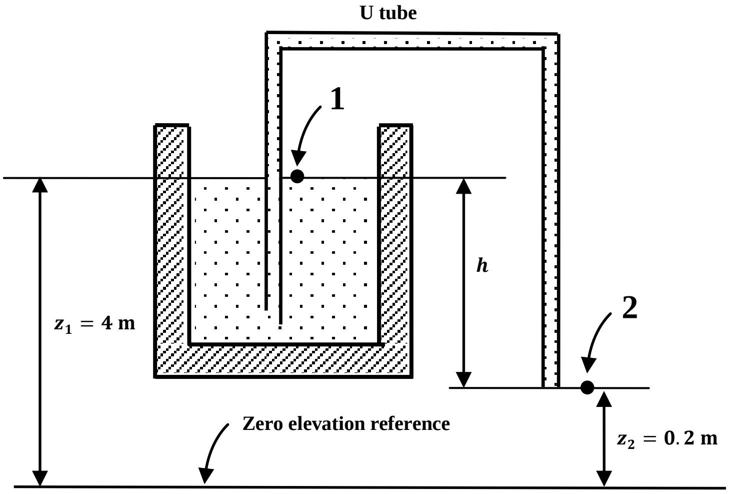


Figure 1.32

Solution: Given z_1 =4 m, z_2= 0.2 m, H_L = 0.5m, D = 30 mm = 30×10^{-3} m. To calculate v_2 and Q_2.This problem can be solved by using the energy equation (modified Bernoulli's theorem) to points (1) and (2) as below:

$$z_1+\frac{p_1}{\gamma}+\frac{v_1^2}{2g}+H_\mathrm{p}-H_\mathrm{m}-H_\mathrm{L}=z_2+\frac{p_2}{\gamma}+\frac{v_2^2}{2g}$$

where $p_1 = p_2$ = 0 (atmospheric pressure), v_1 = 0 (as the tank is quite large, the velocity is negligible), H_p = 0 (no pump), H_m=0 (no motor), $z_1 - z_2 = h$ (the head). Substituting these values in the above equation we obtain

$$h+0+0+0-0-H_\mathrm{L}=0+\frac{v_2^2}{2g}$$

$$\Rightarrow h-H_\mathrm{L}=\frac{v_2^2}{2g}$$

$$\Rightarrow v_2^2=2g(h-H_\mathrm{L})$$

$$\Rightarrow v_2=\sqrt{2g(h-H_\mathrm{L})}\;=\sqrt{2\times9.81(3.8-0.5)}=8.05\,\mathrm{m/s}$$

The flow rate is given by

$$Q_2 = A_2 v_2 = \frac{\pi}{4}(30\times10^{-3})^2 \times 8.05$$

$$\Rightarrow Q_2 = 5.7\times10^{-3}\ \text{m}^3/\text{s}$$

$$\Rightarrow Q_2 = 5.7\times10^{-3}\times10^{3}\times60$$

$$\Rightarrow Q_2 = 342\ \text{LPM}$$

Example 1.25

A siphon is made of a pipe whose inside diameter is 25.4 mm and is used to maintain a constant level in a 6.0975 m deep tank (Fig. 1.33). If the siphon discharge is 9.144 m below the top of tank, what will be the flow rate if the fluid level is 1.524 m below the top of tank?

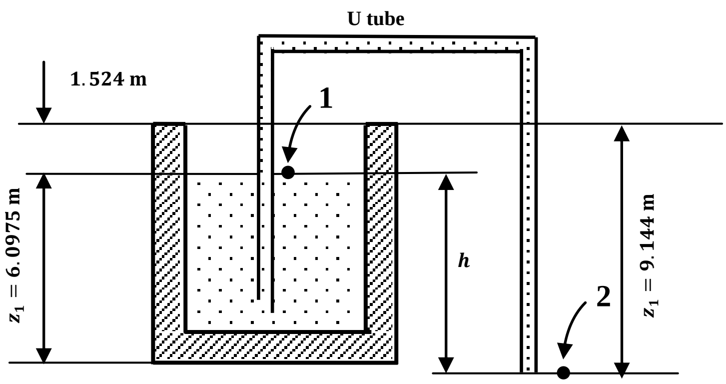


Figure 1.33

Solution: From Fig.1.33, h = (9.144−1.524) = 7.62 m. From the previous problem, we can write using the modified Bernoulli's theorem

$$v_2^2 = 2gh$$

$$\Rightarrow v_2 = \sqrt{2gh} = \sqrt{2\times9.81\times7.62} = 12.2\ \text{m/s}$$

Now

$$Q = v_2 \times A_2 = 12.2\times\left(\frac{\pi}{4}0.0254^2\right)$$

$$= 0.00618\,\text{m}^3/\text{s} = 6.18\ \text{LPS}$$

Objective-Type Questions

Fill in the Blanks

1. Pascal's law states that the pressure exerted on a _______ is transmitted undiminished in _______ and acts with equal force on equal areas and at _______to the surface of the container.

2. The total energy includes potential energy due to elevation and pressure and also _______.

3. The _______ energy of a particle of a fluid depends on its elevation above any arbitrary plane.

4. Pressure energy is possessed by the fluid per unit _______ of fluid virtue of the pressure under which the fluid exists.

5. Torricelli's theorem states that the velocity of the water jet of liquid is _______ proportional to the _______ of the head of the liquid producing it.

State True or False

1. Continuity equation states that the weight flow rate is the same for all cross sections of a pipe.

2. Hydraulic power is equal to the product of pressure and volume flow rate.

3. A pump converts mechanical energy into hydraulic energy by increasing the fluid flow.

4. It is easy to achieve overload protection using hydraulic systems.

5. Hydraulic power in kW $= \dfrac{\text{Flow (LPM)} \times \text{Pressure (bar)}}{320}$.

Review Questions

1. Define hydraulic power. Derive an expression for hydraulic power if the flow is in LPS and pressure in kPa.
2. How will you explain Pascal's law with reference to working of a hydraulic cylinder?
3. State Bernoulli's theorem.
4. What is a continuity equation and what are its implications relative to fluid power?
5. What is the significance of each term in the energy equation?
6. Define pressure head, elevation head and kinetic head.
7. State Torricelli's theorem and mention its significance.
8. Explain how a siphon operates.
9. State Pascal's law.
10. Explain the meaning of Bernoulli's equation and how it affects the flow of a fluid in a hydraulic circuit.
11. Relative to power, there is an analogy among mechanical, electrical and hydraulic systems. Describe this analogy.

12. What is the significance of each term in the energy equation?
13. State the basic principle laws and equations of hydraulics.

Answers

Fill in the Blanks

1. Confined fluid, all directions,right angles
2. Kinetic energy due to velocity
3. Potential
4. Weight
5. Directly, square root

State True or False

1. True
2. True
3. False
4. True
5. False

Lecture 5

DISTRIBUTION OF FLUID POWER

Learning Objectives

Upon completion of this chapter, the student should be able to:

- Write the points to be considered for the selection of hydraulic conductors.
- Differentiatebetween burst pressure and working pressure.
- List commontypes of fittings used in fluid power.
- List common types of screws used in fluid power.
- Explain varioustypes of joints used in fluid power.
- Write guidelinesfor the selection of plastic conductors.
- Explain varioustypes of hoses used in fluid power.
- Explain theuse of rotary joints and quick couplings.

1.1 Introduction

The size of pipe, tubing or hose for plumbing fluid power systems is very important. If the size with too small an inside cross-sectional area is used, the oil is forced to flow at a high rate of speed, and this creates excessive power loss and heat generation in the oil. If the size used is larger than necessary, then the power transfer is good and heat generation is low but the time and cost of installation are more than they should be. Pressure losses are present only when fluid is moving. Force can be transmitted from one end of fluid column to other with virtually no loss but when the fluid starts to move, which is necessary to transmit work or power, then the losses start. This chapter deals with selection of distribution systems, losses in fluid power transmission lines and its effect on the performance of fluid system.

1.2 Choice of Distribution

A fluid distribution system is composed of pipes, tubings, hose assemblies, manifolds and fittings so arranged that the fluid is carried with minimum losses from the reservoir through controls and working components and is then returned. All materials used to convey fluid power are commonly classified as *conductors* and the various fittings for connecting components are classified as *connectors*. More than one type of conductors may be used in the same installations. Conductors are generally steel regardless of conductor material and they may be coated with cadmium or some other corrosion-resistant material. Stainless steel conductors or fittings may be used if extremely corrosive environments are anticipated, but the high cost of such conductors and fitting precludes the general use of this material. Copper can never be used in a hydraulic system because it catalyzes the oxidation of petroleum fluids, while zinc, magnesium and cadmium cannot be used because they are rapidly corroded by glycol fluids. A galvanized pipe is unsatisfactory because galvanization tends to flake off into the system.

The choice of pipe, tube or hose depends on the operating pressures of system and flow. Other important factors include environmental conditions, type of fluid and operating temperature, shock loads, relative motion between connected parts, practicality and compliance with certain standards. The material used must have a continuous operating pressure rating so that it can withstand working pressures and provide a factor of safety for short-lived pressure peaks resulting from a hydraulic shock. Hydraulic shocks occur due to sudden stopping or reversing a flow that is backed by large flow forces, sudden deceleration, stopping or reversing of heavy workloads.

The following points must be considered both while designing the system and selecting a conductor:

- The working strength of conductor must be sufficient to contain fluid under all normal operating conditions, and there must be sufficient reserve strength to withstand shock loads due to system operations.
- The mechanical strength of the conductor must be sufficient to span the distance required by the machine configurations and to withstand mechanical vibrations that may be encountered.
- The interior surface of the conductor must be as smooth as possible to minimize friction.
- The pipe size must be adequate to permit design flow at a reasonable fluid velocity.
- Conductors should be positioned so that they cannot be damaged by normal operations at and around the machine.
- Conductors should be supported in such a way that vibrations and shocks to them are minimized.
- Conductor runs may frequently be reduced by using manifolds or by using machine structural components as conductors.

1.3 Conductor Sizing

The pressure rating of various conductors depends on the tensile strength of the material used and the wall thickness of the conductor. The wall thickness and safety factors recommended by the fluid power industry standards are based on calculations using Barlow's formula

$$\text{Minimum wall thickness} = \frac{\text{Maximum wall thickness} \times \text{OD of conductor}}{2 \times \text{Tensile strength}(\sigma)}$$

The formula is adequate for practical purposes in selecting conductor wall thickness to withstand the maximum rate of surge peak pressures at frequencies developed by cycling of the hydraulic equipment operation. Surge pressures that may be encountered within a system are rarely known and seldom appreciated in their full potential strength.

In addition to normal working pressure and surge pressure peaks, there may be mechanical stresses produced by thermal expansion, abuse and environmental factors. Evaluation of all phenomena is difficult, so a higher factor of safety is used. Table 1.1 shows the typical specification of a hydraulic pipe.

Table 1.1 Typical specification of a hydraulic pipe

Standard followed	DIN 2391
Outer diameter	0.5–12 inch
Thickness	0.039-0.472 inch
Steel grade	16Mn (DIN 2391 ST 35.8 / 37.4)
Composition	C(0.13), Si(0.37),Mn(0.65), P(0.035), Cr(0.3)max, Ni(0.25), Cu(0.2)
Manufactured	Cold drawn
Surface treatment	Copper coated

1.4 Burst Pressure and Working Pressure

When a valve is closed suddenly, high surge pressure can burst pipe lines. The burst pressure is the pressure of the fluid that causes the pipe to burst. This occurs when the tensile stress developed due to pressure (σ) equals the tensile strength (S) of the pipe material:

$$t = \frac{p \times d}{2 \times \sigma}$$

We can rewrite this expression in terms of the tensile strength of material:

$$t = \frac{p \times d}{2 \times S}$$

If p_{BP} is the burst pressure then we can write

$$p_{\mathrm{BP}} = \frac{2t\,S}{d}$$

The working pressure (WP) is the maximum safe operating fluid pressure and is defined as the burst pressure divided by an appropriate factor of safety (FOS):

$$\text{Working pressure} = \frac{\text{Maximum (Burst) pressure}}{\text{Factor of safety}} = \frac{p_{\mathrm{BP}}}{\mathrm{FOS}}$$

A factor of safety ensures the integrity of the conductor by determining the maximum safe level of working pressure. Industry standards recommend the following factors of safety based on corresponding operating pressures:

- FOS = 8 for pressures from 0 to 70 bar
- FOS = 6 for pressures from 70 to 180 bar
- FOS = 4 for pressures above 180 bar

If a fluid system is subjected to high-pressure shocks, then an FOS of 10 is used.

Example 1.1

A pump produces a flow rate of 75 LPM. It has been established that the fluid velocity in a discharge line should be between 6 and 7.5 m/s. Determine the minimum and maximum pipe inside diameter that should be used.

Solution:

Flow rate = Discharge= 75 LPM = 0.075/60=1.25×10^{-3} $\mathrm{m^3/s}$

Now

Discharge = Area × Velocity

$$A_{\min} = \frac{\pi}{4}(d^2) = \frac{\text{Discharge}}{\text{Velocity(maximum)}} = \frac{1.25\times10^{-3}}{7.5} = 0.0001667\ \mathrm{m}^2$$

Solving we get $d = 14.5\ \mathrm{mm}$ (for the maximum velocity). So

$$A_{\max} = \frac{\pi}{4}(d^2) = \frac{\text{Discharge}}{\text{Velocity (minimum)}} = \frac{1.25\times10^{-3}}{6} = 0.0002804\ \mathrm{m}^2$$

Solving, we obtain $d = 16.3\ \mathrm{mm}$ (for the minimum velocity).

Example 1.2

A steel tubing has an outside diameter of 30 mm and an inside diameter of 24 mm. It is made up of commercial steel of the tensile strength of 520 MPa. What is the safe working pressure? Assuming that tubing is subjected to a high-pressure shock, determine the tensile stress for an operating pressure of 10 MPa.

Solution:

$$\text{Bursting pressure} = p_{\text{BP}} = \frac{2\,t\,S}{d} = \frac{2\times 3\times 517}{24} = 129.3\ \text{MPa}$$

Since the tubing is subjected to a high-pressure shock, we can take factor of safety as 10:

$$\text{Working pressure} = \frac{\text{Maximum pressure}}{\text{Factor of safety}} = \frac{129.3}{10} = 12.93\,\text{MPa} = 129.3\,\text{bar}$$

The tensile stress is

$$\sigma = \frac{p\times d}{2\times t} = \frac{10\times 0.024}{2\times 0.003} = 40\,\text{MPa}$$

1.5 Steel Pipes

Steel pipes are still extensively used in fluid power systems, although they are rapidly being supplemented by steel or plastic tubing. The major disadvantages of steel pipes are their weight and the large number of fitting requirement for connection . Its greatest advantage is its mechanical strength and particularly its ability to withstand abuse. Steel pipes are sized according to the nominal diameter that is neither the outside nor the inside diameter, while the wall thickness is specified by a schedule number. Most of the industries seldom use metric designation while designing and buying pipes. The prime considerations for selecting conductors for a hydraulic power system are the type of materials, capacity and pressure rating. Piping has originally been classified by weight as standard, extra heavy and double extra heavy. This classification has been superseded by classification according to schedule numbers. A hot- or cold-drawn seamless pipe is recommended for use in a hydraulic system and must be internally free from rust scale and dirt.
Schedule numbers run from 40 (earlier standard) to 80 (earlier extra duty) and to 160 (earlier double extra duty). Table 1.2 shows the usage of schedule pipes under fluid power standards. Industry hydraulic standard recommends a 4:1 factor of safety for systems operating above 180–200 bar. Steel pipe fittings are most often fabricated from malleable iron that has a sufficient strength and ductility to withstand forces encountered in a fluid power system.

Table 1.2 The usage of schedule pipes under fluid power standards

Normal Pipe Size (in.)	Pipe Outside Diameter (in.)	Schedule 40		Schedule 80		Schedule 160	
		ID (in.)	WP (psi)	ID (in.)	WP (psi)	ID (in.)	WP (psi)
1/8	0.405	0.269	590	0.215	2100	–	–
1/4	0.540	0.364	1250	0.302	2620	–	–
3/8	0.675	0.493	1090	0.423	2300	–	–
1/2	0.840	0.622	1350	0.546	2420	0.466	7380
3/4	1.050	0.824	1160	0.742	2080	0.614	6000
1	1.315	1.049	1260	0.957	2070	0.815	5720
1-1/4	1.660	1.380	850	1.278	1550	1.160	4470
1-1/2	1.900	1.610	800	1.500	1450	1.338	4100
2	2.375	2.067	720	1.939	1330	1.689	3580
21/2	2.875	2.469	970	2.323	1550	2.125	3940
3	3.500	3.068	875	2.900	1400	2.624	3520

It is to be noted that for a given nominal size of pipe, the wall thickness increases as the schedule number increases. Table 1.4 shows the usage of metric pipes. These metric designations are not common in industry.

Table 1.4 The usage of metric pipes under fluid power standards

Tube OD (mm)	**Tube ID (mm)**	**Wall Thickness (mm)**	**Tube OD (mm)**	**Tube ID (mm)**	**Wall Thickness (mm)**	**Tube OD (mm)**	**Tube ID (mm)**	**Wall Thickness (mm)**
4	3	0.5	14	10	2.0	25	19	3.0
6	4	1.0	15	12	1.5	25	17	4.0
6	3	1.5	15	11	2.0	28	24	2.0
8	6	1.0	16	12	2.0	28	23	2.5
8	5	1.5	16	10	3.0	30	24	3.0
8	4	2.0	18	15	1.5	30	22	4.0
10	8	1.0	20	16	2.0	35	31	2.0
10	7	1.5	20	15	2.5	35	29	3.0
10	6	2.0	20	14	3.0	38	30	4.0
12	10	1.0	22	20	1.0	38	28	5.0
12	9	1.5	22	19	1.5	42	38	2.0
12	8	2.0	22	18	2.0	42	36	3.0

Example 1.3

A steel tube of inner diameter 25 mm has a burst pressure of 50 MPa. If the tensile strength is 380 MPa, find the minimum acceptable OD.

Solution: The bursting pressure is given by

$$p_{\mathrm{BP}} = \frac{2\,t\,S}{d}$$

$$\Rightarrow 50 = \frac{2 \times t \times 380}{0.025}$$

Solving we obtain $t = 1.644 \times 10^{-3}$ m = 1.644 mm. Now

$$\text{OD} = \text{ID} + 2t = 25 + 2 \times 1.644 = 28.3 \text{ mm}$$

Example 1.4

What is the minimum size of commercial pipe tubing with a wall thickness of 2 mm required at the inlet and outlet of a 75 LPM pump? The inlet and outlet velocities are limited to 1.2 and 6.1 m/s, respectively.

Solution: We have

$$75 \text{ LPM} = 0.075/60 = 1.25 \times 10^{-3} \text{ m}^3/\text{s}$$

We know that discharge

$$Q = \text{Area} \times \text{Velocity}$$

$$A_{\min} = \frac{\pi}{4}(d^2) = \frac{\text{Discharge}}{\text{Velocity(max)}} = \frac{1.25 \times 10^{-3}}{6.1} = 0.001042 \text{ m}^2$$

Solving, we obtain $d = 36.5$ mm (for the maximum velocity). Referring to Table 1.4 we select 42(OD) $\times$ 36(ID) for the pump inlet.

Again

$$A_{\max} = \frac{\pi}{4}(d^2) = \frac{\text{Discharge}}{\text{Velocity (min)}} = \frac{1.25 \times 10^{-3}}{1.2} = 0.0002049 \text{ m}^2$$

Solving we obtain $d = 16.2$ mm (for the minimum velocity). Referring to Table1.4, we select 22(OD) $\times$ 18(ID) for the pump outlet.

Example 1.5

What is the minimum size of commercial pipe tubing with a wall thickness of 2.4 mm required at the inlet and outlet of a 0.00189 m^3/s pump? The inlet and outlet velocities are limited to 1.92 m/s and 6.08 m/s, respectively.

Solution: This problem has to be solved by the trial-and-error method.

Pump Inlet:

(a) Using Table 1.4, let us select t = 2.5 mm, OD = 28 mm and ID = 23 mm. Now

$$\text{Velocity} = \frac{\text{Discharge}}{\frac{\pi}{4}(d^2)} = \frac{0.00189}{\frac{\pi}{4}(0.023^2)} = 4.54 \text{ m/s}$$

(b) Let us select t = 3 mm, OD = 25 mm and ID = 19 mm. Now

$$\text{Velocity} = \frac{\text{Discharge}}{\frac{\pi}{4}(d^2)} = \frac{0.00189}{\frac{\pi}{4}(0.019^2)} = 6.67 \text{ m/s}$$

(c) Let us select t = 3 mm, OD = 42 mm and ID = 36 mm. Now

$$\text{Velocity} = \frac{\text{Discharge}}{\frac{\pi}{4}(d^2)} = \frac{0.00189}{\frac{\pi}{4}(0.036^2)} = 1.86 \text{ m/s}$$

Referring to Table 1.4,we can use a 28 × 23 × 2.5 mm^3 pipe for the pump outlet and a 42 × 36 × 3 mm^3 pipe for the pump inlet.

Example 1.6

The flow rate of certain fluid in a pipe is 0.001 m^3/s and an operating pressure is 70 bar. The maximum recommended velocity is 6.1 m/s and the factor of safety of 8 is allowed. Select a metric steel tube when

(a) Material is SAE 1010 with a tensile strength of 380 MPa.

(b)Material is AISI 4130 with a tensile strength of 570 MPa.

Solution:

(a) Selection of Pipe for the SAE 1010

$$\text{Flow area} = \frac{\pi}{4}(d^2) = \frac{\text{Discharge}}{\text{Velocity(max)}} = \frac{0.001}{6.1} = 0.0001639 \text{ m}^2$$

Solving we get inside diameter $(d) = 14.4 \text{mm}$.

From Table 1.4 we select t = 1.5 mm, ID = 12 mm and OD = 15mm. So bursting pressure is

$$p_{\text{BP}} = \frac{2\ t\ S}{d} = \frac{2 \times 0.0015 \times 380}{0.012} = 94.9 \text{ MPa}$$

The working pressure is

$$\text{Working pressure} = \frac{\text{Maximum pressure}}{\text{Factor of safety}} = \frac{94.9}{8} = 11.93 \text{MPa} = 119.3 \text{bar}$$

Working pressure is greater than the operating pressure of 70 bar. So we use a pipe of 15 × 12 × 1.5 mm.

(b) Selection of Pipe for the AISI 4130

$$p_{\text{BP}} = \frac{2\ t\ S}{d} = \frac{2 \times 0.0015 \times 517}{0.012} = 129.3 \text{ MPa}$$

The working pressure is

$$\text{Working pressure} = \frac{\text{Maximum pressure}}{\text{Factor of safety}} = \frac{94.9}{8} = 16.2\,\text{MPa} = 162\ \text{bar}$$

Working pressure is greater than the operating pressure of 70 bar. So we use a pipe of 15 $\times$ 12 $\times$ 1.5 mm.

1.6 Screwed Connections

Steel piping in fluid power systems is most often joined by threaded connections. Unfortunately, threading weakens the pipe thereby making it necessary to use heavier walls than would otherwise be required. This difficulty can be overcome by welding, but welded sections are not desirable in fluid power systems that require frequent disassembly. Large-diameter piping systems generally are fabricated with flanged joints.

Taper threads form a seal by an interference fit between a male and a female component when they are tightened together, and some form of jointing compound or flexible plastic tape is added to ensure a good joint. Great care must be taken when screwing taper threads into the body of a component, particularly if it is made of cast iron, otherwise the casting may be cracked.

Parallel threads are easier to manufacture and simpler to use. Joints made with parallel threads must have a sealing washer between the component body and a suitable shoulder on the pipe fitting in order to prevent fluid leakage [Fig. 1.1(c)]. Parallel thread fittings should never be used in taper thread holes or vice versa.

The most commonly used screw thread forms for hydraulic pipe fittings are as follows:

1. British standard pipe threads (BSP).
2. American National pipe threads (NPT).
3. Unified pipe threads (UNF).
4. Metric pipe threads.

1.7 Steel Tubing

Seamless steel tubing is the most widely used material for hydraulic system conductors.One major reason of its popularity is the fact that it can be easily formed to fit irregular paths so that fewer fittings are required. The obvious result is a considerably lessened chance of leakage since every connection is a potential leak point. It is also relatively small and light, thus making it easy to use.

Rigid steel tubing is either drawn or seam welded, the later sometimes used in hydraulic systems for pipe work but is generally unsuitable for higher pressure and is more difficult to manipulate as the seam tends to split when the pipe is bent.

When bending steel tubing, it is always important to use proper tube-bending equipment with fixtures of the correct size, otherwise the pipe is flattened. This reduces its cross-sectional area and causes a higher resistance to the flow of fluid. It is usual to specify hydraulic tubing by reference to the outside diameter (OD) and wall thickness. A range of standard tubes are available in both inch and metric sizes from about 5 mm OD. For most fluid power applications, the tubing used is SAE 1010 dead soft cold-drawn steel tubing. This material is easily worked and has strength equal to or greater than the schedule 80 pipe. If a greater strength is required, a similar tubing fabricated from AISI 4230 steel can be used that can withstand approximately 50% more working pressure.
Various standard wall thicknesses are available for each size of tubing, and tube manufacturers supply tables indicating the safe working pressure for each size. A minimum safety factor of 4:1

should be used when selecting the wall thickness of a tube. It means the bursting pressure of the tube must be four times the maximum fluid pressure. Obviously, thicker tubes are more difficult to manipulate, particularly with a larger diameter.
Flow rates through smooth bore tubes should normally not exceed 5m/s in pressure lines or 1.2 m/s in suction lines. Tubes used for hydraulic systems must be clean and free from rust; otherwise particles of grit may find their way into the precision equipment, causing serious damage to pumps and valves. Tubes in transit or storage should always have their open ends capped to prevent the ingress of dirt and moisture. Tubes are attached to end fittings using compression joint and flared tubes. Very large fittings are usually welded.

1.8 Compression Joints

Compression-type fittings comprise a loose ring having a cone-shaped nose that must face the open end of a tube, a mating tapered barrel and a retaining nut. The end of the tube must always be cut square and deburred before assembly. When the tube is pushed fully in the fitting and the retaining nut is tightened, the compressive action forces the nose of the ring into the surface of the metal tube, creating a permanent and very strong interference fit that is capable of withstanding pressure in excess of 350 bar.

1.8 Plastic Conductors

Plastic tubing is now available in polyethylene, polypropylene, polyvinyl chloride and nylon. Each material has specific characteristics that make it more suitable for some services than for others. The best procedure is to check the manufacturer's literature against the service conditions whenever plastic tubing is being considered.. In general, plastic tubing is most often used in pneumatic systems, primarily because it does not have sufficient strength to be used in most hydraulic systems. The plastics are compatible with most hydraulic fluids, however, and could safely be used in low-pressure applications.
Plastic tubing has gained rapid acceptance in the industry because it is inexpensive and extremely easy to use. It can easily be formed to fit around obstructions without special tools; it is light and easy to handle. It is also available in colors so the different circuit lines can be color coded, especially in chemical industries. Because of its resilience, it is highly resistant to damage crushing although it can be fairly easily cut. It may also be used where flexing or vibration can damage steel tubing. Plastic tubing fittings vary slightly from steel tubing compression fitting.In fact, most steel tubing fittings can be used for the same services if a special sleeve is first inserted in the tubing to give it a crushing resistance at the compression point. Although testing for a specific purpose is recommended, a 4:1 factor of safety is considered good engineering practice in most of the fluid power systems.
In general, plastic tubing can be worked and installed with the ordinary tubing tools. It cuts easily and can be heated and given permanent bends; it can be used with standard metallic compression and flare fittings designed for metal tubing. Many new developments are being made in the nature of tools, fittings, quick disconnects and other devices especially for plastic fabrications.

1.9 Flexible Hoses

Hose assemblies are primarily used to connect fluid power systems to actuators that must be located on movable parts such as a cylinder coupled to a radius arm traversing in an arc, or a motor driving a machine carriage. A hose is manufactured from natural and synthetic rubbers and several plastics. This material is supported by fabric or by wire cloth, and wire braid may be used between plies or as an outside casing for high-pressure applications. Hose assemblies of nearly any length, complete with end connections, are available from most manufacturers. It is only necessary to specify the system service pressure and the fluid that is to be used. Extreme caution should be taken in changing a fluid or replacing hoses, however, to be sure that the hose material and fluid are compatible. Table 1.6 gives some typical hose sizes for single braid high tensile strength steel wire reinforcement with an inner tube made of oil-resistant nittrile (BTN) and a cover compound made up of black neoprene (oil-resistant and abrasion-resistant type).

Table 1.6 Typical hose sizes

Hose ID		Wire ID	Hose OD	Working Pressure		Burst Pressure		Minimum Bend Radius	Weight
in.	mm	mm	mm	MPa	psi	MPa	psi	mm	kg/m
3/16	4.8	9.5	11.8	25	3630	100	14280	90	0.190
1/4	6.4	10.8	12.8	22.5	3270	90	12840	100	0.222
5/16	7.9	12.5	14.5	21.5	3120	85	12280	115	0.261
3/8	9.5	14.6	16.8	18	2610	72	10280	130	0.324
1/2	12.7	17.6	19.8	16	2320	64	9180	180	0.418
5/8	15.9	21.1	23.1	13	1890	52	7420	200	0.476
3/4	19	24.7	27	10.5	1530	42	6000	240	0.619
1	25.4	32.5	35	8.8	1280	35	5020	300	0.883
1−1/4	31.8	39.5	43.5	6.3	920	25	3600	420	1.220
1−1/2	38.1	45.8	49.8	5	730	20	2850	500	1.408
2	50.8	59	63	4	580	16	2280	630	1.889

Steel end fittings are attached in various ways by clamping or squeezing the rubber hose between a serrated inner piece and an outer retaining ring. These end fittings are available for assembly on site by the user in a variety of designs that may require the use of portable tools, or as readymade hose assemblies, in which retaining rings are usually machine swaged at the factory.In some self-assembly versions, the inner end piece is screwed into the hose on a tapered interface to provide a compressive grip. In others, the outer sleeve is split and is held together on the hose by clamping screws to provide the same effect. The advantage of self-assembled hoses is that for emergency repairs, they can be cut to exact length on site, thus reducing the need for large stocks to be maintained, but they tend to be more costly than pre-assembled hoses.
End fittings used in flexible hoses tend to have a smaller internal diameter than their equivalent rigid tube fittings, thus creating slightly more resistance to flow. Most suppliers offer straight, angled and elbow end fittings for flexible hoses, with a variety of male and female threads.

1.9.1 Designation of Hoses

Hoses are fabricated in layers of elastomers and braided fabric or braided wires. The braided fabric or wires are used to increase the strength of the hoses. The hose may have a minimum of three layers including one braided layer or can have several layers to sustain the higher operating pressures. The steel wires have a spiral weave or cross weave. Spiral reinforced hoses have high strength and require fittings to be supplied by the manufacturer. The cross woven braids are re-usable and are easy to assemble. The inner tuber material of the hose should be compatible with the fluid.

Important considerations in the installation of hose assemblies are as follows**:**

1. Hose assemblies must be of proper overall lengths. Since a hose expands under pressure, both the hose length and space allowed for it must be adequate.
2. A hose under variations in working pressure must have enough length to expand and contract.
3. Do not clamp high- and low-pressure hoses together.
4. Never clamp a hose at a bend. Bend radii cannot absorb a change if clamped at the bend.
5. When there is a relative motion between two ends of a hose assembly, always allow the adequate length of travel.
6. To prevent twisting, a hose should be bent in the same plane as the motion of the part to which it is attached.

7. To prevent twisting in hose lines bent in two planes, clamp the hose at the change of the plane.
8. Use the proper hydraulic adaptors to reduce the number of joints and improve performance as well as appearance.
9. Wherever the radius falls below the required minimum bend, an angle adapter should be used.
10. Contact with sharp edges and rubbing against any surface should be avoided.
11. Arrange proper positioning of hose and adaptors before tightening to avoid distortion.
12. Apply clamps properly and keep tight to prevent abrasion due to line surge.
13. Be sure to use the proper strength of hose to maintain a good factor of safety.
14. Select the proper size of hose of stay within the recommended velocity range. Consult velocity-flow nomographs.
15. Prevent dirt, chips or any other foreign materials from entering the system during the fabrication of system.
16. Be sure that all the material used is compatible with the hydraulic fluid designed for the system.

1.11 Quick Disconnect Couplings

Another type of hydraulic fitting in regular use is the quick release coupling. This type of coupling in conjunction with flexible hoses connects movable components together hydraulically. Typical applications are mobile trailers and agriculture machinery towed behind tractors.

Quick release couplings usually comprise a plug and socket arrangement that provides a leak-proof joint when two parts are connected together, and that can be released easily without the use of tools. Each half of the coupling contains a spring-loaded ball or poppet that automatically closes on disconnection, so that two completely leak-free joints are obtained. Leaking during the process of disconnecting or connecting coupling is negligible

Objective-Type Questions

Fill in the Blanks

1. Stainless steel conductors or fittings may be used if extremely ________ environments are anticipated.

2. In fluid power installations, a galvanized pipe is unsatisfactory because galvanization tends to ________ into the system.

3. The choice of pipe, tube or hose depends on operating pressures of system and ________.

4. Pressure rating of various conductors depends on ________ of the material used and the wall thickness of the conductor.

5. Tubes are attached to the end fittings in various ways, some of which use metal-to-metal ________ known as a compression joint.

State True or False

1. Copper and cadmium tubing are most commonly used in a hydraulic system.

2. Steel piping in fluid power is most often jointed by threaded connections.

3. Welded sections are most desirable in fluid power systems.

4. Rigid steel tubing is generally suitable for high pressures.

5. Flow rates through smooth bore tubes should normally not exceed 1.2 m/s in pressure lines or 5 m/s in suction lines.

Review Questions

1. What is the purpose of a fluid distribution system?
2. Why should copper not be used in conductors or fitting?
3. Why can metals not be used with glycol fluid?
4. Why should the conductor have a greater strength than the system working pressure?
5. Why are smooth conductors desired?
6. What are the major disadvantages of steel pipes?
7. What is the recommended factor of safety for a fluid power system design?
8. Why is malleable iron used for steel pipe fittings?
9. What effect do threaded connections have on a fluid power system?
10. Why are unions not required in tubing systems?
11. Why is steel tubing used more often than steel pipe?
12. How do compression fittings prevent the tube from blowing out under pressure?
13. How can steel tube fittings be used with plastic tubings?
14. What major advantages does plastic tubing have over steel tubing?
15. What are the two disadvantages of steel pipes?
16. Define the burst pressure and working pressure of hydraulic pipes.
17. What is the use of quick disconnect coupling?
18. What is the difference between hydraulic tubing and hoses?
19. What is the difference between flared fitting and compression fitting?
20. List the factors influencing the selection of hoses.

Answers

Fill in the Blanks

1. Corrosive
2. Flake off
3. Flow
4. Tensile strength
5. Interference fit

State True or False

1. False
2. True
3. False
4. False
5. False

Lecture 6

ENERGY LOSSES IN HYDRAULIC SYSTEMS

Learning Objectives

Upon completion of this chapter, the student should be able to:

- State the differences between laminar and turbulent flows.
- Define Reynolds number and state its importance.
- Explain the Darcy–Weisbach equation.
- Explain the various types of joints used in fluid power.
- Evaluate the head losses for laminar and turbulent flow.
- Explain the various types of losses in fittings and valves.
- Design a system considering all head losses in the system.

1.1 Introduction

Liquids such as water or petrol flow much easily than other liquids such as oil. The resistance to flow is essentially a measure of the viscosity of a fluid. The greater the viscosity of a fluid, the less readily it flows and the more is the energy required to move it. This energy is lost because it is dissipated as heat.

Energy losses occur in valves and fittings. Various types of fittings, such as bends, couplings, tees, elbows, filters, strainers, etc., are used in hydraulic systems. The nature of path through the valves and fittings determines the amount of energy losses. The more circuitous is the path, the greater are the losses. In many fluid power applications, energy losses due to flow in valves and fittings exceed those due to flow in pipes. Therefore,a proper selection of fitting is essential. In general, the smaller the size of pipe and fittings, the greater the losses.

The resistance to flow of pipes, valves and fittings can be determined using empirical formulas that have been developed by experimentation. The energy equation and the continuity equation can be used to perform a complete analysis of a fluid power system. This includes calculating the pressure drops, flow rates and power losses for all components of the fluid power system.

The purpose of this chapter is to study the detailed circuit analysis of energy losses in fluid power systems containing valves, fittings and other power transmission and energy conversion elements.

1.2 Laminar and Turbulent Flows

When speaking of fluid flow, one refers to the flow of an ideal fluid. Such a fluid is presumed to have no viscosity. This is an idealized situation that does not exist. When referring to the flow of a real fluid, the effects of viscosity are introduced into the problem. This results in the development of shear stresses between neighboring fluid particles when they move at different velocities. In the case of an ideal fluid flowing in a straight conduit, all the particles move in parallel lines with equal velocity. In the flow of a real fluid, the velocity adjacent to the wall is zero; it increases rapidly within a short distance from the wall and produces a velocity profile such as shown in Figure1.1.

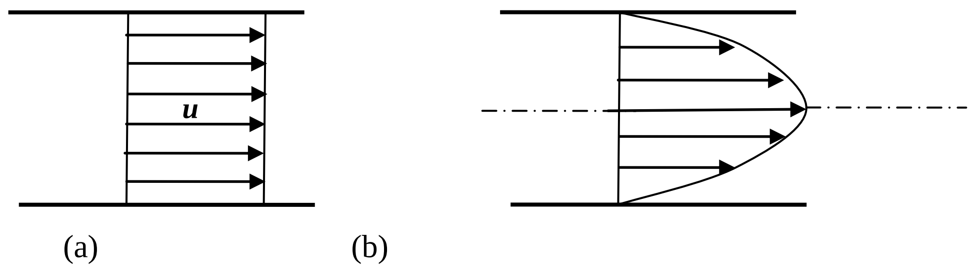


Figure 1.1 Typical velocity profile: (a) Ideal fluid.(b) Real fluid

There are two types of flow in pipes:

1. **Laminar flow:**This is also known as streamline or viscous flow and is illustrated in Fig.1.2**.** In streamline flow, the fluid appears to move by sliding of laminations of infinitesimal thickness relative to adjacent layers; that is, the particles move in definite and observable paths or streamlines. The flow characteristic of a viscous fluid is one in which viscosity plays a significant part.

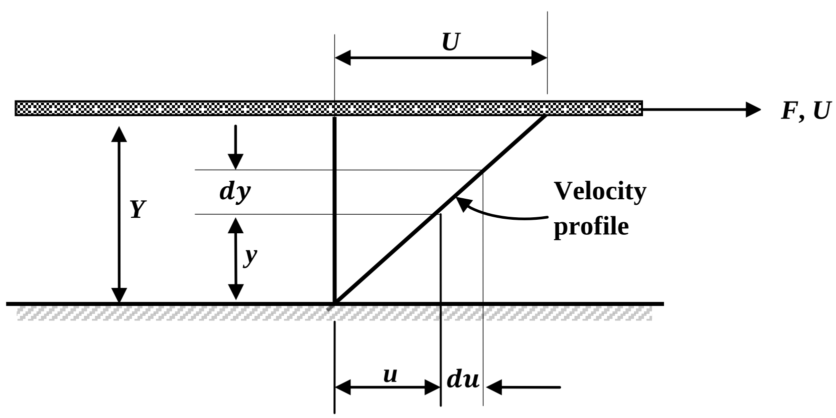


Figure 1.2 Laminar flow

2.**Turbulent flow:** It is illustrated in Fig.1.3. It is characterized by a fluid flowing in random way. The movement of particles fluctuates up and down in a direction perpendicular as well as parallel to the mean flow direction.

This mixing action generates turbulence due to the colliding fluid particles. This causes a considerable more resistance to flow and thus greater energy losses than those produced by laminar flow. A distinguishing characteristic of turbulence is its irregularity, there being no definite frequency, as in wave motion, and no observable pattern, as in the case of large eddies.

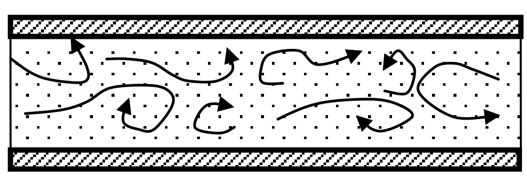


Figure 1.3 Turbulent flow

1.3 Reynolds Number

In the flow of a fluid through a completely filled conduit, gravity does not affect the flow pattern. It is also obvious that capillarity is of no practical importance, and hence significant forces are inertial force and fluid friction due to viscosity. The same is true for an airplane traveling at speed below that at which compressibility of air is appreciable. Also, for a submarine submerged far enough so as not to produce waves on the surfaces, the only forces involved are those of friction and inertia.

Considering the ratio of inertial forces to viscous forces, the parameter obtained is called the Reynolds number, in honor of Osborne Reynolds, who presented this in a publication of his experimental work in 1882. He conducted a series of experiments to determine the conditions governing the transition from laminar flow to turbulent flow. Reynolds came to a significant conclusion that the nature of the flow depends on the dimensionless parameter, that is,

$$\mathrm{Re} = \frac{vD\rho}{\mu}$$

Where v is the fluid velocity, D is the inside diameter of the pipe, ρ is the fluid density and μ is the absolute viscosity of the fluid.

1. If Reis less than 2000, the flow is laminar.
2. If Reis greater than 4000, the flow is turbulent.
3. Reynolds number between 2000 and 4000 covers a critical zone between laminar and turbulent flow.

It is not possible to predict the type of flow that exists within a critical zone. Thus, if the Reynolds number lies in the critical zone, turbulent flow should be assumed. If turbulent flow is allowed to exist, higher fluid temperatures occur due to greater frictional energy losses. Therefore, turbulent flow systems suffering from excessive fluid temperature can be helped by increasing the pipe diameter to establish laminar flow.

Example 1.1

The kinematic viscosity of a hydraulic fluid is 0.0001 m^2/s. If it is flowing in a 30-mm diameter pipe at a velocity of 6 m/s, what is the Reynolds number? Is the flow laminar or turbulent?

Solution: From the definition of Reynolds number, we can write

$$\mathrm{Re} = \frac{vD\rho}{\mu} = \frac{vD}{\mu/\rho} = \frac{vD}{\nu} = \frac{6\times 0.03}{0.0001} = 1800$$ <COMP: v and nu both being used>

SinceRe is less than 2000, the flow is laminar.

1.4 Darcy–Weisbach Equation

If a fluid flows through a length of pipe and pressure is measured at two stations along the pipe, one finds that the pressure decreases in the direction of flow. This pressure decrease is mainly due to the friction of the fluid against the pipe wall. Friction is the main cause of energy losses in fluid power systems. The prediction of this friction loss is one of the important problems in fluid power. It is a very complicated problem and only in special cases, the friction factor is computed analytically.

Head losses ina long pipe in which the velocity distribution has become fully established or uniform along its length can be found by Darcy's equation as

$$H_{\mathrm{L}} = f\left(\frac{L}{D}\right)\left(\frac{v^2}{2g}\right)$$

where*f*is the Darcy friction factor, L is the length of pipe (m), D is the inside diameter of the pipe (m), v is the average velocity (m/s) and g is the acceleration of gravity (m/s^2).

Theactual dependence of f on R_e has to be determined experimentally. It should be apparent that friction factors determined do not apply near the entrance portion of a pipe where the flow changes fairly quickly from one cross-section to the next or to any other flow in which acceleration terms are not negligible.

1.5 Frictional Losses in Laminar Flow

Darcy's equation can be used to find head losses in pipes experiencing laminar flow by noting that for laminar flow, the friction factor equals the constant 64 divided by the Reynolds number:

$$f = \frac{64}{\mathrm{Re}}$$

Substituting this into Darcy's equation gives the Hagen–Poiseuille equation:

$$H_{\mathrm{L}} = \frac{64}{\mathrm{Re}}\left(\frac{L}{D}\right)\left(\frac{v^2}{2g}\right)$$

Example 1.2

The kinematic viscosity of a hydraulic fluid is 0.0001 m^2/s. If it is flowing in a 20-mm diameter commercial steel pipe, find the friction factor in each case:

(a) The velocity is 2 m/s.
(b) The velocity is 10 m/s.

Solution:

a) If the velocity is 2 m/s, then

$$\mathrm{Re} = \frac{vD\rho}{\mu} = \frac{vD}{\mu/\rho} = \frac{vD}{\nu} = \frac{2\times 0.02}{0.0001} = 400$$

The flow is laminar. Now

$$f = \frac{64}{\mathrm{Re}} = \frac{64}{400} = 0.16$$

(b) If the velocity is 10 m/s, then

$$\mathrm{Re} = \frac{vD\rho}{\mu} = \frac{vD}{\mu/\rho} = \frac{vD}{\nu} = \frac{10 \times 0.02}{0.0001} = 2000$$

Theflow is laminar. Now

$$f = \frac{64}{\mathrm{Re}} = \frac{64}{2000} = 0.032$$

Example 1.3

The kinematic viscosity of a hydraulic fluid is 0.0001 m^2/s. If it is flowing in a 30-mm diameter pipe at a velocity of 6 m/s, find the head loss due to friction in units of bars for a 100-m smooth pipe. The oil has a specific gravity of 0.90.

Solution: We have

$$\mathrm{Re} = \frac{vD\rho}{\mu} = \frac{vD}{\mu/\rho} = \frac{vD}{\nu} = \frac{6 \times 0.03}{0.0001} = 1800$$

We can express the head loss in bar as

$$\begin{aligned} H_{\mathrm{L}} &= \frac{64}{\mathrm{Re}}\left(\frac{L}{D}\right)\left(\frac{v^2}{2g}\right) \\ &= \frac{64}{1800}\left(\frac{100}{0.030}\right)\left(\frac{6^2}{2 \times 9.81}\right) \\ &= 217.5 \text{ m} \end{aligned}$$

Hence,

$$\begin{aligned} \Delta p &= \gamma H_{\mathrm{L}} \\ &= 1000 \text{ kg/m}^3 \times 0.90 \times 9.81 \text{ m/s}^2 \times 217.5 \\ &= 1.92 \text{ MN/m}^2 \\ &= 1.92 \text{ MPa} \\ &= 19.2 \text{ bar} \end{aligned}$$

1.6 Frictional Losses in Turbulent Flow

Darcy's equation can be used to find head losses in pipes experiencing turbulent flow. However, the friction factor in turbulent flow is a function of Reynolds number and the relative roughness of the pipe.

1.6.1 Effect of Pipe Roughness

The relative roughness of pipe is defined as the ratio of inside surface roughness (ε) tothediameter:

$$\text{Relative roughness} = \frac{\varepsilon}{D}$$

Table 1.1 gives typical values of absolute roughness for various types of pipes.

Table 1.1 Typical values of absolute roughness for various types of pipe

Type of Pipe	ε **(mm)**
Glass or plastic	Smooth
Drawn tube	0.0015
Wrought iron	0.046
Commercial steel	0.046
Asphalted cast iron	0.12
Galvanized iron	0.15
Cast iron	0.26
Riveted steel	1.8

To determine the values of the friction factor for use in Darcy's equation, we use the Moody diagram. If we know the relative roughness and Reynolds number, the friction factor can be determined easily. No curves are drawn in the critical zone,Re lies in between 2000 and 4000because it is not possible to predict whether flow is laminar or turbulent in this region.At the left end of the chart (Reynolds number less than 2000), the straight line curves give the relationship for laminar flow:

$$f = \frac{64}{\text{Re}}$$

1.7 Frictional Losses in Valves and Fittings

For many fluid power applications, the majority of the energy losses occur in valves and fittings in which there is a change in the cross-section of flow path and a change in the direction of the flow. Tests have shown that head losses in valves and fittings are proportional to the square of the velocity of the fluid:

$$H_{\text{L}} = K\left(\frac{v^2}{2g}\right)$$

Where$K$ is called the loss coefficient of valve or fittings.K factors for commonly used valves are given in Table1.2.

Table 1.2 *K* factors for commonly used valves

Valve or Fitting		***K* Factor**
Globe valve	Wide open	10
	1/2 open	12.5
Gate valve	Wide open	0.20
	3/4 open	0.90
	1/2 open	4.5
	1/4 open	24
Return bend		2.2
Standard tee		1.8
Standard elbow		0.90
45° elbow		0.42
90° elbow		0.75
Ball check valve		4
Union socket		0.04

1.8 Equivalent Length Technique

We can find a length of pipe that for the same flow rate would produce the same head loss as a valve or fitting. This length of pipe, which is called the equivalent length of a valve or fitting, can be found by equating head losses across the valve or fitting and the pipe:

$$K\left(\frac{v^2}{2g}\right) = f\left(\frac{L}{D}\right)\left(\frac{v^2}{2g}\right)$$

Thisgives

$$L_\mathrm{e} = \left(\frac{KD}{f}\right)$$

where L_e is the equivalent length of a valve or fitting.

Example 1.4

For the hydraulic system shown in the Fig. 1.4, the following data are given:

(a) A pump adds 2.984 kW to a fluid (pump hydraulic power = 2.984 kW).
(b) The elevation difference between stations 1 and 2 is 6.096 m.
(c) The pump flow rate is 0.00158 m^3/s.
(d) The specific gravity of oil is 0.9.
(e) The kinematic viscosity of oil is 75 cS.
(f) The pipe diameter is 19.05 mm.
(g) Pipe lengths are as follows: 0.305, 1.22 and 4.88 m.

Find the pressure available at the inlet to the hydraulic motor. The pressure at the oil top surface level in the hydraulic tank is atmospheric (0 Pa gauge).

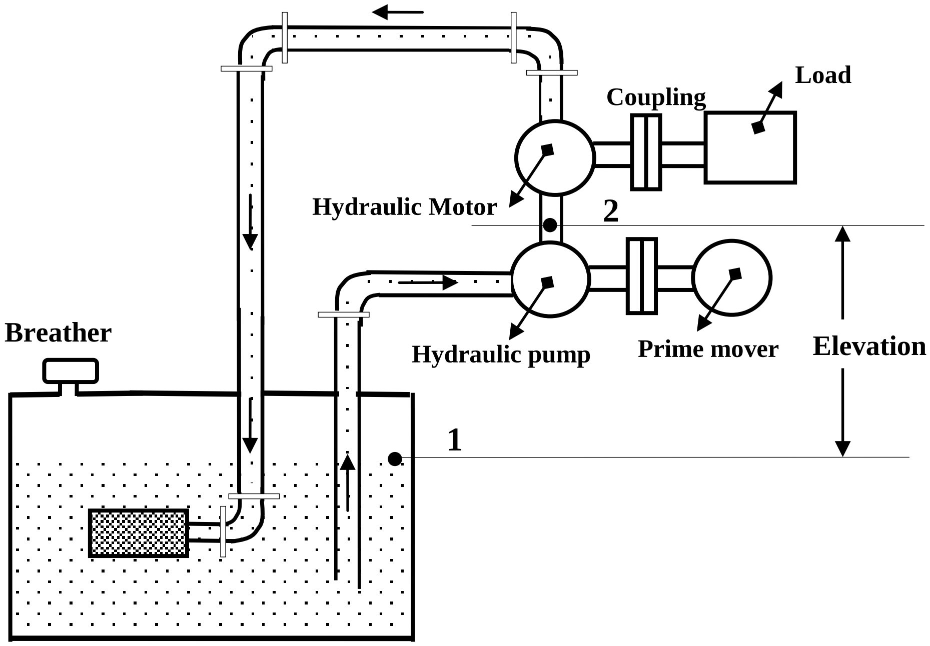


Figure 1.4

Solution:

Kinematic viscosity = 75 cS = 75 × 10^{-6} m^2/s

We write the energy equation between stations 1 and 2:

$$Z_1 + \frac{p_1}{\gamma} + \frac{v_1^2}{2g} + H_\mathrm{p} - H_\mathrm{m} - H_\mathrm{L} = Z_2 + \frac{p_2}{\gamma} + \frac{v_2^2}{2g}$$

Since there is no hydraulic motor between stations 1 and 2,

$$H_\mathrm{m} = 0, v_1 = 0 \text{ and } \frac{P_1}{\gamma} = 0$$

as oil tank is vented to the atmosphere. Now,

$$Z_2 - Z_1 = 6.096 \text{ m}$$

The velocity at point 2 is

$$v_2 = \frac{Q\,(\mathrm{m^3/s})}{A\,(\mathrm{m^2})} = \frac{0.00158}{\frac{\pi(0.01905)^2}{4}} = 5.54\,\mathrm{m/s}$$

The velocity head at point 2 is

$$\frac{v_2^2}{2g} = \frac{5.54^2}{2 \times 9.81} = 1.57 \text{ m}$$

Also,

$$\mathrm{Re} = \frac{vD\rho}{\mu} = \frac{vD}{\mu / \rho} = \frac{vD}{\nu} = \frac{5.54 \times 0.01905}{75 \times 10^{-6}} \cong 1400$$

So the flow is laminar. The friction is

$$f = \frac{64}{\mathrm{Re}} = \frac{64}{1400} = 0.0457$$

We can now find the head loss due to friction between stations 1 and 2:

$$H_{\mathrm{L}}=\frac{fL_{\mathrm{p}}}{D_{\mathrm{p}}}\times\frac{v^2}{2g}$$

where

$$\begin{aligned}L_{\mathrm{p}}&=4.88+0.305+1.22+\left(\frac{KD}{f}\right)_{\text{std elbow}}\\&=6.41+\left(\frac{0.9\times0.01905}{0.4057}\right)\\&=6.79\text{ m}\end{aligned}$$

and $$H_{\mathrm{L}}=\frac{0.0457\times6.79}{0.01905}\times1.57=25.6\text{ m}$$

Next use Bernoulli's equation to solve for P_2/γ:

$$Z_1+\frac{p_1}{\gamma}+\frac{v_1^2}{2g}+H_{\mathrm{p}}-H_{\mathrm{m}}-H_{\mathrm{L}}=Z_2+\frac{p_2}{\gamma}+\frac{v_2^2}{2g}$$

$$\begin{aligned}\Rightarrow\frac{p_2}{\gamma}&=(Z_1-Z_2)+H_{\mathrm{p}}+\frac{p_1}{\gamma}-\frac{v_2^2}{2g}-H_{\mathrm{L}}\\&=-6.096+H_{\mathrm{p}}+0-25.6-1.57\\&=H_{\mathrm{p}}-33.2\end{aligned}$$

The pump head is given by

$$H_{\mathrm{P}}=\frac{P\ (\mathrm{W})}{\gamma\ (\mathrm{N/m^3})\times Q\ (\mathrm{m^3/s})}$$

Now $\gamma=8817\ \mathrm{N/m^3}$ and $P=2984$ W. So

$$H_{\mathrm{P}}=\frac{2984\,\mathrm{W}}{8817\ (\mathrm{N/m^3})\times0.00158\ (\mathrm{m^3/s})}=214.3\text{ m}$$

Now

$$\begin{aligned}\frac{p_2}{\gamma}&=H_{\mathrm{p}}-33.2\\&=214.3-33.2\\&=181.1\text{m of oil}\end{aligned}$$

Finally, we solve for the pressure at station 2:

$$\frac{p_2}{\gamma}=181.1\times8817=1600000=1600\,\mathrm{kPa}$$

Example 1.5

The oil tank for a hydraulic system (shown in Fig. 1.5) has the following details:

(a) The oil tank is air pressurized at 68.97 kPa gauge pressure.
(b) The inlet to the pump is 3.048 m below the oil level.
(c) The pump flow rate is 0.001896 m^3/s.
(d) The specific gravity of oil is 0.9.
(e) The kinematic viscosity of oil is 100 cS.
(f) Assume that the pressure drop across the strainer is 6.897 kPa.

(g) The pipe diameter is 38.1 mm.
(h) The total length of the pipe is 6.097 m.

Find the pressure at station 2.

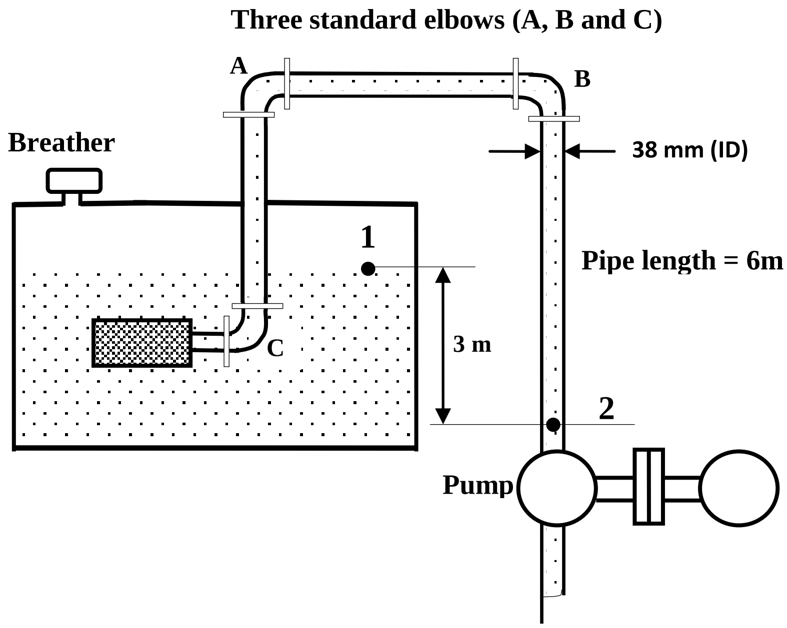


Figure 1.5

Solution: Givenp1 = 68.97kPa, ,Z1−Z2 = 3m, Q = 0.001896m3/s, ps = 6.897kPa, SG = 0.9, Dp = 38.1mm, ν=100 cS = 100 × 10–6 m2/s, Lp= 6m. We have to find P2. This problem can be solved by the application ofmodified Bernoulli's (energy) equation:

$$Z_1 + \frac{p_1}{\gamma} + \frac{v_1^2}{2g} + H_{\mathrm{p}} - H_{\mathrm{m}} - H_{\mathrm{L}} = Z_2 + \frac{p_2}{\gamma} + \frac{v_2^2}{2g}$$

Also, given

$$Z_1 - Z_2 = 3\,\mathrm{m}$$

We can write $H_{\mathrm{m}} = 0,$ as there being no motor between points1 and 2, the motor head is zero. Assuming the oil tank area (cross-section) to be large, the velocity at point 1 is $v_1 = 0$ (negligible velocity). To solve forP_2, let us compute and substitute different quantities into the energy equation.

The pressure head at station 1 is

$$\frac{p_1}{\gamma} = \frac{68970}{8817} = 7.82\,\mathrm{m}$$

The velocity at point 2 is

$$v_2 = \frac{Q\,(\mathrm{m}^3/\mathrm{s})}{A\,(\mathrm{m}^2)} = \frac{0.001896}{\frac{\pi(0.0381^2)}{4}} = 1.66\,\mathrm{m/s}$$

The velocity head at point 2 is

$$\frac{v_2^2}{2g}=\frac{1.66^2}{2\times 9.81}=0.141\,\text{m}$$ The Reynolds number is

$$\text{Re}=\frac{vD\rho}{\mu}=\frac{vD}{\mu/\rho}=\frac{vD}{\nu}=\frac{1.66\times 0.0381}{100\times 10^{-6}}\cong 632$$

So the flow is laminar.The friction is

$$f=\frac{64}{\text{Re}}=\frac{64}{632}=0.101$$

Point 2 is before the pump; therefore, $H_\text{p}=0$.

We can now find the head loss due to friction between stations 1 and 2

$$H_\text{L}=\frac{fL_\text{p}}{D_\text{p}}\times\frac{v^2}{2g}+\text{Head loss across the strainer}$$

Three standard elbows are used:

$$L_\text{p}=6.097+3\left(\frac{KD}{f}\right)_{\text{std elbow}}$$

$$=6.097+3\left(\frac{0.9\times 0.0381}{0.101}\right)=7.12$$

Pressure drop across the strainer is 6.9 kPa. The head loss across strainer can be calculated as

$$\text{Head loss across strainer}=\frac{\Delta p_\text{strainer}}{\gamma}=\frac{6900}{8817}=0.782\text{ m}$$

Now use Bernoulli's theorem

$$Z_1+\frac{p_1}{\gamma}+\frac{v_1^2}{2g}+H_\text{p}-H_\text{m}-H_\text{L}=Z_2+\frac{p_2}{\gamma}+\frac{v_2^2}{2g}$$

$$\Rightarrow\frac{p_2}{\gamma}=(Z_1-Z_2)+\frac{p_1}{\gamma}-H_\text{L}-\frac{v_2^2}{2g}$$

$$\Rightarrow\frac{p_2}{\gamma}=3.048+7.82-3.44-0.141$$

$$= 7.29\text{m of oil}$$

Finally, we solve for p_2:

$$\frac{p_2}{\gamma}=7.29$$

So $$p_2=7.29\times 8817=64300\text{ Pa}=64.3\text{kPa}$$

If point 2 is after the pump, the pump head is given by

$$H_\text{p}=\frac{P\text{ (W)}}{\gamma\text{ (N/m}^3)\times Q\text{ (m}^3\text{/s)}}$$

Here $\gamma=8817\text{ N/m}^3$. We know that

$$\text{Power}(P)=p_1\times Q$$

$$\Rightarrow P=68.97\text{ (kN/m}^3)\times 0.001896\text{ (m}^3\text{/s)}=0.13\text{ kW}$$

So, pump head is

$$H_p = \frac{0.13 \times 10^3 \text{ W}}{8817\,(\text{N/m}^3) \times 0.001896\,(\text{m}^3/\text{s})} = 7.7 \text{ m}$$

We also know that

$$Z_1 + \frac{p_1}{\gamma} + \frac{v_1^2}{2g} + H_p - H_m - H_L = Z_2 + \frac{p_2}{\gamma} + \frac{v_2^2}{2g}$$

$$\Rightarrow \frac{p_2}{\gamma} = (Z_1 - Z_2) + \frac{p_1}{\gamma} - H_L + H_p - \frac{v_2^2}{2g}$$

$$\Rightarrow \frac{p_2}{\gamma} = 3.048 + 7.82 - 3.44 + 7.7 - 0.141 \cong 15\text{m of oil}$$

Finally, we solve for p_2:

$$\frac{p_2}{\gamma} \cong 15$$

$$\Rightarrow p_2 = 15 \times 8817 = 132255 \text{ Pa} = 132.26 \text{kPa}$$

Example 1.7

For the system shown in Fig. 1.6, the following new data are applicable:
Pipe 1: length = 8m, ID = 25mm
Pipe 2: length =8m, ID = 25mm
The globe valve is 25mm in size and is wide open.
SG =0.90 kinematic viscosity $(\nu = 0.0001\,\text{m}^2/\text{s})$ and Q =0.0025 m³/s
Find $p_2 - p_1$ in units of bars.

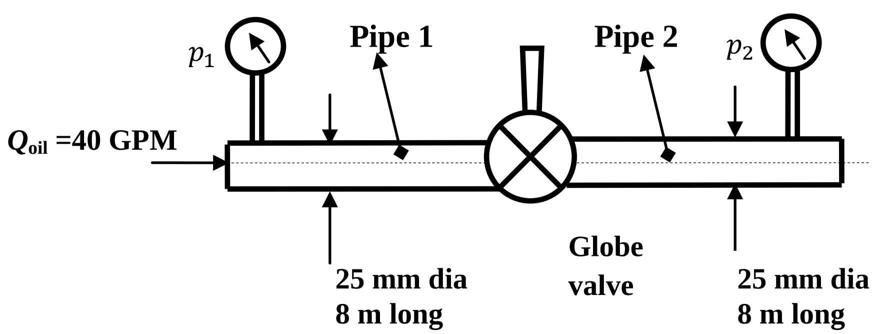


Figure 1.6

Solution: The velocity can be calculated as

$$v = \frac{Q\,(\text{m}^3/\text{s})}{A\,(\text{m}^2)} = \frac{0.0025}{\frac{\pi(0.025^2)}{4}} = 5.09\,\text{m/s}$$

We know that

$$\mathrm{Re}=\frac{vD\rho}{\mu}=\frac{vD}{\mu/\rho}=\frac{vD}{\nu}=\frac{5.09\times0.025}{0.0001}\cong1272$$

So the flow is laminar. Now friction factor is

$$f=\frac{64}{\mathrm{Re}}=\frac{64}{1272}=0.0503$$

Also, head loss is

$$H_{\mathrm{L}}=\frac{f\,L_{\mathrm{p}}}{D_{\mathrm{p}}}\times\frac{v^2}{2g}$$

Now

$$L_{\mathrm{p}}=8+8+\left(\frac{KD}{f}\right)_{\text{std elbow}}=16+\left(\frac{10\times0.025}{0.0503}\right)=21\,\mathrm{m}$$

K = 10 for globe valve (fully open). So

$$H_{\mathrm{L}}=\frac{0.0503\times21}{0.025}\times\frac{5.09^2}{2\times9.81}=55.8\text{ m of oil}$$

Now

$$\frac{\Delta p}{\gamma}=H_{\mathrm{L}}$$

$$\Rightarrow\Delta p=\gamma H_{\mathrm{L}}$$

$$=(1000\times0.9\times9.81)\times55.8$$

$$=493000\ \mathrm{N/m^2}$$

So

$$p_2-p_1=-\Delta p$$

$$=493000\ \mathrm{N/m^2}$$

$$=-493\,\mathrm{kPa}=-4.93\,\mathrm{bar}$$

Example 1.8

For the system shown in Fig. 1.7, the following data are applicable: $P_1=7\,\mathrm{bar}$, Q =0.002 $\mathrm{m^3/s}$.

Pipe: total length = 15m and ID = 38mm

Oil: SG =0.90 and kinematic viscosity $(\nu=0.0001\,\mathrm{m^2/s})$

Solve for p_2 in units of bars.

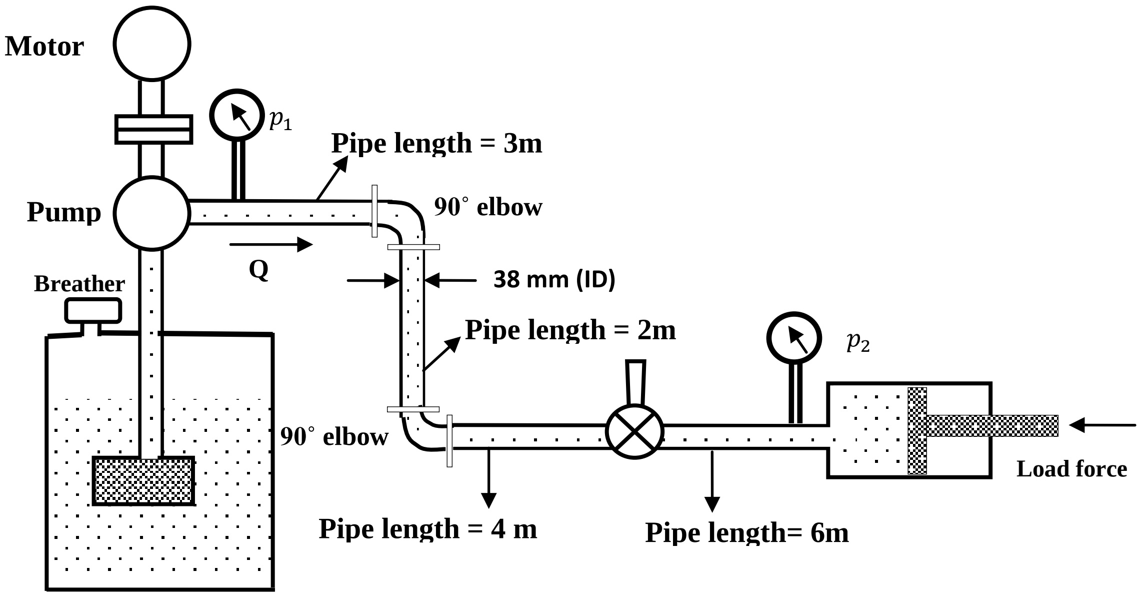


Figure 1.7

Solution: The velocity v is

$$v = \frac{Q\,(\mathrm{m^3/s})}{A\,(\mathrm{m^2})} = \frac{0.002}{\frac{\pi (0.038)^2}{4}} = 1.76 \text{ m/s}$$

We know that

$$\mathrm{Re} = \frac{vD\rho}{\mu} = \frac{vD}{\mu/\rho} = \frac{vD}{\nu} = \frac{1.76 \times 0.038}{0.0001} \cong 669$$

So the flow is laminar. The friction is

$$f = \frac{64}{\mathrm{Re}} = \frac{64}{669} = 0.096$$

Now we can calculate the equivalent length (L_e) as

$$\begin{aligned} L_e &= L_{\text{pipe}} + \left(\frac{KD}{f}\right)_{\text{globe valve}} + 2\left(\frac{KD}{f}\right)_{90^\circ\text{elbow}} \\ &= 15 + \left(\frac{10 \times 0.038}{0.096}\right) + 2\left(\frac{0.75 \times 0.038}{0.096}\right) \\ &= 15 + 4 + 0.6 = 19.6 \text{ m} \end{aligned}$$

Head loss due to friction is given by

$$H_L = \frac{0.096 \times 19.6}{0.038} \times \frac{1.76^2}{2 \times 9.81} = 7.82 \text{ m of oil}$$

Now we can express the head loss due to friction in terms as pressure using

$$\frac{\Delta p}{\gamma} = H_{\mathrm{L}}$$

$$\Rightarrow \Delta p = \gamma H_{\mathrm{L}} = (1000 \times 0.9 \times 9.81) \times 7.82$$

$$= 690000 \text{ N/m}^2 = 0.69 \,\text{bar}$$

From the pressure drop we can calculated p_2 as

$$p_1 - p_2 = \Delta p = 0.69 \,\text{bar}$$

$$\Rightarrow 7 - p_2 = 0.69 \,\text{bar}$$

$$\Rightarrow p_2 = 6.31 \text{ bar}$$

Example 1.9

For the fluid power system shown in Fig. 1.8, determine the external load *F* that a hydraulic cylinder can sustain while moving in an extending direction. Take frictional pressure losses into account. The pump produces a pressure increase of 6.9 MPa from the inlet port to the discharge port and a flow rate of 0.00253 m^3/s. The following data are applicable.

Kinematic viscosity	$0.0000930 \text{ m}^2/\text{s}$
Weight density of oil	7840 N/m^3
Cylinder piston diameter	0.203 m
Cylinder rod diameter	0.102 m

All elbows are at 90° with *K* factor = 0.75. Pipe length and inside diameters are given in Fig. 1.7.

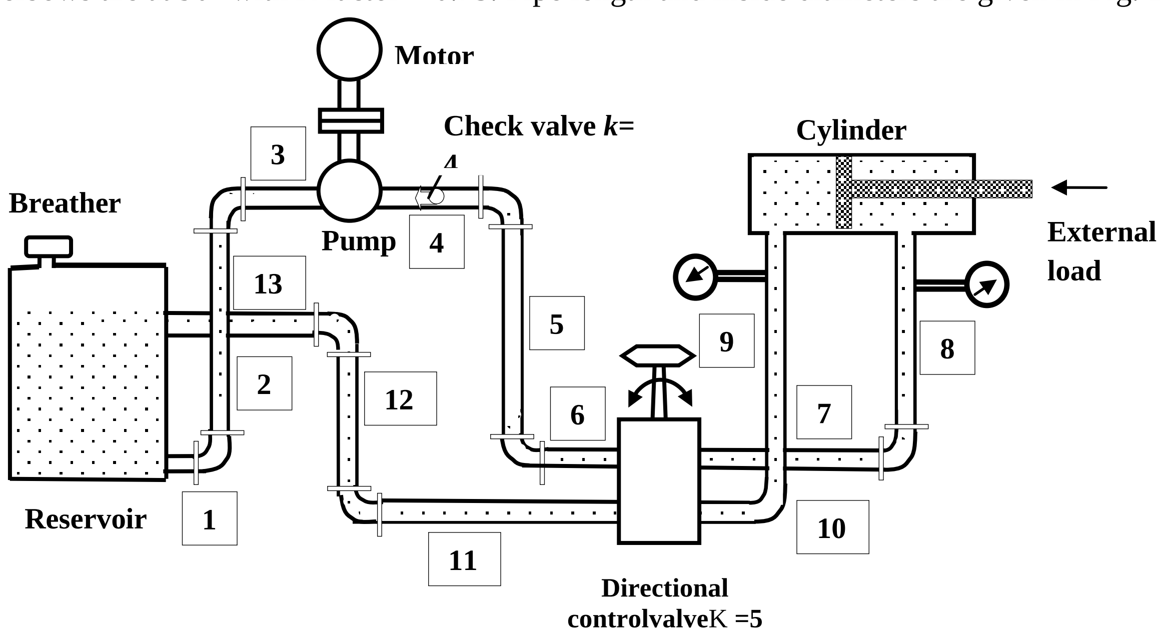


Figure 1.8

Pipe Number	Length (m)	Diameter	Pipe Number	Length (m)	Diameter
1	0.610	0.0381	8	1.52	0.0254
2	1.83	0.0381	9	1.52	0.0190
3	0.610	0.0381	10	1.52	0.0190
4	15.2	0.0254	11	18.3	0.0190
5	3.05	0.0254	12	3.05	0.0190
6	1.52	0.0254	13	6.10	0.0190
7	1.52	0.0254			

(a) Determine the heat generation rate.
(b) Determine the extending and retracting speeds of cylinder.

Solution: We can use the following equations

$$H_{\mathrm{L}} = \sum_{1}^{13}\left(\frac{fL_{\mathrm{p}}}{D_{\mathrm{p}}} + K\right)\frac{v^2}{2g}$$

Now

$$v = \frac{Q\,(\mathrm{m^3/s})}{A\,(\mathrm{m^2})}$$

We know that

$$\mathrm{Re} = \frac{vD\rho}{\mu} = \frac{vD}{\mu/\rho} = \frac{vD}{\nu}$$

Also

$$Q_{\mathrm{return}} = 0.00253 \times \frac{(0.203^2 - 0.102^2)}{0.203} = 0.00189\ \mathrm{m^3/s}$$

Velocity calculation:

$$v_{1,2,3} = \frac{0.00253\,(\mathrm{m^3/s})}{\frac{\pi(0.0381^2)}{4}\ \mathrm{m^2}} = 2.22\ \mathrm{m/s}$$

$$v_{4,5,6} = \frac{0.00253\,(\mathrm{m^3/s})}{\frac{\pi(0.0254^2)}{4}\ \mathrm{m^2}} = 4.99\ \mathrm{m/s}$$

$$v_{7,8} = \frac{0.00189\,(\mathrm{m^3/s})}{\frac{\pi(0.0254^2)}{4}\ \mathrm{m^2}} = 3.73\ \mathrm{m/s}$$

$$v_{9,10} = \frac{0.00253\,(\mathrm{m^3/s})}{\frac{\pi(0.0190^2)}{4}\ \mathrm{m^2}} = 8.92\ \mathrm{m/s}$$

$$v_{11,12,13} = \frac{0.00189\,(\mathrm{m^3/s})}{\frac{\pi(0.0190^2)}{4}\ \mathrm{m^2}} = 6.67\ \mathrm{m/s}$$

Reynolds number calculation:

$$\mathrm{Re}_{(1,2,3)} = \frac{2.22\times 0.0381}{0.000093} = 909$$

$$\mathrm{Re}_{(4,5,6)} = \frac{4.99\times 0.0254}{0.000093} = 1362$$

$$\mathrm{Re}_{(7,8)} = \frac{3.73\times 0.0254}{0.000093} = 1018$$

$$\mathrm{Re}_{(9,10)} = \frac{8.92\times 0.019}{0.000093} = 1822$$

$$\mathrm{Re}_{(11,12,13)} = \frac{6.67\times 0.019}{0.000093} = 1363$$

All flows are laminar. Now

$$f = \frac{64}{\mathrm{Re}}$$

Now head loss can be calculated for each element:

$$H_{\mathrm{L}(1,2,3)} = \left(\frac{64}{909}\frac{3.05}{0.0381} + 1.5\right)\frac{2.22^2}{2\times 9.81} = 1.79\ \mathrm{m} = 14000\,\mathrm{Pa}$$

$$H_{\mathrm{L}(4,5,6)} = \left(\frac{64}{1362}\frac{19.8}{0.0254} + 10.5\right)\frac{4.99^2}{2\times 9.81} = 59.8\ \mathrm{m} = 469000\,\mathrm{Pa}$$

$$H_{\mathrm{L}(7,8)} = \left(\frac{64}{1018}\frac{3.05}{0.0254} + 0.75\right)\frac{3.73^2}{2\times 9.81} = 5.89\ \mathrm{m} = 46200\,\mathrm{Pa}$$

$$H_{\mathrm{L}(9,10)} = \left(\frac{64}{1822}\frac{3.05}{0.019} + 0.75\right)\frac{8.92^2}{2\times 9.81} = 25.9\ \mathrm{m} = 203000\,\mathrm{Pa}$$

$$H_{\mathrm{L}(11,12,13)} = \left(\frac{64}{1363}\frac{27.4}{0.019} + 1.5\right)\frac{6.67^2}{2\times 9.81} = 157\ \mathrm{m} = 1230000\,\mathrm{Pa}$$

Nowexternal load F that a hydraulic cylinder can sustain while moving in an extending directioncan be calculated as

$$F = [\text{Pressure on piston side} \times \text{Area}] - [\text{Pressure on rod side} \times \text{Area}]$$

$$F = \left[[(6900000) - (14000 + 469000 + 203000)] \times \frac{\pi(0.203^2)}{4}\right] - \left[(46200 + 1230000) \times \frac{\pi(0.203^2 - 0.102^2)}{4}\right]$$

$$F = (201000) - (30900) = 170000\,\mathrm{N}$$

(a) Heat generation rate (power loss in watts) = Pressure × Discharge

Power loss = $\{(14000 + 469000 + 20300) \times (0.00253) + (46200 + 1230000) \times (0.00189)\}$

= 1740 + 2410

$= 4150$ W $= 4.15$ kW

(b) Now to calculate forward and retracting speed of the cylinder

$$\text{Forward velocity of the piston} = \frac{Q_{\text{pump}}}{\text{Area of the piston}}$$

$Q_{\text{pump}} = 0.00253 \text{ m}^3/\text{s}$

Cylinder piston diameter = 0.2032 m

Area of piston $A = \dfrac{\pi(0.2032^2)}{4} \text{ m}^2$

Knowing the area of piston and discharge we can calculate the forward velocity of piston as

$$V_{\text{extending}} = \frac{Q\ (\text{m}^3/\text{s})}{A\,(\text{m}^2)} = \frac{0.00253}{\dfrac{\pi(0.2032^2)}{4}} = 0.0780 \text{ m/s}$$

Now

$$\text{Return velocity of the piston} = \frac{Q_{\text{pump}}}{\text{Annulus area}}$$

Cylinder rod diameter = 0.1016 m

Area of rod = $\dfrac{\pi(0.1016^2)}{4} \text{ m}^2$

Annulus area = $\dfrac{\pi(0.2032^2)}{4} - \dfrac{\pi(0.1016^2)}{4}$

Knowing the annulus area and discharge we can calculate the return velocity of piston as

$$V_{\text{extending}} = \frac{Q\ (\text{m}^3/\text{s})}{A\,(\text{m}^2)} = \frac{0.00253}{\dfrac{\pi(0.2032^2)}{4} - \dfrac{\pi(0.1016^2)}{4}} = 0.104 \text{ m/s}$$

Objective-Type Questions
Fill in the Blanks

1. In fluid power systems, energy losses due to flow in valves and fittings may ______ those due to flow in pipes.
2. Darcy's equation can be used to find head losses in pipes experiencing ______ flow.
3. In the case of an ideal fluid flowing in a straight conduit, all the particles move in parallel lines with equal ______.
4. The flow characteristic of a viscous fluid is one in which ______ plays a significant part.
5. Energy loss in turbulent flow is ______ than laminar flow.

State True or False

1. Reynolds number is the ratio of viscous forces to inertial force.
2. In the flow of a fluid through a completely filled conduit, gravity affects the flow pattern.
3. Friction is the main cause of energy losses in fluid power systems.
4. In the flow of a real fluid, the velocity adjacent to the wall is maximum.
5. A distinguishing characteristic of turbulence is its irregularity, no definite and no observable pattern.

Review Questions

1. Differentiate between laminar flow and turbulent flow.
2. List the main causes of turbulence in fluid flow.
3. Define Reynolds number and list its range for laminar and turbulent flows.
4. Briefly explain the method to calculate the equivalent length of a valve or fitting.
5. Define the relative roughness and *K* factor of a valve or fitting.
6. Write an expression for pressure drop down a pipe in terms of friction factor.
7. What are the important conclusions of the Reynolds experiment?
8. Name two causes of turbulence in fluid flow.
9. What is meant by the equivalent length of a valve or fitting?
10. Why is it important to select properly the size of pipes, valves and fittings in hydraulic systems?

Answers

Fill inthe Blanks

1.Exceed
2.Turbulent
3.Velocity
4.Viscosity
5.Less

State True or False

1.False
2.False
3.True
4.False
5.True

Lecture 7

HYDRAULIC PUMPS

Learning Objectives

Upon completion of this chapter, the student should be able to:

- Classify the hydraulic pumps used in the industry.
- Differentiate between positive displacement and non-positive displacement pumps.
- Explain the working and construction of gear, vane and piston pumps.
- Evaluate the discharge parameters of gear, vane and piston pumps.
- Define mechanical, volumetric and overall efficiency of pumps.
- Evaluate the performance parameters of gear, vane and piston pumps.
- Differentiate between internal and external gear pumps.
- Differentiatebetween a bent-axis piston pump and a swash plate.
- State the advantage of balance vane pumps.
- Explain cavitation and various means to control it.
- Explain the importance of noise control in pumps.
- Write a computer program to evaluate the performance of the system.

1.1Introduction

The function of a pump is to convert mechanical energy into hydraulic energy. It is the heart of any hydraulic system because it generates the force necessary to move the load. Mechanical energy is delivered to the pump using a prime mover such as an electric motor.Partial vacuum is created at the inlet due to the mechanical rotation of pump shaft. Vacuum permits atmospheric pressure to force the fluid through the inlet line and into the pump. The pump then pushes the fluid mechanically into the fluid power actuated devices such as a motor or a cylinder.

Pumps are classified into three different ways and must be considered in any discussion of fluid power equipment.

1. **Classification based on displacement:**
   - Non-positive displacement pumps (hydrodynamic pumps).
   - Positive displacement pumps (hydrostatic pumps).

2. **Classification based on delivery:**
   - Constant delivery pumps.
   - Variable delivery pumps.

3. **Classification based on motion:**
   - Rotary pump.
   - Reciprocating pump.

1.2 Classification of Pumps

1.2.1Classification Based on Displacement

1.2.1.1 Non-Positive Displacement Pumps

Non-positive displacement pumps are primarily velocity-type units that have a great deal of clearance between rotating and stationary parts. Non-displacement pumps are characterized by a high slip that increases as the back pressure increases, so that the outlet may be completely closed without damage to the pump or system. Non-positive pumps do not

develop a high pressure but move a large volume of fluid at low pressures. They have essentially no suction lift. Because of large clearance space, these pumps are not self-priming. In other words, the pumping action has too much clearance space to seal against atmospheric pressure. The displacement between the inlet and the outlet is not positive. Therefore, the volume of fluid delivered by a pump depends on the speed at which the pump is operated and the resistance at the discharge side. As the resistance builds up at the discharge side, the fluid slips back into the clearance spaces, or in other words, follows the path of least resistance. When the resistance gets to a certain value, no fluid gets delivered to the system and the volumetric efficiency of the pump drops to zero for a given speed. These pumps are not used in fluid power industry as they are not capable of withstanding high pressure. Their maximum capacity is limited to 17–20 bar.These types of pumpsare primarily used for transporting fluids such as water, petroleum,etc.,from one location to another considerable apart location. Performance curves for positive and non-positive displacement pumps are shown in Fig.1.1.
Thetwo most common types of hydrodynamic pumps are the centrifugal and the axial flow propeller pumps.

Advantages and disadvantages ofnon-positive displacement pumps

The advantages are as follows:

1.Non-displacement pumps have fewer moving parts.
2.Initial and maintenance cost is low.
3. They give smooth continuous flow.
4. They are suitable for handling almost all types of fluids including slurries and sledges.
5.Their operation is simple and reliable.

The disadvantages are as follows:
1.Non-displacement pumps are not self-priming and hence they must be positioned below the fluid level.
2. Discharge is a function of output resistance.
3.Low volumetric efficiency.

1.2.1.2Positive Displacement Pumps

Positive displacement pumps, in contrast, have very little slips, are self-priming and pump against very high pressures, but their volumetric capacity is low. Positive displacement pumps have a very close clearance between rotating and stationary parts and hence are self-priming. Positive displacement pumps eject a fixed amount of fluid into the hydraulic system per revolution of the pump shaft. Such pumps are capable of overcoming the pressure resulting from mechanical loads on the system as well as the resistance of flow due to friction. This equipment must always be protected by relief valves to prevent damage to the pump or system. By far, a majority of fluid power pumps fall in this category, including gear, vane and piston pumps.Performance curves for positive and non-positive displacement pumps are shown in Fig. 1.1.

Positive displacement pumps are classified based on the following characteristics:

1. **Type of motion of pumping element:** Based on the type of motion of pumping element, positive displacement pumps are classified as follows:
- Rotary pumps, for example, gear pumps and vane pumps.
- Reciprocating pumps, for example, piston pumps.

2. **Displacement characteristics:** Based on displacement characteristics, positive displacement pumps are classified as follows:
- Fixed displacement pumps.
- Variable displacement pumps.
3. **Type of pumping element**.

The advantages of positive displacement pumps over non-positive displacement pumps are as follows:

1. They can operate at very high pressuresof up to 800 bar (used for lifting oils from very deep oil wells).
2. They can achieve a high volumetric efficiency of up to 98%.
3. They are highly efficient and almost constant throughout the designed pressure range.
4. They are a compact unit, havinga high power-to-weight ratio.
5. They can obtain a smooth and precisely controlled motion.
6. By proper application and control, they produce only the amount of flow required to move the load at the desired velocity.
7. They have a great flexibility of performance. They can be made to operate over a wide range of pressures and speeds.

1.2.2 Classification Based on Delivery

1.2.2.1 Constant Delivery Pumps

Constant volume pumps always deliver the same quantity of fluid in a given time at the operating speed and temperature. These pumps are generally used with relatively simple machines, such as saws or drill presses or where a group of machines is operated with no specific relationship among their relative speeds. Power for reciprocating actuators is most often provided by constant volume pumps.

1.2.2.2Variable Delivery Pumps

The output of variable volume pumps may be varied either manually or automatically with no change in the input speed to the pump. Variable volume pumps are frequently used for rewinds, constant tension devices or where a group of separate drives has an integrated speed relationship such as a conveyor system or continuous processing equipment.

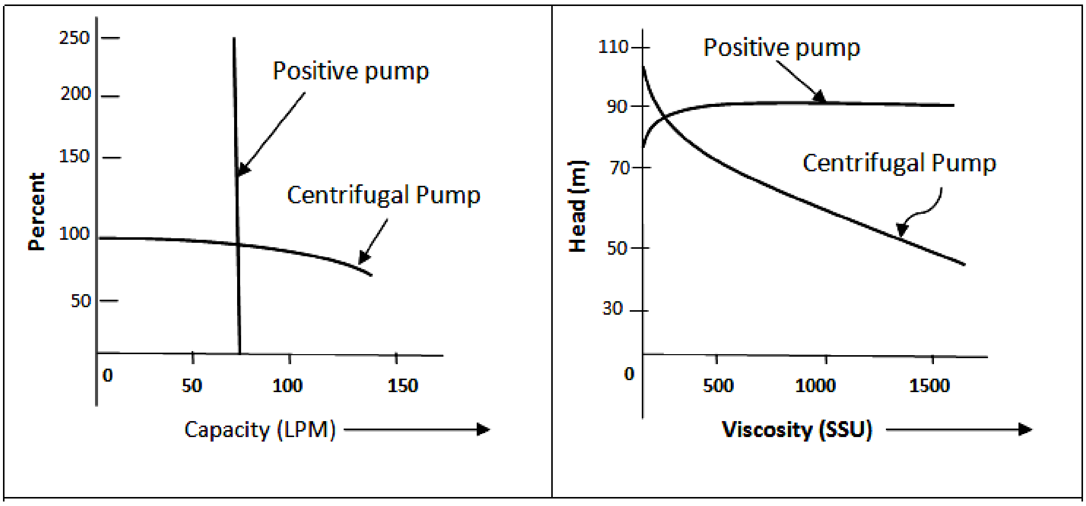
1.2.3Classification Based on Motion

This classification concerns the motion that may be either *rotary* or *reciprocating*. It was of greater importance when reciprocating pumps consisted only of a single or a few relatively large cylinders and the discharge had a large undesirable pulsation. Present-day reciprocating pumps differ very little from rotary pumps in either external appearance or the flow characteristics.

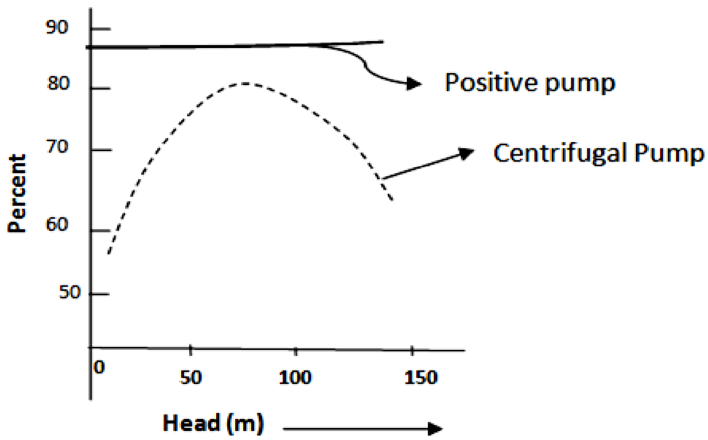
Differences between positive displacement pumps and non-positive displacement pumps are enumerated in Table 1.1.

Table 1.1 Differences between positive displacement pumps and non-positive displacement pumps

Positive Displacement Pumps	Non-positive Displacement Pumps
The flow rate does not change with head	The flow rate decreases with head
The flow rate is not much affected by the viscosity of fluid	The flow rate decreases with the viscosity
Efficiency is almost constant with head	Efficiency increases with head at first and then decreases



(a)



(b)

Figure 1.1 Performance curves for positive and non-positive displacement pumps

1.3Pumping Theory

A positive displacement hydraulic pump is a device used for converting mechanical energy into hydraulic energy.It is driven by a prime mover such as an electric motor.It basically performs twofunctions.First, it creates a partial vacuum at thepump inlet port.This vacuum enables atmospheric pressure to force the fluid from the reservoir into the pump.Second,the mechanical action of the pump traps this fluid within the pumping cavities,transports it through the pump and forces it into the hydraulic system. It is important to note that pumps create flow not pressure. Pressure is created by the resistance to flow.

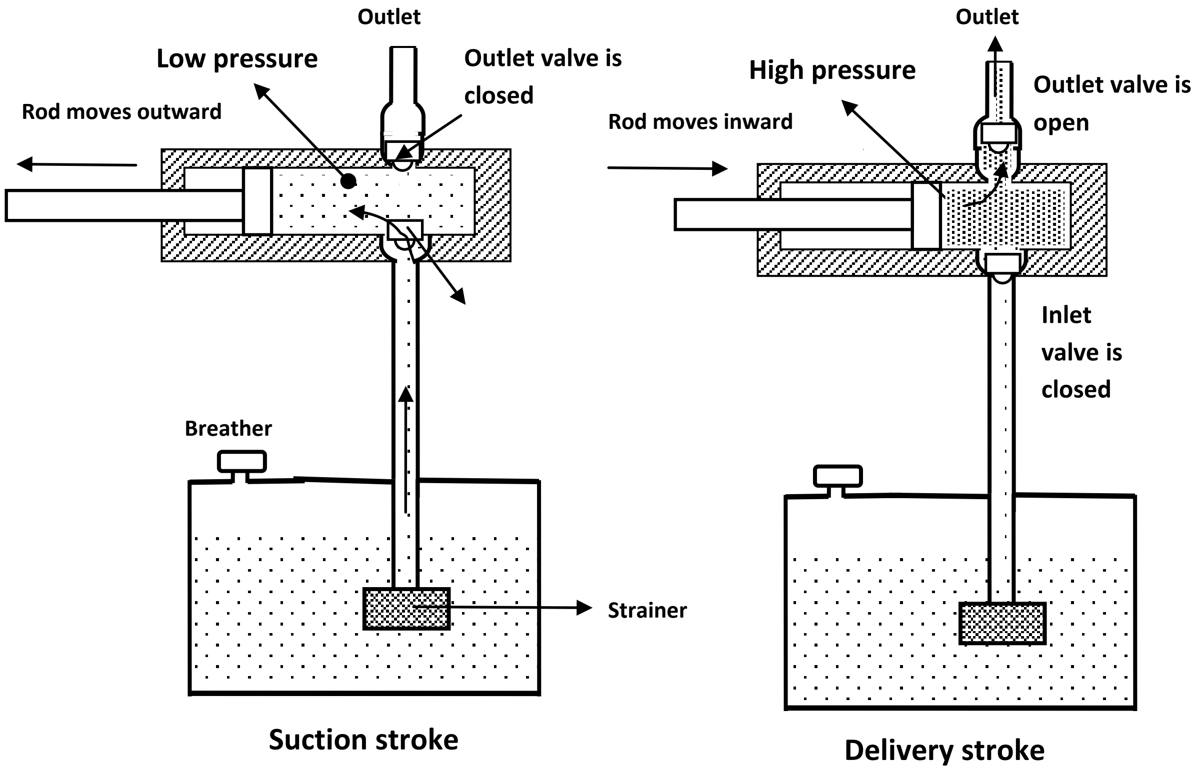


Figure 1.2 Illustration of pumping theory

All pumps operate by creating a partial vacuum at the intake, and a mechanical force at the outlet that induces flow. This action can be best described by reference to a simple piston pump shown in Fig.1.2.

1. As the piston moves to the left, a partial vacuum is created in the pump chamber thatholds the outlet valve in place against its seat and induces flow from the reservoir that is at a higher (atmospheric) pressure.As this flow is produced, the inlet valve is temporarily displaced by the force of fluid, permitting the flow into the pump chamber (suction stroke).
2. When the piston moves to the right, the resistance at the valves causes an immediate increase in the pressure that forces the inlet valve against its seat and opens the outlet valve thereby permitting the fluid to flow into the system. If the outlet port opens directly to the atmosphere, the only pressure developed is the one required to open the outlet valve(delivery stroke).

1.4Gear Pumps

Gear pumps are less expensive but limited to pressures below 140 bar.It is noisy in operation than either vane or piston pumps. Gear pumps are invariably of fixed displacement type, which means that the amount of fluid displaced for each revolution of the drive shaft is theoretically constant.

1.4.1 External Gear Pumps

External gear pumps are the most popular hydraulic pumps in low-pressure ranges due to their long operating life, high efficiency and low cost. They are generally used ina simple machineThe most common form of external gear pump is shown in Figs. 1.3and 1.4 It consist of a pump housing in which a pair of preciselymachined meshing gears runs with minimal radial and axial clearance.One of the gears, called a driver,is driven by a prime mover. The driver drives another gear called a follower.As the teeth of the two gears separate, the fluid from the pump inlet gets trapped between the rotating gear cavities and pump housing.The trapped fluid is then carried around the periphery of the pump casing and delivered to outlet port. The teeth of precisely meshed gears provide almost a perfect seal between the pumpinlet and the pump outlet.When the outlet flow is resisted, pressure in the pump outlet chamber builds up rapidly and forces the gear diagonally outward against the pump inlet. When the system pressure increases, imbalance occurs. This imbalance increases mechanical friction and the bearing load of the two gears.Hence, the gear pumps are operated to the maximum pressure rating stated by the manufacturer.

It is important to note that the inlet is at the point of separation and the outlet at the point of mesh. These units are not reversible if the internal bleeds for the bearings are to be drilled to both the inlet and outlet sides.So that the manufacturer’s literature should be checked before attempting a reversed installation. If they are not drilled in this manner, the bearing may be permanently damaged as a result of inadequate lubrications.

Advantages and disadvantages of gear pumps

The advantages are as follows:

1.They are self-priming.
2.They give constant delivery for a given speed.
3. They are compact and light in weight.
4. Volumetric efficiency is high.

The disadvantages are as follows:

1. The liquid to be pumped must be clean, otherwise it will damage pump.
2. Variable speed drives are required to change the delivery.
3. If they run dry, parts can be damaged because the fluid to be pumped is used as lubricant.

Expression for the theoretical flow rate of an external gear pump

Let

D_o =the outside diameter of gear teeth

D_i= the inside diameter of gear teeth

L =the width of gear teeth

N=the speed of pump in RPM

V_D=the displacement of pump in m/rev

M= module of gear

z=number of gear teeth

α= pressure angle

Volume displacement is

$$V_D = \frac{\pi}{4}(D_o^2 - D_i^2)L$$

$$D_i = D_o - 2(\text{Addendum} + \text{Dendendum})$$

Theoretical discharge is

$$Q_T\ (\text{m}^3/\text{min}) = V_D\,(\text{m}^3/\text{rev}) \times N\ (\text{rev/min})$$

If the gear is specified by its module and number of teeth, then the theoretical discharge can be found by

$$Q_T = 2\pi L m^2 N\left[z + \left(1 + \frac{\pi^2 \cos^2 20}{12}\right)\right] \text{m}^3/\text{min}$$

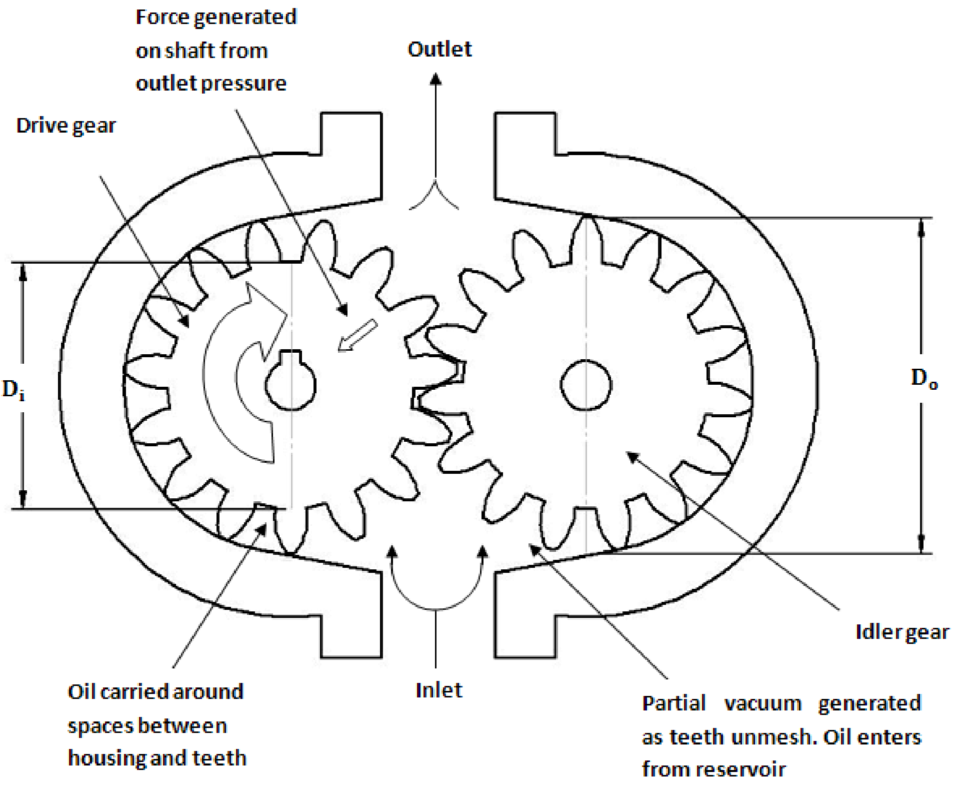


Figure 1.3 Operation of an external gear pump

1.4.2Internal Gear Pumps

Another form of gear pump is the internal gear pump, which is illustrated in Fig. 1.5. They consist of two gears:An external gear and an internal gear. The crescent placed in between these acts as a seal between the suction and discharge.(Fig.1.6) When a pump operates, the external gear drives the internal gear and both gears rotate in the same direction. The fluid fills the cavities formed by the rotating teeth and the stationary crescent. Both the gears transport the fluid through the pump. The crescent seals the low-pressure pump inlet from the high-pressure pump outlet. The fluid volume is directly proportional to the degree of separation and these units may be reversed without difficulty. The major use for this type of pump occurs when a through shaft is necessary, as in an automatic transmission. These pumps have a higher pressure capability than external gear pumps.

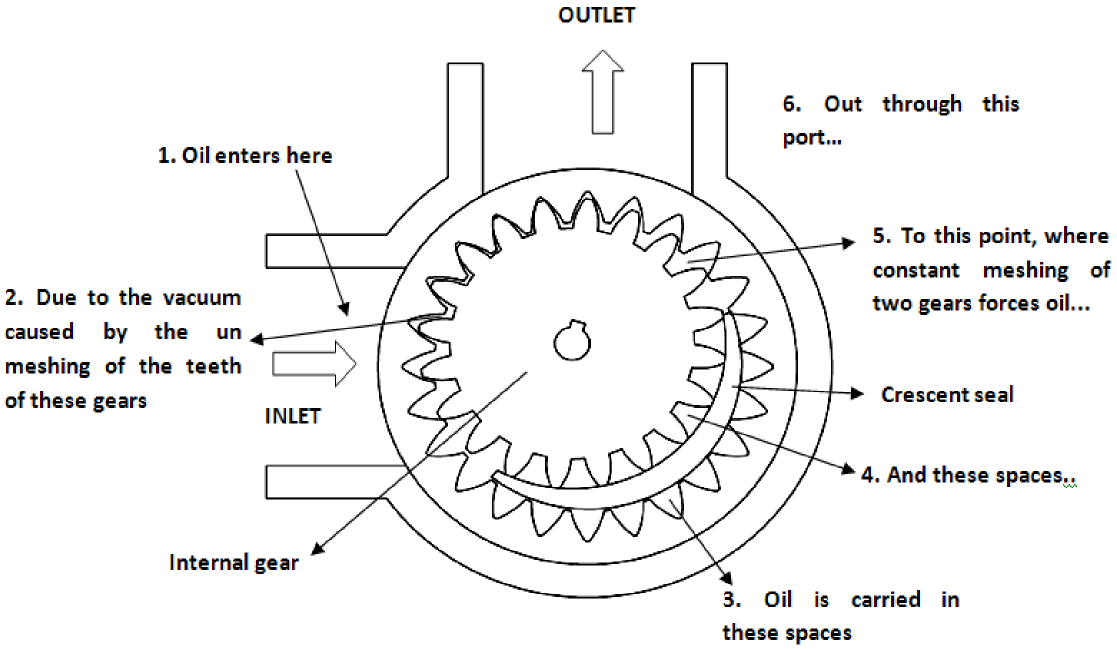


Figure 1.5 Operation of an internal gear pump

1.4.3 Gerotor Pumps

Gerotor pumps operate in the same manner as internal gear pumps. The inner gear rotor is called a gerotor element. The gerotor element is driven by a prime mover and during the operation drives outer gear rotor around as they mesh together. The gerotor has one tooth less than the outer internal idler gear. Each tooth of the gerotor is always in sliding contact with the surface of the outer element. The teeth of the two elements engage at just one place to seal the pumping chambers from each other. On the right-hand side of the pump, shown in Fig. 1.7, pockets of increasing size are formed, while on the opposite side, pockets decrease in size. The pockets of increasing size are suction pockets and those of decreasing size are discharge pockets. Therefore, the intake side of the pump is on the right and discharge side on the left.

Pumping chambers are formed by the adjacent pair of teeth, which are constantly in contact with the outer element, except for clearance. Refer to Fig 1.7,asthe rotor is turned, its gear tips are accurately machined sothat they precisely follow the inner surface of the outer element. The expanding chambers are created as the gear teeth withdraw. The chamber reaches its maximum size when the female tooth of the outer rotor reaches the top dead center. During the second half of the revolution, the spaces collapse, displacing the fluid to the outlet port formed at the side plate. The geometric volume of the gerotor pump is given as

$$V_D = b\, Z\,(A_{max} - A_{min})$$

where b is the tooth height, Z is the number of rotor teeth, A_{max} is the maximum area between male and female gears (unmeshed – occurs at inlet) and A_{min} is the minimum area between male and female gears (meshed – occurs at outlet).

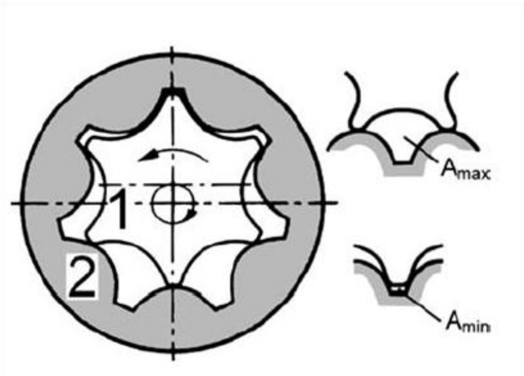


Figure 1.7 Gerotor gear pump

Example 1.1

The inlet to a hydraulic pump is 0.6 m below the top surface of an oil reservoir. If the specific gravity of the oil used is 0.86, determine the static pressure at the pump inlet.

Solution: We know that

$$\text{Pressure} = \rho g h$$

The density of water is 1 g/cm^3 or 1000 kg/m^3.

Therefore, the density of oil is $0.86 \times 1\ \text{g/cm}^3$ or $860\,\text{kg/m}^3$.

Pressure at the pump inlet is

$$P = 860 \times 0.6\,\text{kg/m}^2 = 516\,\text{kg/m}^2 = 0.0516\,\text{kg/cm}^2 = 0.0516 \times 0.981\,\text{bar}$$
$$= 0.0506\,\text{bar}$$

(Note: $1\text{kg/cm}^2 = 0.981\,\text{bar}$.)

Example 1.2

A hydraulic pump delivers 12 L of fluid per minute against a pressure of 200 bar. (a) Calculate the hydraulic power. (b) If the overall pump efficiency is 60%, what size of electric motor would be needed to drive the pump?

Solution:

(a) Hydraulic power is given by

$$\text{Hydraulic power (kW)} = 12\ \text{L/min} \times \frac{200\ \text{(bar)}}{600} = 4\ \text{kW}$$

(b) We have

$$\text{Electric motor power (power input)} = \frac{\text{Hydraulic power}}{\text{Overall efficiency}}$$

Substituting we get

$$\text{Electric motor power (power input)} = \frac{4}{0.6} = 6.67kW$$

$$\text{Electric motor power} = \frac{4}{0.6} = 6.67\,\text{kW}$$

Example 1.3

A gear pump has an outside diameter of 80mm, inside diameter of 55mm and a width of 25mm. If the actual pump flow is 1600 RPM and the rated pressure is 95 LPM what is the volumetric displacement and theoretical discharge.

Solution: We have

Outside diameter $D_o = 80\,\text{mm}$

Inside diameter $D_i = 55\,\text{mm}$

Width $d = 25\text{mm}$

Speed of pump N = 1600 RPM

Actual flow rate = 95 LPM

Now

$Q_A = 95\,\text{LPM} = 95\times10^{-3}\,\text{m}^3/\text{min}$

$$V_D = \frac{\pi}{4}\times(D_o^{\ 2} - D_i^{\ 2})\times L$$

$$V_D = \frac{\pi}{4}\times(0.080^2 - 0.055^2)\times 0.025 = 6.627\times10^{-5}\,m^3/rev$$

Theoretical flow rate

$$Q_T = \frac{\pi}{4}\times(D_o^{\ 2} - D_i^{\ 2})\times L\times N$$

$$= \frac{\pi}{4}\times(0.080^2 - 0.055^2)\times 0.025\times 1600$$

$$= 0.106\,\text{m}^3/\text{min}$$

Example 1.4

Calculate the theoretical delivery of a gear pump. Module of the gear teeth is 6mm and width of gear teeth is 25mm. Number of teeth on driver gear is 18 and pressure angle of the gear is 20°. Pump speed is 1000 RPM. Volumetric efficiency is 90%.

Solution: If the gear is specified by its module and number of teeth, then the theoretical discharge can be found by

$$Q_T = 2\pi L m^2 N\left[z+\left(1+\frac{\pi^2\cos\alpha}{12}\right)\right]\ \mathrm{m^3/min}$$

$$= 2\pi(0.025)(6\times10^{-3})^2\times1000\times\left[18+\left(1+\frac{\pi^2\cos^2 20}{12}\right)\right]\ \mathrm{m^3/min}$$

$$= 0.1118\ \mathrm{m^3/min}$$

Example 1.5

Calculate the theoretical delivery of a gear pump. Module of the gear teeth is 6mm and width of gear teeth is 65mm. Number of teeth on driver gear is 16 and pressure angle of the gear is 20°. Pump speed is 1600 RPM. Outer diameter of gear is 108 mm and Dedendum circle diameter is 81 mm. Volumetric efficiency is 88%at 7 MPa.

Solution: If the gear is specified by its module and number of teeth, then the theoretical discharge can be found by

$$Q_T = 2\pi L m^2 N\left[z+\left(1+\frac{\pi^2\cos^2 20}{12}\right)\right]\ \mathrm{m^3/min}$$

$$= 2\pi(0.065)(6\times10^{-3})^2\times1600\times\left[16+\left(1+\frac{\pi^2 0.939^2}{12}\right)\right]\ \mathrm{m^3/min}$$

$$= 0.416\ \mathrm{m^3/min}$$

Alternatively we can use

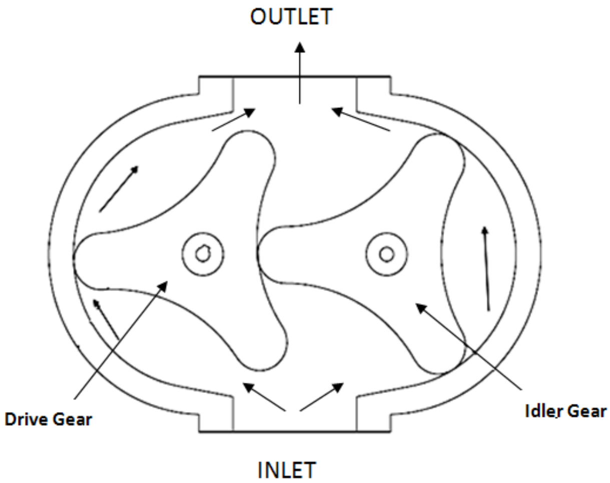
$$V_D = \frac{\pi}{4}\times(D_o^{\,2}-D_i^{\,2})\times L$$

$$Q_T = \frac{\pi}{4}\times(0.108^2-0.081^2)\times0.065\times1600 = 0.416\ \ \mathrm{m^3/rev}$$

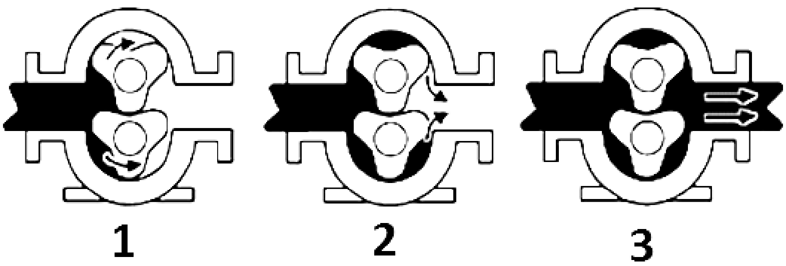
1.5Lobe Pumps

The operation of lobe pump shown in Fig.1.9 is similar to that of external gear pump, but they generally have a higher volumetric capacity per revolution. The output may be slightly greater pulsation because of the smaller number of meshing elements.

Lobe pumps, unlike external gear pumps,have both elements externally driven and neither element hasany contact with the other.For this reason, they are quieter when compared to other types of gear pumps. Lobe contact is prevented by external timing gears located in the gearbox.Pump shaft support bearings are located in the gearbox, and because the bearings are out of the pumped liquid, pressure is limited by bearing location and shaft deflection. They do not lose efficiency with use. They are similar to external gear pumps with respect to the feature of reversibility.



Stages of operation of Lobe pump



1.As the lobes come out of mesh, they create expanding volume on the inlet side of the pump.Liquid flows into the cavity and is trapped by the lobes as they rotate.

2.Liquid travels around the interior of the casing in pockets between the lobes and the casing (it does not pass between the lobes).

3.Finally, the meshing of the lobes forces the liquid through the outlet port under pressure.

Lobe pumps are frequently used in food applications because they are good at handling solids without inflicting damage to the product. Solid particle size can be much larger in lobe pumps than in other positive displacement types.Because lobes do not make contact, and clearances are not as close as in other positive displacement pumps, this design handles low-viscosity liquids with diminished performance.Loading characteristics are not as good as other designs and suction ability is low.High-viscosity liquids require reduced speeds to achieve satisfactory performance.Reductions of 25% of rated speed and lower are common with high-viscosity liquids.

1.5.1Advantages

The advantages of lobe pumps are as follows:

1. Lobe pumps can handle solids, slurries, pastes and many liquid.
2. No metal-to-metal contact.
3. Superior CIP(Cleaning in Place) /SIP(Sterilization in Place) capabilities.
4. Long-term dry run (with lubrication to seals).
5. Non-pulsating discharge.

1.5.2Disadvantages

The disadvantages of lobe pumps are as follows:

1. Require timing gears.
2. Require two seals.
3. Reduced lift with thin liquids.

1.5.3Applications

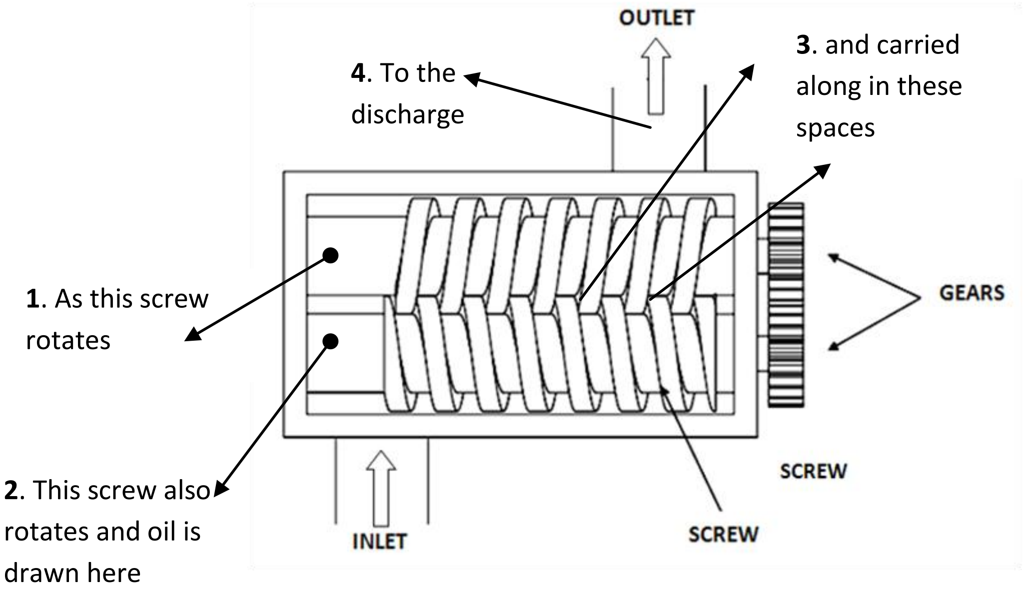
Common rotary lobe pump applications include, but are not limited to, the following:

1. Polymers.
2. Paper coatings.
3. Soaps and surfactants.
4. Paints and dyes.
5. Rubber and adhesives.
6. Pharmaceuticals.
7. Food applications.

1.6Screw Pumps

These pumps have two or more gear-driven helical meshing screws in a closefitting caseto develop the desired pressure. These screws mesh to form a fluid-type seal between the screws and casing.

A schematic diagram of a screw pump is shown in Fig 1.10. A two-screw pump consists of two parallel rotors with inter-meshing threads rotating in a closely machined casing. The driving screw and driven screw are connected by means of timing gears. When the screws turn, the space between the threads is divided into compartments. As the screws rotate, the inlet side of the pump is flooded with hydraulic fluid because of partial vacuum.When the screws turn in normal rotation, the fluid contained in these compartments is pushed uniformly along the axis toward the center of the pump, where the compartments discharge the fluid. Here the fluid does not rotate but moves linearly as a nut on threads. Thus, there are no pulsations at a higher speed; it is a very quiet operating pump.



Ina screw pump, a chamber is formed between thread and housing as shown in Fig.1.11. The following expression gives the volumetric displacement

$$V_{\mathrm{D}}=\frac{\pi}{4}(D^2-d^2)s-D^2\left\{\frac{\alpha}{2}-\frac{\sin 2\alpha}{2}\right\}s$$

Heres is the stroke length and $\cos(\alpha)=\frac{D+d}{2D}$

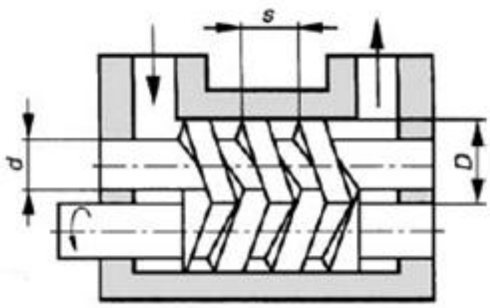


Figure 1.11 Volumetric displacement of a screw pump

Advantages and disadvantages of screw pump

Theadvantages are as follows:

1.They are self-priming and more reliable.
2. They are quiet due to rolling action of screw spindles.
3.They can handle liquids containing gases and vapor.
4. They have long service life.

The disadvantages are as follows:

1.They are bulky and heavy.
2.They are sensitive to viscosity changes of the fluid.
3. They have low volumetric and mechanical efficiencies.
4. Manufacturing cost of precision screw is high.

Lecture 8

HYDRAULIC PUMPS [CONTINUED]

1.7 Vane Pumps

There are two types of vane pumps:

1. **Unbalanced vane pump:** Unbalanced vane pumps are of two varieties:

- Unbalanced vane pump with fixed delivery.
- Unbalanced vane pump with pressure-compensated variable delivery.

2. **Balanced vane pump.**

1.7.1 Unbalanced Vane Pump with Fixed Delivery

A simplified form of unbalanced vane pump with fixed delivery and its operation are shown in Figs. 1.12 and 1.13. The main components of the pump are the cam surface and the rotor. The rotor contains radial slots splined to drive shaft. The rotor rotates inside the cam ring. Each radial slot contains a vane, which is free to slide in or out of the slots due to centrifugal force. The vane is designed to mate with surface of the cam ring as the rotor turns. The cam ring axis is offset to the drive shaft axis. When the rotor rotates, the centrifugal force pushes the vanes out against the surface of the cam ring. The vanes divide the space between the rotor and the cam ring into a series of small chambers. During the first half of the rotor rotation, the volume of these chambers increases, thereby causing a reduction of pressure. This is the suction process, which causes the fluid to flow through the inlet port. During the second half of rotor rotation, the cam ring pushes the vanes back into the slots and the trapped volume is reduced. This positively ejects the trapped fluid through the outlet port. In this pump, all pump action takes place in the chambers located on one side of the rotor and shaft, and so the pump is of an unbalanced design. The delivery rate of the pump depends on the eccentricity of the rotor with respect to the cam ring.

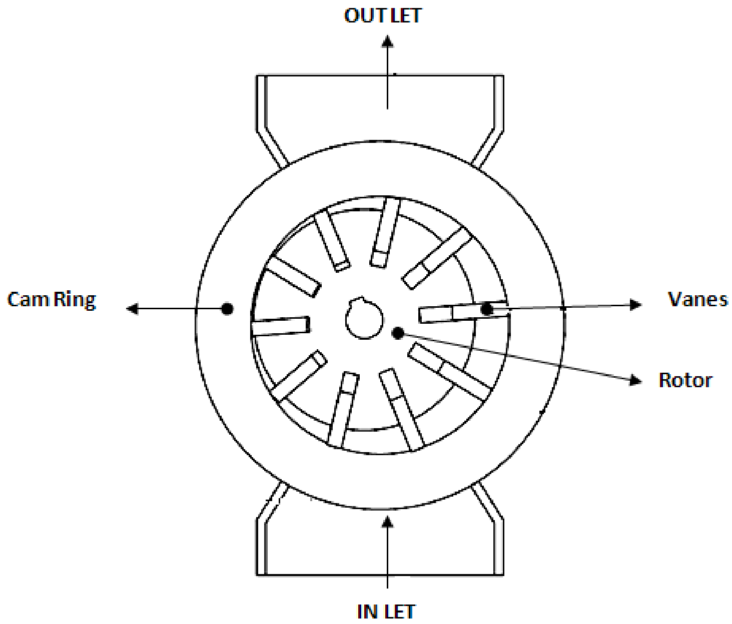


Figure 1.12 Simple vane pump

1.7.4 Pressure-Compensated Variable Displacement Vane Pump (an Unbalanced Vane Pump with Pressure-Compensated Variable Delivery)

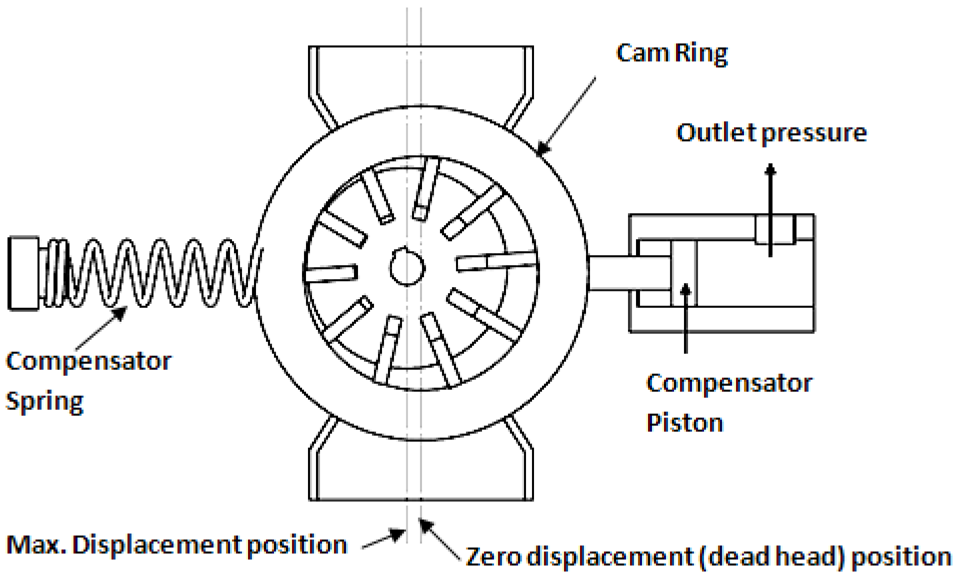


Figure 1.14 Operation of a variable displacement vane pump

Schematic diagram of variable displacement vane pump is shown in Fig.1.14.Variable displacement feature can be brought into vane pumps by varying eccentricity between the rotor and the cam ring. Here in this pump, the stator ring is held against a spring loaded piston.The system pressure acts directly through a hydraulic piston on the right side. This forces the cam ring against a spring-loaded piston on the left side. If the discharge pressure is large enough, it overcomes the compensated spring force and shifts the cam ring to the left.

This reduces the eccentricity and decreases the flow. If the pressure continues to increase, there is no eccentricity and pump flow becomes zero.

1.7.5 Balanced Vane Pump with Fixed Delivery

A balanced vane pump is a very versatile design that has found widespread use in both industrial and mobile applications. The basic design principle is shown in Fig. 1.15. The rotor and vanes are contained within a double eccentric cam ring and there are two inlet segments and two outlet segments during each revolution. This double pumping action not only gives a compact design, but also leads to another important advantage: although pressure forces acting on the rotor in the outlet area are high, the forces at the two outlet areas are equal and opposite, completely canceling each other. As a result, there are no net loads on shaft bearings. Consequently, the life of this type of pump in many applications has been exceptionally good. Operating times of 24000 h or more in industrial applications are widespread. In more severe conditions encountered in mobile vehicles, 5000–10000 h of trouble-free operation is frequently achieved.

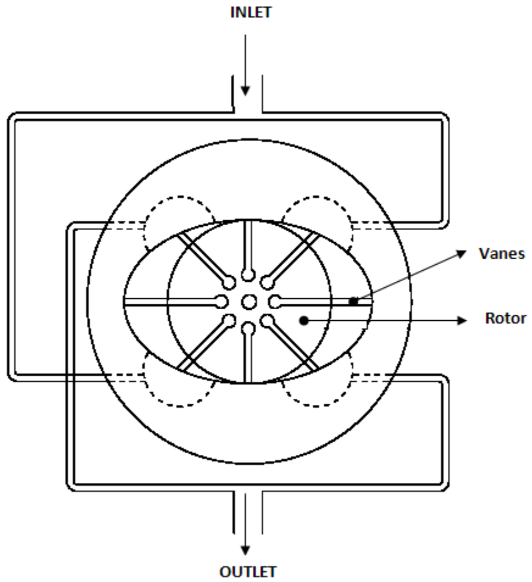


Figure 1.15 Operation of a balanced vane pump

1.7.2 Advantages and disadvantages of Vane Pumps

The advantages of vane pumps are as follows:

1. Vane pumps are self-priming, robust and supply constant delivery at a given speed.
2. They provide uniform discharge with negligible pulsations.
3. Their vanes are self-compensating for wear and vanes can be replaced easily.
4. These pumps do not require check valves.
5. They are light in weight and compact.
6. They can handle liquids containing vapors and gases.
7. Volumetric and overall efficiencies are high.
8. Discharge is less sensitive to changes in viscosity and pressure variations.

The disadvantages of vane pumps are as follows:

1. Relief valves are required to protect the pump in case of sudden closure of delivery.
2. They are not suitable for abrasive liquids.
3. They require good seals.
4. They require good filtration systems and foreign particle can severely damage pump.

Advantages and disadvantagesofbalancedvane pumps

The advantages of balanced vane pumps are as follows:

1. The balanced pump eliminates the bearing side loads and therefore high operating pressure can be used.
2.The service life is high compared to unbalanced type due to less wear and tear.

The disadvantages of balanced vane pumps are as follows:

1. They are fixed displacement pumps.
2. Design is more complicated.
3. Manufacturing cost is high compared to unbalanced type.

1.7.4 Expression for the Theoretical Discharge of Vane Pumps

Let D_C be the diameter ofa cam ring in m, $D_R$the diameter of rotor in m, L the width of rotor in m, e the eccentricity in m, V_D the pump volume displacement in m^3/rev and e_{max} the maximum possible eccentricity in m.

From geometry (Fig.1.13) the maximum possible eccentricity,

$$e_{max} = \frac{D_C - D_R}{2} \qquad (1.1)$$

Themaximum value of eccentricity produces the maximum volumetric displacement

$$V_{D(max)} = \frac{\pi}{4}(D_C^2 - D_R^2)L \qquad (1.2)$$

Using Equation (1.1), Equation (1.2) can be simplified as

$$V_{D(max)} = \frac{\pi}{4}(D_c - D_g)(D_c + D_g)L$$

<

$$V_{D(max)} = \frac{\pi}{4}(D_c + D_g)(2e_{max})L$$

The actual volumetric displacement occurs when $e_{max} = e$. Hence,

$$V_{D(max)} = \frac{\pi}{2}(D_C + D_R)e\,L \text{ m}^3\text{/rev}$$

When the pump rotates at N rev/min (RPM),the quality of discharge by the vane pump is given by

$$Q_T = v_D \times N$$

Theoretical discharge

$$Q_T = \frac{\pi}{2}(D_C + D_R)e\,L \ \ \mathrm{m^3/min}$$

Example 1.6

A vane pump has a rotor diameter of 63.5 mm, a cam ring diameter of 88.9 mm and a vane width of 50.8 mm. What must be eccentricity for it to have a volumetric displacement of 115cm^3?

Solution: Volumetric displacement is

$$V_D = \pi\left(\frac{D_C + D_R}{2}\right)L\,e$$

where$D_C$is the diameter of the cam ring, D_R is the diameter of the rotor, e is the eccentricity and L is the width of the vane pump. So we have

$$115\times10^{-6} = \pi\times\frac{0.0889+0.0635}{2}\times e\times0.0508$$

Therefore eccentricity

$$e = 9.456\times10^{-3}\ \mathrm{m} = 9.456\ \mathrm{mm}$$

1.8Piston Pumps

Piston pumps are of the following two types:

1. **Axial piston pump:**These pumps are of two designs:

   - Bent-axis-type piston pump.
   - Swash-plate-type piston pump.

2. **Radial piston pump.**

1.8.1Bent-Axis-Type Piston Pump

Schematic diagram and detailed cut section of bent axis type piston pump is shown in Fig.1.16. It contains a cylinder block rotating with a drive shaft. However,thecenterline of the cylinder block is set at an offset angle relative to the centerline of the drive shaft.Thecylinder block contains a number of pistons arranged along a circle.The piston rods are connected to the drive shaft flange by a ball and socket joints. The pistons are forced in and out of their bores as the distance between the drive shaft flange and cylinder block changes. A universal link connects the cylinder block to the drive shaft to provide alignment and positive drive.

The volumetric displacement of the pump depends on the offset angle θ. No flow is produced when the cylinder blockis centerline.θcan vary from 0°to a maximum of about 30°.For a fixed displacement, units are usually provided with 23° or 30° offset angles.

1.8.2 Swash-Plate-Type Piston Pump

Schematic diagram of swash plate type piston pump is shown in Fig. 1.17. In this type, the cylinder block and drive shaft are located on the same centerline. The pistons are connected to a shoe plate that bears against an angled swash plate. As the cylinderrotates,the pistons reciprocatebecausethe piston shoesfollow the angled surface of the swash plate. The outlet andinlet ports are located in the valve plate so that the pistons pass theinlet as they are being pulledout and pass the outlet as they are being forced back in. This type of pump can also be designed to have a variable displacementcapability. The maximumswash plate angle is limited to 17.5° by construction.

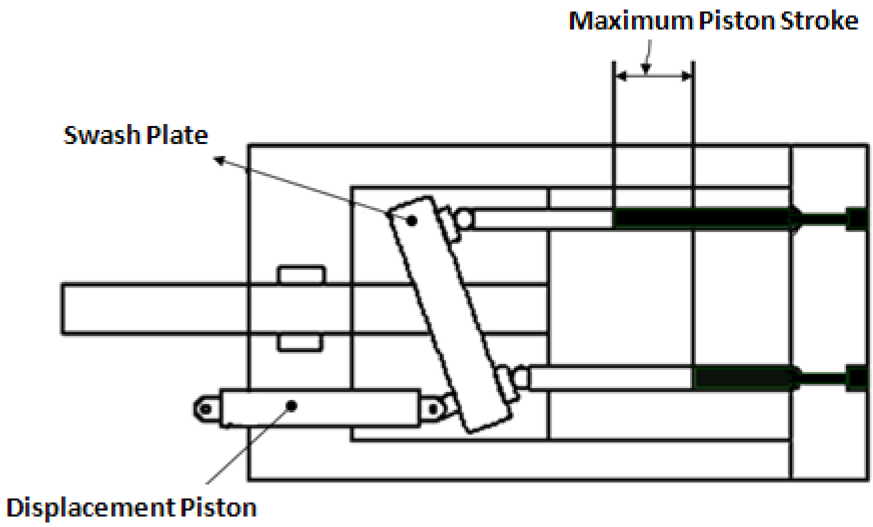
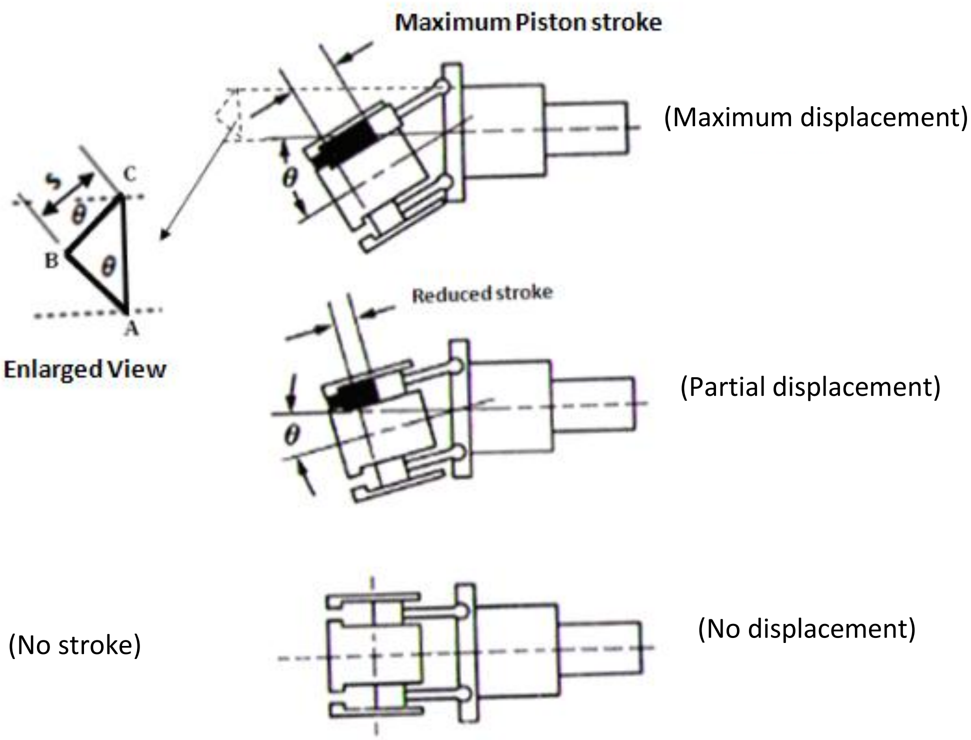


Figure 1.17 Operation of a swash-plate-type piston pump

1.8.4 Volumetric Displacement and Theoretical Flow Rate of an Axial Piston Pump

Figure 1.19(a) shows in and out position of the pistons of axial piston pump.Figure1.19(b) gives schematic diagram of stroke change with respect to offset angle.

Letθ be an offset angle, S the piston stroke in m, D the piston circle diameter, Y the number of pistons, A the piston area inm^2, N the piston speed in RPM and Q_T the theoretical flow rate in m^3/min.



(b)

Figure 1.19 Stroke changes with offset angle

From a right-angled triangle ABC [Fig. 1.19(b)]

$$\tan\theta = \frac{\mathrm{BC}}{\mathrm{AB}} = \frac{S}{D}$$

$$\Rightarrow S = D \times \tan\theta \qquad (1.3)$$

The displacement volume of one piston = $AS\mathrm{m}^3$

Total displacement volume of Ynumber of pistons = $YAS\mathrm{m}^3$

$$V_{\mathrm{D}} = YAS \qquad (1.4)$$

From Eqs. (1.3) and (1.4), we have

$$V_{\mathrm{D}} = YAD\tan\theta \ \mathrm{m}^3/\mathrm{rev} \qquad (1.5)$$

Theoretical flow rate is

$$Q_{\mathrm{T}} = DANY\tan\theta \ \mathrm{m}^3/\mathrm{min}$$

Example 1.7

What is the theoretical flow rate from a fixed-displacement axial piston pump with a nine-bore cylinder operating at 2000 RPM? Each bore has a diameter of 15 mm and stroke is 20 mm.

Solution: Theoretical flow rate is given by

$$Q_{\mathrm{T}} = \text{Volume} \times \text{RPM} \times \text{Number of pistons}$$

$$= \frac{\pi}{4} \times D^2 \times L \times N \times n$$

$$= \frac{\pi}{4} \times 0.015^2 \times 0.02 \times \frac{2000}{60} \times 9$$

$$= 10.6 \times 10^{-3} \mathrm{m}^3/\mathrm{s}$$

$$= 1.06 \text{ LPS} = 63.6 \text{ LPM}$$

1.9Comparison of Hydraulic Pumps

Pump design with a wide range of operating characteristics are available. A designer must select carefully to achieve a circuit design that meets the functional objective while minimizing total cost which includes both ownership cost and operating cost over the life of component. Pump selection is important decision in circuit design. Designer must compare the various options available and then choose the optimum pump. Table 1.2 gives a typical comparison of all pumps.

The major factor in adopting a pump to a particular system is the system's overall needs. It would be wrong to use a pump with high delivery in a system that requires only a lowdelivery rate. On the contrary, using a pump that must produce at its peak continuously just to meet the minimum requirements of the system is equally wrong. Making either of these mistakes produces a poor system due to excessive initial pump costs or maintenance cost.

One should use a pump that is suited to the system, whether a gear pump which has fewer moving precision parts or a piston pump whichhas many parts fitted to close tolerance and is therefore more expensive.

Table 1.2

	Pressure (Bar)	**Discharge(LPM)**	**MaximumSpeed (RPM)**	**Overall Efficiency**
Gear pump	20–175	7–570	1800–7000	75–90
Vane pump	20–175	2–950	2000–4000	75–90
Axial piston pump	70–350	2–1700	600–6000	85–95
Radial piston pump	50–250	20–700	600–1800	80–92

1.10Pump Performance

The performance of a pump is a function of the precision of its manufacture. An ideal pump is one having zero clearance between all mating parts. Because this is not possible, working clearances should be as small as possible while maintaining proper oil films for lubrication between rubbing parts. The performance of a pump is determined by the following efficiencies:

1. Volumetric efficiency (η_v): It is the ratio of actual flow rate of the pump to the theoretical flow rate of the pump. This is expressed as follows:

$$\text{Volumetric efficiency } (\eta_v) = \frac{\text{Actual flow rate of the pump}}{\text{Theoretical flow rate of the pump}}$$

$$= \frac{Q_A}{Q_T}$$

Volumetric efficiency (η_v) indicates the amount of leakage that takes place within the pump.This is due to manufacture tolerances and flexing of the pump casing under designed pressure operating conditions.

For gear pumps, η_v = 80%–90%.

For vane pumps, η_v = 92%.

For piston pumps, η_v = 90%–98%.

2. Mechanical efficiency (η_m): Itis the ratio of the pump output power assuming no leakage to actual power delivered to the pump:

$$\text{Mechanical efficiency } (\eta_m) = \frac{\text{Pump output power assuming no leakages}}{\text{Actual power delivered to the pump}}$$

Mechanicalefficiency(η_m) indicates the amount of energy losses that occur for reasons other than leakage. This includes friction in bearings and between mating parts. This includes the energy losses due to fluid turbulence. Mechanical efficiencies are about 90%–95%. We also have the relation

$$\eta_m = \frac{p\,Q_T}{T_A N}$$

where$p$ is the pump discharge pressure in Pa or N/m^2, Q_T is the theoretical flow rate of the pump in m^3/s, T_A is the actual torque delivered to the pump in Nm and N is the speed of the pump in rad/s.

It (η_m) can also be computed in terms of torque as follows:

$$\eta_m = \frac{\text{Theoretical torque required to operate the pump}}{\text{Actual torque delivered to the pump}}$$

$$= \frac{T_T}{T_A}$$

The theoretical torque (T_T) required to operate the pump is the torque that would be required if there were no leakage.

The theoretical torque (T_T) is determined as follows

$$T_T(\text{N m}) = \frac{VD_N}{2\pi}\left(\text{m}^3 \times \frac{N}{\text{m}^2}\right) = \text{N m}$$

The actual torque (T_A) is determined as follows

$$\text{Actual torque } T_A \text{ (N m)} = \frac{P}{\omega}\left(\frac{\text{N m/s}}{\text{rad/s}}\right) = \text{N m}$$

where $\omega = 2\pi N/60$. Here N is the speed in RPM.

3. Overall efficiency (η_o)**:** It is defined as the ratio of actual power delivered by the pump to actual power delivered to the pump

$$\text{Overall efficiency } (\eta_o) = \frac{\text{Actual power delivered by the pump}}{\text{Actual power delivered to the pump}}$$

Overall efficiency (η_o) considers all energy losses and can be represented mathematically as follows:

$$\text{Overall efficiency } (\eta_o) = \eta_v \eta_m$$

$$\Rightarrow \eta_o = \frac{Q_A}{Q_T} \times \frac{pQ_T}{T_A N}$$

Example 1.8

A gear pump has an outside diameter of 82.6 mm, inside diameter of 57.2 mm and a width of 25.4 mm. If the actual pump flow is 1800 RPM and the rated pressure is 0.00183 m^3/s, what is the volumetric efficiency?

Solution: We have

Outside diameter $D_o = 82.6\text{ mm}$

Inside diameter $D_i = 57.2\text{ mm}$

Width $d = 25.4\text{ mm}$

Speed of pump N = 1800 RPM

Actual flow rate = 0.00183 m^3/s

Theoretical flow rate

$$Q_T = \frac{\pi}{4} \times (D_o{}^2 - D_i{}^2) \times d \times \frac{N}{60}$$

$$= \frac{\pi}{4} \times (0.0826^2 - 0.0572^2) \times 0.0254 \times \frac{1800}{60}$$

$$= 2.125 \times 10^{-3}$$

Volumetric efficiency is

$$\eta_v = \frac{0.00183}{2.125 \times 10^{-3}} \times 100 = 86.11\%$$

Example 1.9

A pump having a volumetric efficiency of 96% delivers 29 LPM of oil at 1000 RPM. What is the volumetric displacement of the pump?

Solution:

Volumetric efficiency of the pump η_v = 96%
Discharge of the pump = 29 LPM
Speed of pump $N = 1000$ rpm
Now

$$\eta_v = \frac{\text{Actual flow rate of the pump}}{\text{Theoritical flow rate of the pump}} = \frac{Q_A}{Q_T}$$

$$\Rightarrow 0.96 = \frac{29}{Q_T}$$

$$\Rightarrow Q_T = 30.208 \text{ LPM}$$

Volumetric displacement

$$V_D = \frac{Q_T}{N} = \frac{30.208 \times 10^{-3} \times 60}{60 \times 1000}$$

$$= 30.208 \times 10^{-6} \text{ m}^3/\text{rev} = 0.0302 \text{ L/rev}$$

Example 1.10

A positive displacement pump has an overall efficiency of 88% and a volumetric efficiency of 92%. What is the mechanical efficiency?

Solution: Theoverall efficiency is

$$\eta_o = \eta_m \times \eta_v$$

$$\Rightarrow \eta_m = \frac{\eta_o}{\eta_v} = \frac{88}{92} \times 100 = 95.7\%$$

Example 1.11

Determine the overall efficiency of a pump driven by a 10 HP prime mover if the pump delivers fluid at 40 LPM at a pressure of 10 MPa.

Solution:

$$\text{Output power} = pQ$$

$$= 10 \times 10^6 \text{ N/m}^2 \times 40 \text{ L/min} \times \frac{\text{m}^3/\text{s}}{1000 \text{ L/s}} \times \frac{1 \text{ min}}{60 \text{ s}}$$

$$= 6670 \text{ W}$$

$$\text{Input power} = 10 \text{ HP} \times \frac{746 \text{ W}}{1 \text{ HP}} = 7460 \text{ W}$$

Now

$$\eta_o = \frac{\text{Pump output power}}{\text{Pump input power}}$$

$$= \frac{6670}{7460} = 0.894 = 89.4\%$$

Example 1.12

How much hydraulic power would a pump produce when operating at 140 bar and delivering 0.001 m^3/s of oil? What power rated electric motor would be selected to drive this pump if its overall efficiency is 85%?

Solution:
Operating pressure of the pump = 140 bar
Flow rate Q = 0.001m^3/s. Now

$$\text{Power of pump} = \text{Pressure} \times \text{Flow rate}$$
$$= 140 \times 10^5 \times 0.001$$
$$= 14 \text{ kW}$$

Overall efficiency of pump $\eta_o = 85\%$
Power to be supplied is

$$\frac{\text{Power of pump}}{\eta_o} = \frac{14\,\text{kW}}{0.85} = 16.47\,\text{kW}$$

Example 1.13
A pump has a displacement volume of 98.4 cm^3. It delivers 0.0152 m^3/s of oil at 1000 RPM and 70 bar. If the prime mover input torque is 124.3 Nm. What is the overall efficiency of pump? What is the theoretical torque required to operate the pump?

Solution:
Volumetric discharge $= 98.4\,\text{cm}^3$
Theoretical discharge is

$$Q_T = V_D \times \frac{N}{60} = 98.4 \times \frac{1000}{60} = 1.64 \times 10^{-3}\ \text{m}^3/\text{s}$$

Volumetric efficiency is

$$\eta_v = \frac{1.52 \times 10^{-3}}{1.64 \times 10^{-3}} \times 100 = 92.68\ \%$$

Overall efficiency is

$$\eta_o = \frac{Q_A \times \text{pressure}}{T \times \omega} = \frac{1.52 \times 10^{-3} \times 70 \times 10^5 \times 60}{124.3 \times 2 \times 1000 \times \pi} \times 100 = 81.74\%$$

The mechanical efficiency is

$$\eta_{\text{mechanical}} = \frac{\eta_{\text{overall}}}{\eta_{\text{volumetric}}} = \frac{81.74}{92.78} = 88.2$$

Now

Theoretical torque = Actual torque × $\eta_{\text{mechanical}}$ = 124.3 × 0.882 = 109.6 Nm

Note: Mechanical efficiency can also be calculated as

$$\eta_m = \frac{pQ_T}{T\omega}$$
$$= \frac{70 \times 10^5\ \text{N/m}^2 \times 0.00164\ \text{m}^3/\text{s}}{124.3\ (\text{N m}) \times \frac{1000}{60} \times 2\pi\ \text{rad/s}}$$
$$= 0.882 = 88.2\%$$

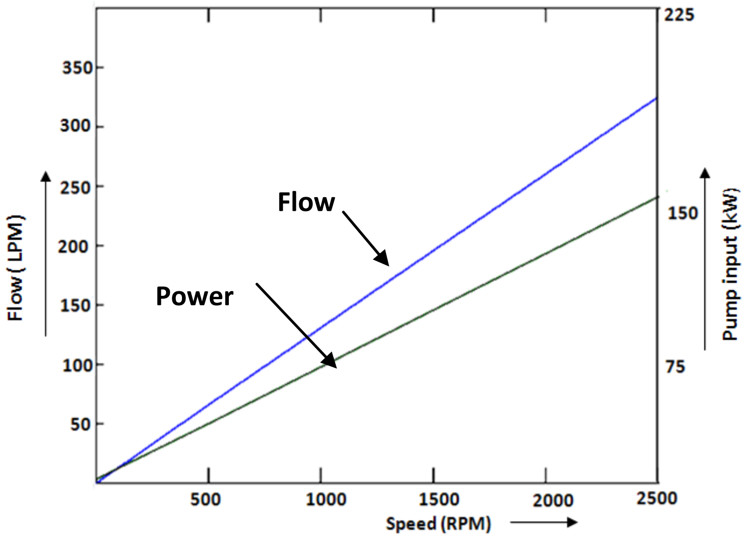
Lecture 9

HYDRAULIC PUMPS [CONTINUED]

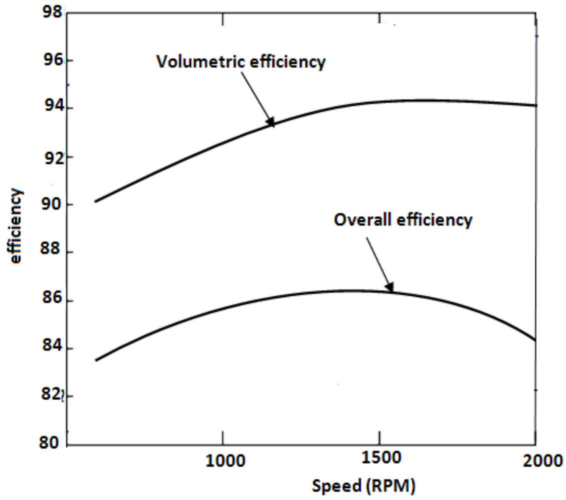
1.11 Pump Performance Curve

Pump performance characteristics are first analyzed independently of the rest of hydraulic system and then as a part of the system. Both sets of data are valuable to the designer. Analyzing the pump by itself gives an indication of its capabilities and performance based on the speed of rotation, internal geometry, cost factors, etc., whereas analyzing pump performance in system essentially determines pump system compatibility. In the first case, the system designer may observe performance curves to see if a specific pump has the pressure and volume flow rate to operate a given set of actuators. In a second instance, the system designer may be computing the noise, vibration, cavitation and flow characteristics of a specific pump before or after installation to determine if the pump and existing system are compatible. Where the two are necessarily complimentary, in practice much of hands-on work is completed independently. Pump performance characteristics are interpreted from data in tabular form and then graphed.

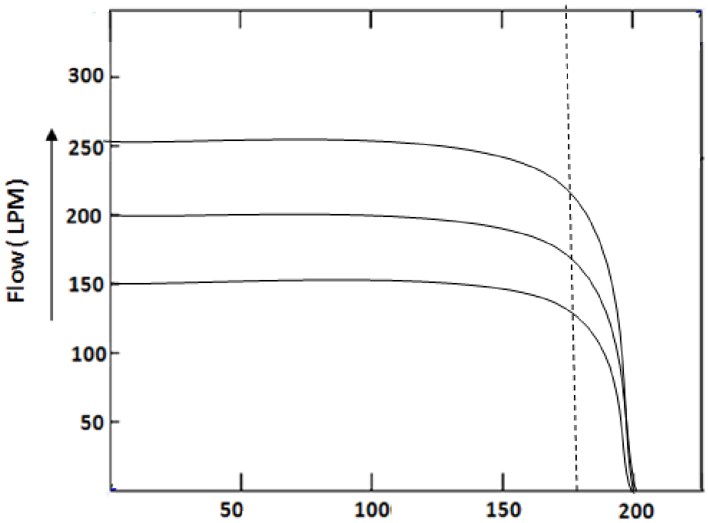
Figure 1.20 shows a graphical representation of a typical positive displacement pump. Figure 1.20(a) represents the relationship between input power and pump output flow of a variable displacement piston pump as a function of pump speed.Observe the linear relationship between the discharge flow and pump speed. Figure 1.20(b) gives curves of overall and volumetric efficiencies as a function of speed. Performance curves of radial piston pump are given in Fig.1.20(c). Discharge flow of these pumps is nearly constant over a broad pressure range.Discharge flow can be varied infinitely between the point of inflection on the constant discharge portion of the curve and zero flow.



(a)



(b)



(c)

Figure 1.20 Pump performance curves

1.12Pump Noise

Pump noise is an important parameter used to determine the performance. Any increase in noise indicates increased wear and eventually pump failure. Pumps are good generators but poor radiators of noise. Noise is not just the sound coming directly from the pump, but also from the vibration and fluid pulsation produced by the pump. Pumps are small in size and hence, they are poor radiators of noise. Reservoirs, electric motors and piping being largerin size are better radiators. Hence,a pump-induced vibration can cause audible noise greater than that coming from the pump. Fixed displacement pumps are less noisy than variable displacement pumps because of their rigid construction.

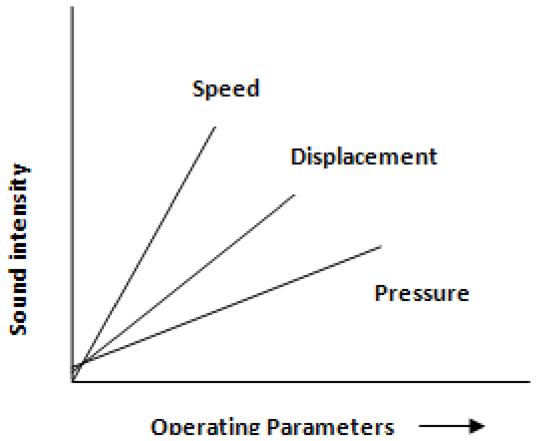


Figure 1.21 Pumpnoise characteristics

As can be seen from Fig.1.21, the pump speed has a strong effect on noise compared to displacement and pressure. To reduce the noise levels, electric motors are used and the most advantageous combination of size and pressure is selected to produce the needed power.

1.13Pump Cavitation

During the working of a positive displacement pump, vacuum is created at the inlet of the pump. This allows atmospheric pressure to push the fluid in. In some situations, the vacuum may become excessive, and a phenomenon known as cavitation occurs. When the pressure of the liquid reaches a low enough level, it vaporizes or boils. Cavitation is the formation of oil vapor bubbles due to a very low pressure (high vacuum) on the inside of the pump. The low pressure also causes air, which is dissolved in the oil to come out of the solution and form bubbles. These air and oil vapor bubbles collapse when they reach the outlet side of the pump, which is under a high pressure. The collapsing of these vapor bubbles causes extremely high localized pressure and fluid velocity. These pressures are so high that they cause pitting of metal and consequently decrease the life and efficiency of the pump.

1.13.1 Factors Causing Cavitation

Cavitation is caused by the following factors:

1. Undersized plumbing.
2. Clogged lines or suction filters.
3. High fluid viscosity.
4. Too much elevation head between the reservoir and the pump inlet.

1.13.2 Rules to Eliminate (Control) Cavitation

Following are the rules to control cavitation:

1. Keep suction line velocities below 1.2 m/s.
2. Keep the pump inlet lines as short as possible.
3. Minimize the number of fittings in the inlet line.
4. Mount the pump as close as possible to the reservoir.
5. Use low-pressure drop inlet filters.
6. Use proper oil as recommended by the pump manufacturer.

Example 1.14

Calculate the pipe bores required for the suction and pressure lines of a pump delivering 40 L/min using a maximum flow velocity in the suction line of 1.2 m/s and a maximum flow velocity in the pressure line of 3.5 m/s.

Solution: Consider the suction line

$$\text{Flow} = \text{Average velocity} \times \text{Flow area}$$

$$\text{Area of pipe} = \frac{\text{Flow through pipe}}{\text{Velocity of flow}}$$

Now

$$\text{Flow} = 40 \text{ LPM} = 40/60 \text{ LPS} = 40/60 \times 10^{-3}\,\text{m}^3/\text{s}$$

$$\text{Area of pipe} = \frac{40 \times 10^{-3}}{60 \times 1.2} = 0.555 \times 10^{-3}\,\text{m}^2$$

Let the bore of the pipe be of diameter D. Then

$$\text{Area of pipe } = 0.555 \times 10^{-3}\text{ m}^2 = \frac{\pi D^2}{4}$$

$$\Rightarrow D = 0.0266 \text{ m}$$

Minimum bore suction pipe = 26.6 mm.

Note: in all calculations great care must be taken to ensure that units are correct.

Alternatively, if a flow velocity of 1m/s is used then suction pipe bore can be of diameter 29 mm. The required diameter of the pressure line can be calculated in a similar manner taking the flow velocity as 3.5 m/s. Here the minimum bore of pressure pipe is equal to 15.6 mm.

It is unlikely that a pipe having the exact bore is available, in which case select a standard pipe having a larger bore. Alternatively, a smaller bore pipe may be chosen but it will be necessary to recheck the calculation to ensure that the flow velocity falls within the recommended range. That is, a standard pipe with an outside diameter of 20 mm and a wall thickness of 2.5 mm is available. This gives an internal diameter of 15 mm.

$$\text{Flow velocity} = \frac{\text{Flow through pipe}}{\text{Area of pipe bore}}$$

Now

$$\text{Area of pipe bore} = \frac{\pi}{4} \times 15^2\text{ mm}^2 = 177\text{ mm}^2 = 177 \times 10^{-6}\text{ m}^2$$

So

$$\text{Flow velocity} = \frac{60 \times 10^{-3}}{60 \times 177 \times 10^{-6}} = 3.77\text{ m/s}$$

This is satisfactory. It is also important to ensure that the wall thickness of pipe is sufficient to withstand the working pressure of the fluid.

1.14Pump Selection

The main parameters affecting the selection of a particular type of pump are as follows:

1. Maximum operating pressure.
2. Maximum delivery.
3. Type of control.
4. Pump drive speed.
5. Type of fluid.
6. Pump contamination tolerance.

7. Pump noise.
8. Size and weight of a pump.
9. Pump efficiency.
10. Cost.
11. Availability and interchangeability.
12. Maintenance and spares.

1.14.1 Maximum Operating Pressure

This is determined by the power requirement of the circuit, the particular application,availability of components, type of fluid and to some extent the environment and level of labor both using and maintaining the equipment.

In general, the higher the operating pressure, the higher the component cost and the lower the choice of components. The main advantage of higher working pressures is the reduction in fluid flow rates for a given system power, resulting in smaller pumps, smaller bore pipes and smaller components. The disadvantage is that at higher working pressures, the compressibility of the fluid used can have considerable adverse effects where precision control is required over a wide range of loads.

The general tendency is toward increased operating pressures. Typical maximum pressures for fewapplications are given in Table 1.3. The operating pressures of pumps depend to some extent on the fluid used. A fire-resistant fluid is generally not as good lubricant as a mineral oil. So to give a reasonable pump life expectancy when using a fire-resistant fluid, the maximum operating pressure must be reduced and it is advisable to consult the pump manufacturer.

The maximum operating pressure and range of flow rates for different types of currently available hydraulic pumps are shown in Table 1.4.The figures given cover a range of sizes and makes; maximum valuesof delivery and pressure are not applicable to one pump.

Table 1.3 System maximum pressure in relation to application

Application	**Pressure (bar)**
Machine tool	200
Mechanical handling	250
Mobile	300
Press work	800

Table 1.4 Operating pressure and size ranges for hydraulic pump types

Pump Type	Maximum Pressure		Maximum Delivery (L/min)		Speed(RPM)		Min. Filtration(μm)	Pulsation	Noise Level(dB)	η (%)
	From	To	From	To	From	To				
External gear	40	300	0.25	760	500	3000	100	High	90	70–90
Internal gear	100	210	0.6	740	3000	4000	100	Low	85	75–90
Vane	50	140	6	360	500	3000	50	Low	80	65–80
Balanced vane	140	175	2	620	500	300	50	Low	85	70–90
Axial piston (swash plate)	200	350	1	1450	200	2000	25	High	90	80–90
Axial piston (bent-axis)	250	350	17	3500	200	2000	25	High	90	50–90
Radial piston	350	1720	0.3	1000	200	2000	50	High	90	80–90

1.14.2 Maximum Delivery

The pump system selected must be capable of delivering the maximum flow rate demanded by the circuit. If the circuit demand is reasonably constant, a fixed displacement pump is chosen. When the demand is at a series of fixed levels, a multi-pump system is used. For demands which vary within a relatively narrow band, a variable displacement pump is used. If there is a wide variance in system demand, an accumulator circuit may best satisfy the requirements.

Pump capacities are stated by manufactures for a particular viscosity fluid at given operating temperatures and pressures. Any increase in temperature and hence a reduction in viscosity or an increase in operating pressure causes more leakage across the pump and consequently reduces the pump delivery. As the pump wears the leakage will increase.It is usual to select a pump with a capacity about 10% higher than that required to make an allowance for the reduction in volumetric efficiency with wear. Pumps are available with flows from a fraction of 1 LPM to–1000 LPM and above.

1.14.3 Type of Control

Various types of pump controls are available such as manual servo control, pressure compensated control, constant power control and constant flow control. The choice of control is dependent upon the circuit requirement such as complexity, accuracy of control, cost, type of machining operation,etc.The designer has to choose carefully the type of control after a detailed study of system characteristics.

1.14.4Pump Drive Speed

Amajority of pumps are driven directly from the prime mover – electric motor or internal combustion engine–so the proposed drive speed is known. The fluid delivery rate is proportional to the speed of rotation. Each design has a minimum and maximum operating speed: the faster the pump runs, the shorter its life.

1.14.5Types of Fluid

Pumps are designed to operate within a particular range of fluid viscosity. Mineral oils of the correct viscosity work satisfactory with most pumps provided the oil is clean. Operating with synthetic or water-based fluids reduces the working life of a pump that relies on the hydraulic fluid to lubricate the bearings and moving parts. When any fluid other than a mineral oil is to be used, it is advisable to seek the pump manufacturer's advice.

1.14.6Fluid Contamination

Any fluid contamination causes pump damage. Precision pumps with very fine clearances are more susceptible to damage. If a contaminated fluid has to be pumped, such as in a cleanup loop, particular attention must be paid to pump selection. Non-precision gear pumps, lobe pumps and gerotor pumps are the most dirt tolerant.Whichever type is used, a strainer must be fitted in the suction line. In the case of precision pumps, the manufacturer's recommendation on filtration must be followed; otherwise the life of pump will be drastically reduced and the maker's warranty voided.

1.14.7Pump Noise

Noise has become increasingly important environmentally. Operating levels vary considerably between the pumps of the same type but of different makes.The manufacturers are working on those aspects which most affect the emission of noise– port plate design, bearings, flow passages, pressure controls, materials and methods of mounting. Generally, the sound generated increases with speed and pressure. Certain kinds do, however, propagate lower noise levels, in particular, those with internal gears. A multi-stage internal gear pump is marketed by one manufacturer under the name Q pump, with Q signifying quiet.

Example 1.15

The intensity (in units of W/m^2) of the noise of a pump increases by a factor of 10 due to cavitation. What is the corresponding increase in noise level in decibels?

Solution:

$$\text{dB increase} = 10\times\ \log\frac{I(\text{final})}{I(\text{initial})} = 10\times\log_{10} = 10\,\text{dB}$$

1.14.8Size and Weight of a Pump

Generally, not only the overall size and weight of a hydraulic system is important in mobile installations, but also the whole system is important, as the size and weight of a pump is only part of the whole system. In a mobile hydraulic field, the trend is to reduce the weight of the hydraulic system by increasing the operating pressure,reducing the size of the reservoir and using efficient oil coolers.

The best power-to-weight ratios can usually be achieved in the 200–300 bar operating pressure range. The actual sizeand weight of a pump depend upon the particular manufacture's design. Very light compact units have been developed for use in the aerospace industry but these tend to be extremely expensive.

1.14.9Efficiency

Reciprocating pumps tend to have higher efficiencies than rotary pumps. The actual efficiency depends on design, operating pressure, speed and fluid viscosity.Table1.5gives an indication of the range of efficiencies of various types of pumps.

Table 1.5 Efficiency ranges of pumps

Pump Type	Volumetric Efficiency	Overall Efficiency
Piston		
Plunger in line	≤ 99	≤ 95
Radial	> 95	> 90
Axial	> 95	> 90
Precision gear pumps	≤ 95	≤ 90
Vane pump	≤ 90	≤ 80

1.14.10 Cost

The initial cost of a pump is usually of secondary importance to running and maintenance costs. Gear pumps are cheaper, vane and piston pumps are expensive.

1.14.11 Availability and Interchangeability

A number of gear pump manufacturers produce units to CETOP and SAE standards so far as the external dimensions are concerned. This gives direct interchangeability between gear pumps of different manufacturers. The shafts, mounting flanges and port connections of most of the other types also comply with various international standards allowing a degree of interchangeability.

1.14.12 Maintenance and Spares

In every type of pump, the components involved in pumping worn out after a time and need replacing. In gear pumps, it is usual to replace the entire pump. With some types of vane pumps, the wear parts are grouped together as a cartridge that can easily be replaced without dismantling the pump drive. In the case of piston pumps, it may be advisable to ensure that the manufacturer offers a fast overall service for critical applications to carry a spare pump in stock.

Example 1.16

A pump has a displacement volume of 120 cm^3. It delivers 0.0015 m^3/s at 1440 RPM and 60 bar. If the prime mover input torque is 130 Nm. What is the overall efficiency of the pump? What is the theoretical torque required to operate the pump. The pump is driven by an electric motor having an overall efficiency of 88%. The hydraulic system operates 12 h/d for 250 days per year. The cost of electricity is Rs 8 per kWh. Determine the yearly cost of electricity to operate the hydraulic system. The amount of the yearly cost of electricity that is due to the inefficiencies of the electric motor and pump.

Solution: Given volumetric displacement, $V_\mathrm{D} = 120\ \mathrm{cm}^3$, $Q_\mathrm{A} = 0.0015\ \mathrm{m}^3/\mathrm{s}$, $N = 1440\ \mathrm{rpm}$, $P = 60\ \mathrm{bar}$, input torque $T_\mathrm{A} = 130\ \mathrm{N\,m}$.

Total number of working hours available $= 250 \times 12 = 3000\ \mathrm{h}$

Volumetric displacement in m^3/rev is

$$V_\mathrm{D} = \frac{120\ \mathrm{cm}^3}{\mathrm{rev}} \times \left(\frac{1\ \mathrm{m}}{100\ \mathrm{cm}}\right)^3 = 0.000120\ \mathrm{m}^3/\mathrm{rev}$$

Theoretical dischargecan be calculated as

$$Q_T = V_D N = 0.000120 \times \frac{1440}{60} \text{ rev/s} = 0.00288 \text{ m}^3/\text{s}$$

Now we can calculate the volumetric efficiency as

$$\eta_v = \frac{Q_A}{Q_T} = \frac{0.0015}{0.00288} = 52.08\%$$

Mechanical efficiency is given by

$$\eta_m = \frac{pQ_T}{T_A \omega} = \frac{60 \times 10^5 \times 0.00288}{130 \times 1440 \times \frac{2\pi}{60}} = \frac{17280}{19603} = 88.2\%$$

Note the product $T_A \omega$ gives power in units of Nm/s (W) where T_A has a unit of Nm and shaft speed has units of rad/s.

The overall efficiency is

$$\eta_o = \eta_m \times \eta_v = 0.882 \times 0.5208 = 0.459 = 45.9\%$$

Alternativelyoverall efficiency can also be calculated as

$$\eta_o = \frac{pQ_A}{T_A \omega} = \frac{60 \times 10^5 \times 0.0015}{130 \times 1440 \times \frac{2\pi}{60}} = \frac{9000}{19603} = 45.9\%$$

Now since the mechanical efficiency is known, we can calculate the theoretical torque as

$$T_T = T_A \times \eta_m = 130 \times .882 = 114.7 \text{ N m}$$

Thus, due to mechanical losses within the pump, 130 Nm of torque are required to drive the pump instead of 114.7 Nm.

First we calculate the mechanical input power the electric motor delivers to the pump.

Pump input power (kW) isgiven by

$$T_A \omega = 130 \times 1440 \times \frac{2\pi}{60} = 19603 \text{ W} = 19.603 \text{ kW}$$

Next we calculate the electrical input power.

Electric motor input power is given by

$$\frac{\text{Electric motor output power}}{\text{Electric motor overall efficiency}} = \frac{19.603}{0.88} = 22.28 \text{ kW}$$

So

Yearly cost = Power rate $\times$ Time per year $\times$ Unit cost of electricity
= 22.28 $\times$ 12 h/d $\times$ 250 d/y $\times$ 8 Rs/kWh
= Rs 534627

The total loss equals kW loss due to electric motor plus the kW loss due to pump. Thus, we have

$$\text{Total loss} = (1-0.88)\times 22.28 + (1-0.459)\times 19.603$$
$$= 2.674 + 10.61 = 13.284\,\text{kW}$$

Yearly cost due to inefficiencies is

$$\frac{13.284}{22.28}\times 534627 \text{ Rs/year} = 318760 \text{ Rs / year}$$

Since

$$\frac{13.284}{22.28} = 59.2$$

we conclude that 59.2% of the total cost of electricity is due to inefficiency of the electric motor and pump. This also means that only 49.8% of the electrical power entering the electric motor is transferred into hydraulic power at the pump outlet port.

Example 1.17

For the fluid power system of Fig. 1.22, the following data are given:

Cylinder piston diameter	0.203 m
Cylinder rod diameter	0.102 m
Extending speed of cylinder	0.0762 m/s
External load on a cylinder	178000 N
Pump volumetric efficiency	92%
Pump mechanical efficiency	90%
Pump speed	1800 RPM
Pump inlet pressure	−27600 Pa

(i)The total pressure drop in the line from the pump discharge port to the blank end of the cylinder is 517000 Pa.

(ii) The total pressure drop in the return line from the rod end of the cylinder = 345000 Pa.

Determine the

(a) Volumetric displacement of the pump.

(b) Input power required to drive the pump.

(c) Input torque required to drive the pump.

(d) Percentage of pump input power delivered to the load.

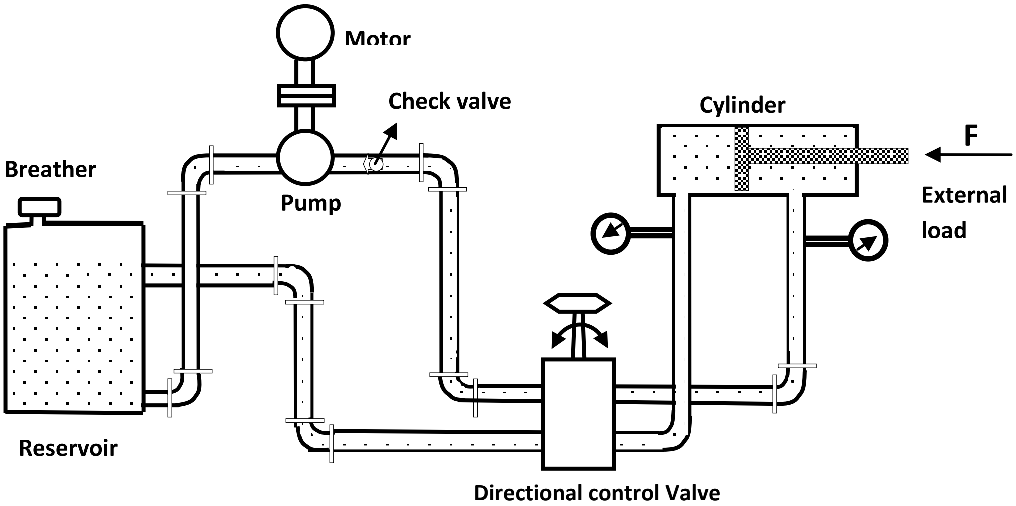


Figure 1.22</figure caption>

Solution:

(a) Volumetric displacement of pump.

$$Q_{\text{pump-actual}} = A_{\text{piston}} \times V_{\text{piston ext}}$$

$$= \frac{\pi}{4} \times 0.203^2 \times 0.0762 = 0.00247\,\text{m}^3/\text{s}$$

$$= 2.47 \text{ LPS}$$

$$Q_{\text{pump-theoretical}} = \frac{Q_{\text{pump-actual}}}{\eta_{\text{Vol}}} = \frac{0.00247}{0.92} = 0.00268\,\frac{\text{m}^3}{\text{s}}$$

Now

$$Q_{\text{pump-theoretical}} = V_{\text{D}} N$$

$$\Rightarrow 0.00268 = V_{\text{D}} \times \frac{1800}{60}$$

$$\Rightarrow V_{\text{D}} = 0.0000893\ \text{m}^3 = 0.0893\text{L}$$

(b) Input power required to drive the pump

$$\text{Pump output power} = (\Delta p)Q_{\text{actual}}$$

$$\Rightarrow p_{\text{blank-end}} \times A_{\text{piston}} - p_{\text{rod-end}} \times (A_{\text{piston}} - A_{\text{rod}}) = F_{\text{external load on cylinder}}$$

$$\Rightarrow p_{\text{blank-end}} \times \frac{\pi}{4} \times 0.203^2 - 345000 \times \frac{\pi}{4} \times (0.203^2 - 0.102^2) = 178000$$

$$\Rightarrow p_{\text{blank-end}} = 5758000\,\text{Pa} = 5758 \text{ kPa}$$

Pump output power = $\{(5758 + 517 + 27.6)0.00247\} = 15.6 \text{ kW}$

Pump input power = $\dfrac{\text{Pump output power}}{\eta_{\text{v}} \times \eta_{\text{m}}} = \dfrac{15.6}{0.92 \times 0.90} = 18.8\,\text{kW}$

(c) Input torque required to drive the pump

Pump input power is

$$T\omega = T \times 1800\,\frac{\text{rev}}{\text{min}} \times \frac{1\,\text{min}}{60}\,\frac{2\pi\,\text{rad}}{\text{s}} = 188\,\text{rad/s}$$

Now

$$18800\,\text{W} = T \times 188\,\text{rad/s}$$

So torque required to drive pump is T = 100 Nm.

(d) Power delivered to load is

$$F_{\text{external load on cylinder}} \times V_{\text{cyl.ext}}$$

$$= 178000 \times 0.0762 = 13600 \text{ W} = 13.6 \text{ kW}$$

Percent of pump input power delivered to load = 13.6/18.8 $\times$100 = 72.3%

Example 1.19

The system of in Example 1.17 contains a fixed displacement pump with a pressure relief valve set at 6871 kPa. The system operates 20 h/d for 250 days in a year. The cylinder is stalled in its fully extended position 70% of the time. When the cylinder is fully extended, 0.0633 LPS leaks past its piston.

(a) If the electric motor driving the pump has an efficiency of 85% and the cost of electricity is Rs 10 per kWh, find the annual cost of electricity for powering the system

(b) It is being considered to replace fixed displacement pump with a pressure-compensated pump (compensator set at 6871 kPa) that cost Rs 250000 more. How long will it take for the pressure-compensated pump to pay for itself if its overall efficiency is same as fixed displacement pump?

Solution

(a) Annual cost of electricity for powering the system

$$p_{\text{blank-end}} = 5758000\,\text{Pa} = 5758\,\text{kPa}$$

The total pressure drop in the line from the pump discharge port to the blank end of the cylinder is 517000 Pa.

Pump inlet pressure = −27600 Pa

Pump discharge pressure = $5758000\,\text{Pa} + 517000\,\text{Pa} - 27600\,\text{Pa} = 6247.4\text{ kPa}$

Pump input power = 18.8 kW

Electric motor input power = 18.8/0.85 = 22.1 kW

Thus with the cylinder fully extended (pressure relief valve set at 6871 kPa) we have

$$\text{Electric motor input power} = \frac{6871}{6247.4} \times 22.1 = 24.3\text{ kW}$$

Thus, the yearly cost of electricity is

Yearly cost = Power rate × Time per year × Unit cost of electricity

= 0.30 × 22.1 (kW) ×20 h/d × 250 d/yr × 10/kWh + 0.70 ×24.3 ×20 ×250 ×10

= Rs 1182000/year

(b) The fixed displacement pump produces 2.47 LPS at 6871 kPa when the cylinder is fully extended. Leakis 0.0633 LPS through the cylinder plus 2.407 LPS through the relief valve. Thus, when the cylinder is fully extended, we have power lost with a fixed displacement pump

$$pQ = 6871 \times 0.00247 = 16.97 \text{ kW}$$

Hence, the electric motor input power is

$$\frac{16.97}{0.828 \times 0.85} = 24.1$$

The overall efficiency of the pump 82%.The pressure-compensated pump would produce only 0.0633 LPS at 6871 kPa when the cylinder is fully extended. For this case we have the power lost with pressure-compensated pump is

$$pQ = 6871 \times 0.0000633 = 0.44 \text{ kW}$$

Hence, the electric motor input power is

$$\frac{0.44}{0.828 \times 0.85} = 0.63 \text{kW}$$

Thus, the kW power saved while cylinder is fully extended = 24.1−0.63 = 23.47 kW

Savings per year = 23.47 ×0.70 × 20 ×250 ×10 = Rs 821450 per year

Time to pay for pump = Rs 250000/821450 = 0.3 years

Objective-Type Questions

Fill in the Blanks

1. Non-positive displacement pumps are primarily ______type units that have a ______clearance between rotating and stationary parts.
2. Flow rate of pump does ______ with head in the case of a positive displacement pump, while it ______ with head in the case of a non-positive displacement pump.
3. In a lobe pump, the output may be slightly ______ because of the smaller number of ______ elements.
4. In a screw pump, there are no pulsations at ______ speed; it is a very quiet operating pump.
5. In a balanced vane pump, the rotor and vanes are contained within a ______ eccentric cam ring and there are ______ inlet segments and two outlet segments during each revolution.

State True or False

1. The volumetric capacity of a positive displacement pump is less than that of a non-positive displacement pump.
2. Too low elevation head between the reservoir and the pump inlet causes cavitation.
3. Efficiency is almost constant with the head in the case of non-positive displacement pumps.
4. The sole purpose of pumps is to create pressure.
5. Mechanical efficiency indicates the amount of energy losses that occur for reasons other than leakage.

Review Questions

1. What is a positive displacement pump? In what ways does it differ from a centrifugal pump?
2. Define the source of hydraulic power (pump).
3. Explain the working principle of a pump.
4. Pumps do not pump pressure. Justify this statement.
5. What is the function of a pump in a hydraulic system?
6. How is the pumping action in positive displacement pumps accomplished?
7. How the volumetric efficiency of a positive displacement pump is determined?
8. List the advantages of hydrostatic pumps over hydrodynamic pumps.
9. Give the classification of hydrostatic pumps used in a fluid power system.
10. What is the difference between a fixed displacement pump and a variable displacement pump?
11. What types of pumps are available in variable displacement designs?
12. How can the vane pump/piston pumps be made as variable displacement pumps?
13. Name three designs of external gear pumps.
14. Name two designs of internal gear pumps.
15. What are the advantages of screw pumps over other gear pumps?
16. Why is the operation of screw pump quiet?
17. How can the unbalanced vane pump be used as a variable displacement pump?
18. What is a pressure-compensated vane pump and how does it work?
19. What is meant by a balanced design vane pump?
20. Name the important considerations when selecting a pump for a particular application.

21. Why a gear pump cannot be used as a variable displacement pump?
22. How can the displacement of an axial piston pump be varied?
23. What is pump cavitation and what is its cause?
24. How is pressure developed in hydraulics systems?
25. Why centrifugal pumps are rarely used in fluid power systems?
26. Draw the graphical symbols for the fixed displacement and variable displacement pumps.
27. Which parameters affect the noise level of a positive displacement pump?
28. What is meant by the pressure rating of a positive displacement pump?
29. Name the four rules that control or eliminate cavitation of a pump.
30. Comment on the relative comparison in performance among gear, vane and piston pumps.
31. What are the two ways of expressing a pump size?
32. What are pump characteristic curves? Draw the same for the positive displacement pumps.
33. How is the capability of a variable displacement pump affected by the addition of pressure compensation?
34. Name the three principal ways in which noise reduction can be accomplished.

35. What are the most common things apart from pressure or speed that can cause a pump to fail? Explain each.
36. Where are external gear pumps used?

Answers
Fill in the Blanks

1.Velocity, large
2.Not change, decreases
3.Greater pulsation,meshing
4.Higher;
5.Double, two

State True or False

1.True
2.False
3.False
4.False
5.True

Lecture 10

HYDRAULIC MOTORS

Learning Objectives

Upon completion of this chapter, the student should be able to:

- Differentiatebetween a hydraulic motor and a hydraulic pump.
- List various applications of hydraulic motor in fluid power.
- Discuss variousclassifications of hydraulic motor.
- Explain the construction and working of gear, vane and piston motors.
- Discuss the various types of limited-rotation motors.
- Explain various types of efficiency terms used in hydraulic motors.
- Evaluate the performance parameters of systems using motors.

1.1Introduction

Hydraulic motors are rotary actuators. However, the name rotary actuator is reserved for a particular type of unit that is limited in rotation to less than 360°.A hydraulic motor is a device which converts fluid power into rotary power or converts fluid pressure into torque. Torque is a function of pressure or, in other words, the motor input pressure level is determined by the resisting torque at the output shaft. A hydraulic pump is a device which converts mechanical force and motion into fluid power. A hydraulic motor is not a hydraulic pump when run backward. A design that is completely acceptable as a motor may operate very poorly as a pump in a certain applications. Differences between a hydraulic motor and a hydraulic pump are given in Table 1.1.

Table1.1 Differences between a hydraulic motor and a hydraulic pump

Hydraulic Motor	Hydraulic Pump
It is a device for delivering torque at a given pressure. The main emphasis is on mechanical efficiency and torque that can be transmitted.	It is a device for delivering flow at a given pressure. The main emphasis is on volumetric efficiency and flow.
Motors usually operate over a wide range of speed, from a low RPM to high RPM.	Pumps usually operate at high RPM.
Most motors are designed for bidirectional applications such as braking loads, rotary tables.	In most situations, pumps usually operate in one direction.
Motors may be idle for long time (as in index table).	Pumps usually operate continuously.
Motors are subjected to high side loads (from gears, chains, belt-driven pulleys).	Majority of pumps are not subjected to side loads. Usually pumps are pad mounted on power pack top and shaft is connected to the prime mover directly.

1.2Applications

Hydraulic motors have becomepopular in industries. Hydraulic motors can be applied directly to the work.They provide excellent control for acceleration, operating speed, deceleration, smooth reversals and positioning. They also provide flexibility in design and eliminate much of bulk and weight of mechanical and electrical power transmission. The applications of hydraulic motors in their various combinations with pumping units are termed hydrostatic transmission.

A hydrostatic transmission converts mechanical power into fluid power and then reconverts fluid power into shaft power. The advantages of hydrostatic transmissions include power transmission to remote areas, infinitely variable speed control, self-overload protection, reverse rotation capability, dynamic braking and a high power-to-weight ratio. Applications include material-handling equipment, farm tractors, railway locomotives, buses, lawn mowers and machine tools.

New fields of applications are being discovered constantly for hydrostatic transmissions. Farm implements, road machinery, material-handling equipment, Numerical Control(NC) machineshigh-performance aircrafts, military uses and special machinery are only a few of new fields expanding through the use of fluid power transmission. Many automobiles, railway locomotives and buses usea hydrostatic transmission.

1.3 Comparison Betweena Hydraulic Motor and an Electric Motor

Table 1.2 gives the comparison between a hydraulic motor and an electric motor.

Table 1.2 Comparison between a hydraulic motor and an electric motor

Electric Motor	Hydraulic Motor
Electric motors cannot be stopped instantly. Their direction of rotation cannot be reversed instantly. This is because of air gap between the rotor and stator and the weak magnetic field.	Hydraulic motors can be stalled for any length of time. Their direction of rotation can be instantly reversed and their rotational speed can be infinitely varied without affecting their torque. They can be braked instantly and have immense torque capacities.
Electric motors are heavy and bulky.	Hydraulic motors are very compact compared to electric motors. For the same power, they occupy about 25% of the space required by electric motors and weigh about 10% of electric motors.
Moment of inertia-to-torque ratio is nearly 100.	Moment of inertia-to-torque ratio is nearly 1.

1.4Classification of Hydraulic Motors

There are two types of hydraulic motors: (a) High-speed low-torque motors and (b) low–speed high-torque motors. In high-speed low-torque motors, the shaft is driven directly from either the barrel or the cam plate, whereas in low-speed high-torque motors, the shaft is driven through a differential gear arrangement that reduces the speed and increases the torque. Depending upon the mechanism employed to provide shaft rotation, hydraulic motors can be classified as follows:

1. Gear motors.
2. Vane motors.
3. Piston motors:
   - Axialpiston-type motors.
   - Radial piston-type motors.

Gear motors are the least efficient, most dirt-tolerant and have the lowest pressure rating of 3. Piston motors are the most efficient, least dirt-tolerant and have high pressure ratings. Vane and piston motors can be fixed or variable displacement, but gear motors are available with only fixed displacement.

1.5Gear Motors: A gear motor develops torque due to hydraulic pressure acting against the area of one tooth. There are two teeth trying to move the rotor in the proper direction, while one net tooth at the center mesh tries to move it in the opposite direction. In the design of a gear motor, one of the gears is keyed to an output shaft, while the other is simply an idler gear. Pressurized oil is sent to the inlet port of the motor. Pressure is then applied to the gear teeth, causing the gears and output shaft to rotate. The pressure builds until enough torque is generated to rotate the output shaft against the load. The side load on the motor bearing is quite high, because all the hydraulic pressure is on one side. This limits the bearing life of the motor. Schematic diagram of gear motor is shown in Fig.1.1.

Most of the gear motors are bidirectional. Reversing the direction of flow can reverse the direction of rotation.As in the case of gear pumps, volumetric displacement is fixed. Due to the high pressure at the inlet and low pressure at the outlet, a large side load on the shaft and bearings is produced. Gear motors are normally limited to 150 bar operating pressures and 2500 RPM operating speed. They are available with a maximum flow capacity of 600 LPM. The gear motors are simple in construction and have good dirt tolerance, but their efficienciesare lower than those of vane or piston pumps and they leak more than the piston units. Generally,they are not used as servo motors.Hydraulic motors can also be of internal gear design. These types can operate at higher pressures and speeds and also have greater displacements than external gear motors.

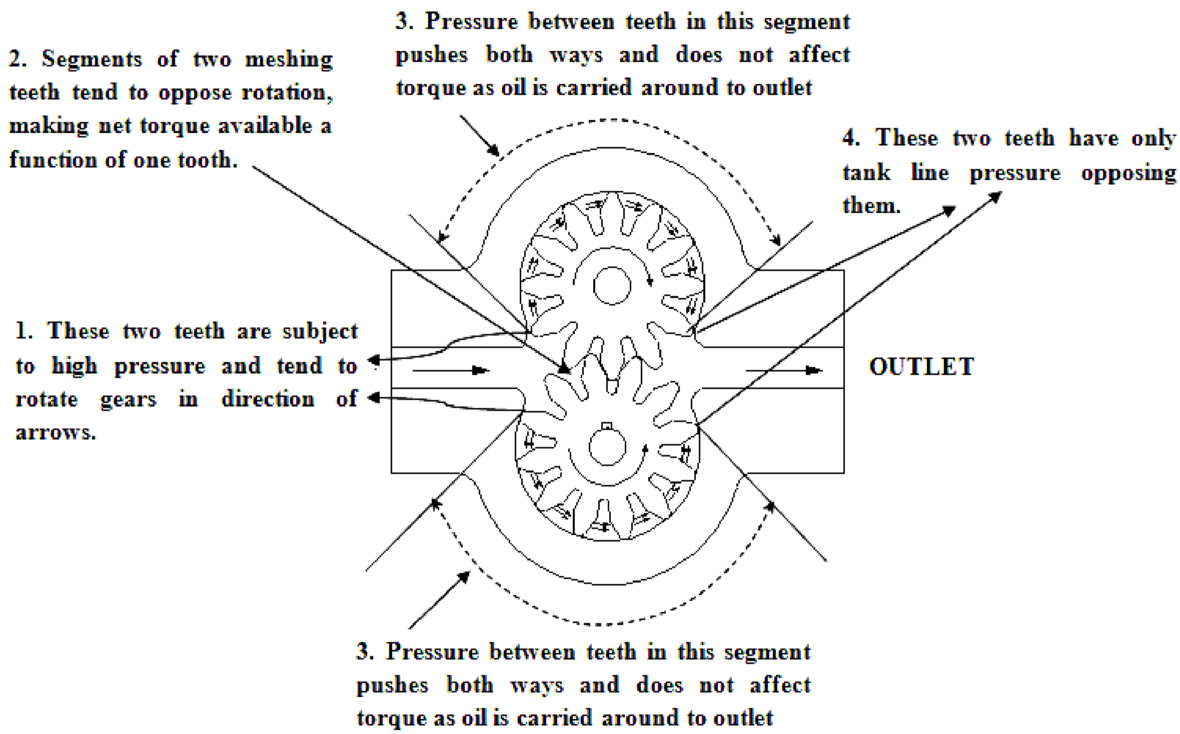


Figure 1.1 Gear motor

1.6Vane Motors

Figure 1.2 shows an unbalanced vane motor consisting of a circular chamber in which there is an eccentric rotor carrying several spring or pressure-loaded vanes. Because the fluid flowing through the inlet port finds more area of vanes exposed in the upper half of the motor, it exerts more force on the upper vanes, and the rotor turns counterclockwise. Close tolerances are maintained between the vanes and ring to provide high efficiencies.

The displacement of a vane hydraulic motor is a function of eccentricity. The radial load on the shaft bearing of an unbalanced vane motor is also large because all its inlet pressure is on one side of the rotor.

Figure 1.3 shows the balanced vane motor. The radial bearing load problem is eliminated in this design by using a double-lobed ring with diametrically opposite ports. Side force on one side of bearing is canceled by an equal and opposite force from the diametrically opposite pressure port. The like ports are generally connected internally so that only one inlet and one outlet port are brought outside. The balanced vane-type motor is reliable open-loop control motor but has more internal leakage than piston-type and therefore generally not used as a servo motor.

1.7Piston Motors

Piston motors are classified into the following types:

1. According to the piston of the cylinder block and the drive shaft, piston motors are classified as follows:

- Axial piston motors.
- Radial piston motors.

2. According to the basis of displacement, piston motors are classified as follows:

- Fixed-displacement piston motors.
- Variable-displacement piston motors.

1.7.1 Axial Piston Motors

In axial piston motors, the piston reciprocates parallel to the axis of the cylinder block. These motors are available with both fixed-and variable-displacement feature types. They generate torque by pressure acting on the ends of pistons reciprocating inside a cylinder block. Figure 1.4 illustrates the inline design in which the motor, drive shaft and cylinder block are centered on the same axis. Pressure acting on the ends of the piston generates a force against an angled swash plate. This causes the cylinder block to rotate with a torque that is proportional to the area of the pistons. The torque is also a function of the swash-plate angle. The inline piston motor is designed either as a fixed- or a variable-displacement unit. The swash plate determines the volumetric displacement. Refer Fig.1.5.

Figure1.4 Inline piston motor

In variable-displacement units, the swash plate is mounted on the swinging yoke. The angle can be varied by various means such as a lever, hand wheel or servo control.If the offset angel is increased, the displacement and torque capacity increase but the speed of the drive shaft decreases. Conversely, reducing the angle reduces the torque capability but increases the drive shaft speed.

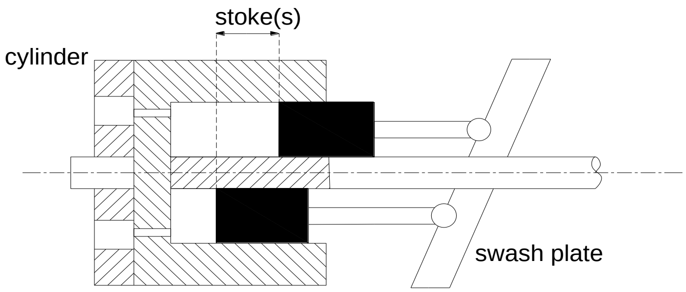


Figure1.5 Swash-plate piston motor

1.7.2 Bent-Axis Piston Motors

A bent-axis piston motor is shown in Fig.1.6. This type of motor develops torque due to pressure acting on the reciprocating piston. In this motor, the cylinder block and drive shaft mount at an angel to each other so that the force is exerted on the drive shaft flange.

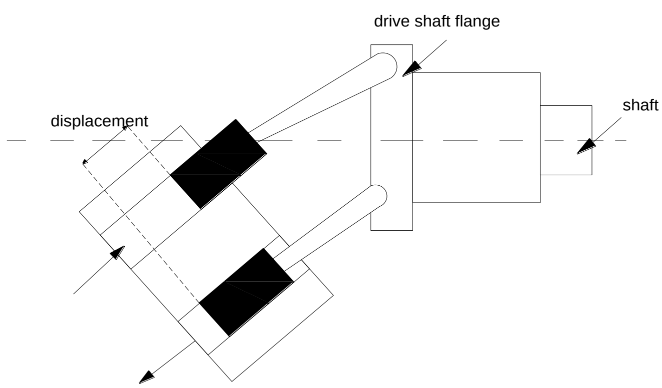


Figure1.6 Inline piston motor

Speed and torque depend on the angle between the cylinder block and the drive shaft. The larger the angle, the greater the displacement and torque, and the smaller the speed. This angle varies from 7.5° (minimum) to 30° (maximum). This type of motor is available in two types, namely fixed-displacement type and variable-displacement type. Refer Fig.1.7

7.3 Radial Piston Motors

In radial piston-type motors, the piston reciprocates radially or perpendicular to the axis of the output shaft. The basic principle of operation of the radial piton motors is shown in Fig.1.8.Radial piston motors are low-speed high-torque motors which can address a multifarious problem in diverse power transfer applications.

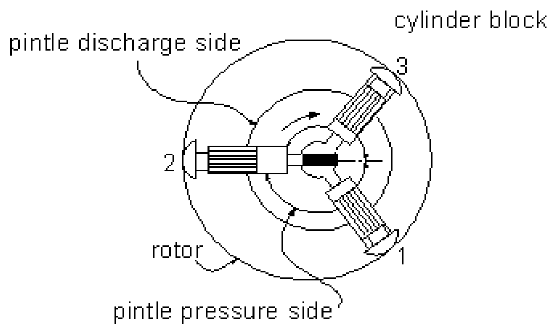


Figure1.8 Radial piston motor

1.8Semi-Rotary Actuators

These are devices used to convert fluid energy into a torque which turns through an angle limited by the design of the actuator. With the majority of designs, the angle of rotation is limited to 360° although it is possible to considerably exceed this when using piston-operated actuators.

1.8.1 Vane-Type Semi-Rotary Actuator (Single Vane)

A single-vane rotary actuator is shown in Fig. 1.9. A semi-rotary actuator allows only a partial revolution.A vane-type semi-rotary actuator consists of a vane connected to an output shaft. When hydraulic pressure is applied to one side of the vane, it rotates. A stop prevents the vane from rotating continuously. The rotation angle in the case of a single-vane semi-rotary actuator is 315°.

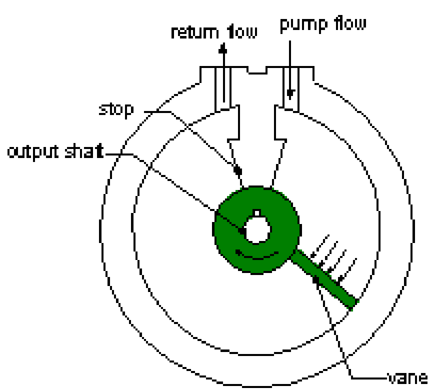


Figure 1.9 Vane-type semi-rotary

1.8.2Two-Vane-Type Semi-Rotary Actuator

A two-vane rotary actuator is shown in Fig.1.10. The advantage of this design is that the torque output is increased because the area subjected to pressure is large. However, two-vane models cannot rotate as many degrees as can single-vane models.It is limited to 100°. Passageways are used to connect the different chambers of the rotary actuator.

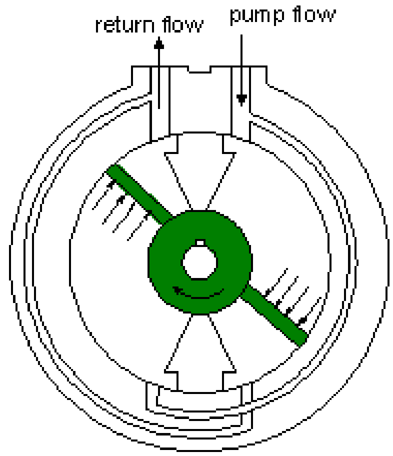


Figure1.10 Two-vane type

1.8.3Analysis ofa Semi-Rotary Single-Vane Motor

Let

R_R = Outer radius of the output shaft (m)

R_V = Outer radius of the vane (m)

L = Width of the vane (m)

p = Hydraulic pressure (Pa)

F = Hydraulic force acting on the vane (N)

A = Surface area of vane in contact with oil (m^2)

T = Torque capacity (N m)

The force on the vane equals the pressure times the vane surface area:

$$F = pA = p(R_V - R_R)L$$

The torque equals the vane force times the mean radius of the vane:

$$T = p(R_V - R_R)L\frac{R_V + R_R}{2}$$

On rearranging, we have

$$T = p(R^2{}_V - R^2{}_R)L \qquad (1.1)$$

A second equation for torque can be developed by noting the following relationship for volumetric displacement V_D :

$$V_D = \pi(R^2{}_V - R^2{}_R)L \quad (1.2)$$

Combining Equations. (1.1) and (1.2) yields

$$T = \frac{p \times V_D}{2\pi}$$

Example 1.1A single-vane rotary actuator has the following physical data:

Outer radius of rotor = 0.5 cm
Outer radius of vane = 1.5 cm
Width of vane = 1 cm

If the torque load is 1000 Ncm, what pressure must be developed to overcome the load?

Solution: The volumetric displacement is given by

$$V_D = \pi(1.5^2 - 0.5^2)1 = 6.28 \text{ cm}^3$$

The pressure is

$$p = \frac{2\pi T}{V_D} = \frac{2\pi(1000)}{6.28} = 1000 \text{ N/cm}^2$$

$$= 1000 \times 10^4 \text{ N/m}^2$$

$$= 10 \text{ N/mm}^2 = 10 \text{ MPa}$$

1.9Chain and Sprocket Semi-Rotary Actuator

In this design (Fig. 1.11), an endless chain and a sprocket areused.It is suitable for multi-revolution applications. The chain is anchored to two pistons, one large and other small, which when in their respective bores separate the half of the unit. The larger cylinder is the power cylinder and the smaller cylinder is the chain return or seal cylinder. The idler is automatically a tensioned one, so that a constant tension is maintained. Pressure is applied to one port of the actuator. The larger piston moves away from the port due to differential areas of the two pistons. The movement of larger piston pulls the chain, causing the sprocket and output shaft to rotate.

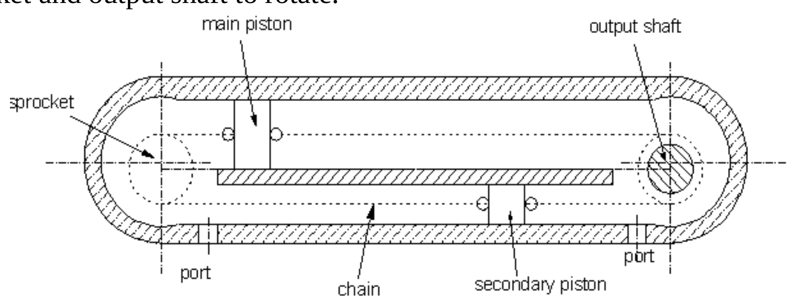


Figure1.11 Chain and sprocket

1.10Rack and Pinion Rotary Actuator

A rack and pinion rotary actuator (Fig. 1.12) is a commonly used design for obtaining partial revolution actuation. This consists of a hydraulic cylinder with a rack and pinion gear mechanism. The rack gear on the piston rod turns the pinion gear, thereby converting the linear motion of the piston into rotary motion, which is transmitted to the load through the output shaft.

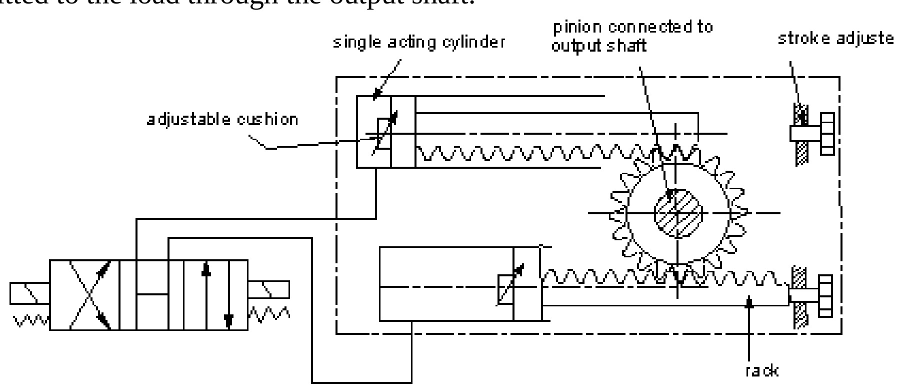


Figure1.12 Rack and rotary actuator

Another design of a rack and pinion semi-rotary actuator is shown in Fig.1.13. In this design, the cylinder drives a pinion gear and the rack is an integral part of the piston rod. The angle of rotation depends upon the stroke of the cylinder, rack and the pitch circle diameter of the pinion. The start and finish of the stroke are adjusted by means of an internal stop(stroke adjuster).

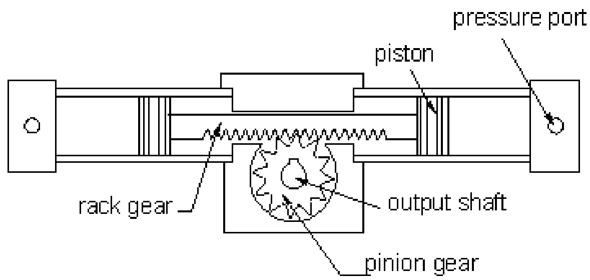


Figure 1.13 Rack and rotary actuator

1.11Hydraulic Motor:Theoretical Torque, Power and Flow Rate

The torque generated by africtionless hydraulic motor is known as a theoretical torque.Theoretical torque can be calculated by the following formula:

$$T_{\mathrm{T}} = \frac{p \times V_{\mathrm{D}}}{2\pi}$$

where$V_{\mathrm{D}}$is the volumetric displacement in m^3/rev and p is the pressure in N/m^2.The power developed by a frictionless motor is known as theoretical power.It can be calculated by the following formula:

$$P_{\mathrm{T}} = T_{\mathrm{T}} \times \omega$$

$$(W) = (\mathrm{Nm})\ \mathrm{rad/s} = W$$

where T_{T} is the theoretical torque in Nm, ω is the speed of the motor in rad/s and $\omega = 2\pi N / 60$, where N is the speed of the motor in rev/min.The flow ratea hydraulic motor would consume if there were no leakage is known as the theoretical flow rate Q_{T}.Mathematically, theoretical flow rate is given by

$$Q_{\mathrm{T}} = V_{\mathrm{D}} n$$

where V_{D} is the volumetric discharge in m^3/rev, n is the speed of motor in rev/s = N/60 and N is the speed of motor in rpm.

Lecture 11

HYDRAULIC MOTORS [CONTINUED]

1.12Performance of Hydraulic Motors

The performance of hydraulic motors depends upon many factors such as precision of their parts, tolerances between the mating parts, etc.Internal leakage between the inlet and outlet affects the volumetric efficiency. Friction between mating parts affects the mechanical efficiency of a hydraulic motor.
Gear motors typically have an overall efficiency of 70–75% as compared to vane motors which have 75–85% and piston motors having 85–95%.
Motor torque is divided into three separate groups:

1. **Starting torque:** The starting torque is the turning force the motor exerts from a dead stop.
2. **Running torque:** Running torque is exerted when the motor is running and changes whenever there is a change in fluid pressure.
3. **Stalling torque:** Stalling torque is the torque necessary to stop the motor.

In most hydraulic motors, the stalling and starting torques are equal. Usually, starting torque is 75–80% of the maximum design torque.

1. Volumetric efficiency: The volumetric efficiency of a hydraulic motor is the ratio of theoretical flow rate to actual flow rate required to achieve a particular speed. The motor uses more flow than the theoretical due to leakage:

$$\eta_{\mathrm{v}} = \frac{\text{Theoretical flow rate the motor should be supplied with}}{\text{Actual flow rate supplied to the motor}} = \frac{Q_{\mathrm{T}}}{Q_{\mathrm{A}}}$$

2. Mechanical efficiency: The mechanical efficiency of a hydraulic motor is the ratio of actual work done to the theoretical work done per revolution. The output torque of a hydraulic motor is less than theoretical torque due to mechanical friction between the mating parts:

$$\eta_{\mathrm{m}} = \frac{\text{Actual torque delivered by the motor}}{\text{Torque the motor should theoretically deliver}} = \frac{T_{\mathrm{A}}}{T_{\mathrm{T}}}$$

Here,theoretical torque and actual torque are given by

$$T_{\mathrm{T}} = \frac{V_{\mathrm{D}} \times p}{2\pi}$$

$$T_{\mathrm{A}} = \frac{\text{Actual wattage delivered by the motor}}{N}$$

3. Overall efficiency: The overall efficiency of a motor is the ratio of output power to input power of the motor. Output power is mechanical power output at the shaft and input power is fluid energy supplied to the inlet of the hydraulic motor:

$$\eta_{\mathrm{o}} = \frac{\text{Actual power delivered by the motor (mechanical)}}{\text{Actual power delivered to the motor (hydraulic)}}$$

$$\eta_{\mathrm{o}} = \frac{T_{\mathrm{A}} \times N}{p \times Q_{\mathrm{A}}}$$

$$= \frac{T_A \times T_T \times N}{T_T \times p \times Q_A}$$

$$= \frac{T_A \times V_D \times p \times N}{T_T \times p \times Q_A \times 2\pi}$$

$$= \frac{T_A \times Q_T}{T_T \times Q_A}$$

$$\Rightarrow \eta_o = \eta_V \eta_m$$

So

Overall efficiency = Volumetric efficiency $\times$ Mechanical efficiency

Note: The actual power delivered to a motor by a fluid is called hydraulic power and the actual power delivered to a load by a motor via a rotating shaft is called brake power.

Example 1.2

A hydraulic motor is required to drive a load at 500 rpm with 1000 Nm of torque. What is the output power?

Solution

N = 500 rpm $= \dfrac{500 \times 2\pi}{60} = 52.36 \text{ rad/s}$

$T_A = 1000$ N m

Now

Power $= T_A (\text{N m}) \times \text{N (rad/s)}$

$$= 1000 \times 52.36$$

$$= 52360 \text{ W}$$

The output power is 52.360 kW.

Example1.3

A hydraulic motor receives a flow rate of 72 LPM at a pressure of 12000 kPa. If the motor speed is 800 RPM, determine the actual torque delivered by the motor assuming the efficiency 100%?

Solution

Method I

Actual flow rate

$$Q_A = 72 \text{ LPM} = \frac{72 \times 10^{-3}}{60} = 1.2 \times 10^{-3} \text{ m}^3/\text{s}$$

Speed of motor N = 800 RPM. So

$$\omega = 800 \times 2\pi / 60 = 83.78 \text{ rad/s}$$

Pressure = 12000 × 10^3 Pa.

Overall efficiency can be calculated using

$$\eta_o = \frac{T_A \times N}{P \times Q_A}$$

Substituting the values we get

$$1 = \frac{T_A \times 83.78}{12000 \times 10^3 \times 1.2 \times 10^{-3}}$$

$$\Rightarrow T_A = 171.88 \text{ N m}$$

So the actual torque $T_A = 171.88$ N m.

Method II

$$\text{Hydraulic power} = pQ = 12000 \times \frac{72}{60} \times 10^{-3} = 14.4 \text{ kW}$$

$$T\ (\text{Nm}) \times \omega(\text{rad/s}) = 14400 \text{ W}$$

So

$$T = \frac{14400}{800 \times \frac{2\pi}{60}} = 172 \text{ N m}$$

Example 1.4

A hydraulic motor has a 100 cm^3 volumetric displacement. If it has a pressure rating of 140 bar and receives oil from a 0.001 m^3/s theoretical flow rate pump, find the motor (a) speed, (b) theoretical torque, (c) theoretical kW power.

Solution:

(a) Speed: We have the theoretical flow rate given by

$$Q_\text{T} = V_\text{D} \times n$$

$$\Rightarrow 0.001 = 100 \times 10^6 \times n$$

$$\Rightarrow n = 10 \text{ RPS(revolutions per second)}$$

and

$$N = 600 \text{ RPM}$$

(b) Theoretical torque

$$T_\text{T} = \frac{p \times V_\text{D}}{2\pi} = \frac{140 \times 10^5 \times 100 \times 10^{-6}}{2\pi} = 222.82 \text{ N m}$$

(c) Theoretical kW power

$$P = Q_\text{T} \times p = 0.001 \text{ m}^3/\text{s} \times 140 \times 10^5 \text{ N/m}^2 = 14000 \text{ W} = 14 \text{ kW}$$

Alternately,

$$\text{Power} = T_\text{T}\omega = 222.82 \times 10 \times 2\pi = 14000 \text{ W} = 14 \text{ kW}$$

Example 1.5

The pressure rating of the components in a hydraulic system is 10^5 kPa. The system contains a hydraulic motor to turn a 0.3 m radius drum at 30 RPM to lift a weight of load 4000 N as shown in Fig. 1.14. Determine the flow rate and brake power if the motor efficiency is 90%.

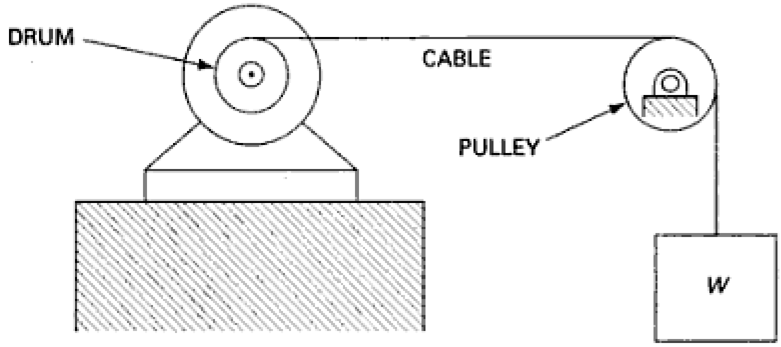


Figure 1.14

Solution: We have the theoretical torque given by

$$T_{\mathrm{T}} = \frac{p \times V_{\mathrm{D}}}{2\pi}$$

$$\Rightarrow 4000 \times 0.3 = \frac{10^8 \times V_{\mathrm{D}}}{2\pi}$$

$$\Rightarrow V_{\mathrm{D}} = 7.54 \times 10^{-5}\ \mathrm{m}^3 = 0.0754\ \mathrm{L}$$

Theoretical flow rate is

$$Q_{\mathrm{T}} = V_{\mathrm{D}} \times N = 7.54 \times 10^{-5} \times \frac{30}{60} = 0.0000377\ \mathrm{m}^3/\mathrm{s}$$

Power

$$P = pQ = 1 \times 10^8\ \mathrm{N/m^2}(0.0000377\ \mathrm{m^3/s}) = 3770\ \mathrm{W} = 3.77\ \mathrm{kW}$$

Example 1.6

A hydraulic system contains a pump that discharges oil at 13.8 MPa and 0.00632 m^3/s to a hydraulic motor shown in Fig. 1.15. The pressure at the motor inlet is 12.40 MPa due to pressure drop in the line. If oil leaves the motor at 1.38 MPa, determine the power delivery by the 100% efficient motor.

(a) What torque would a hydraulic motor deliver at a speed of 1750 RPM if it produces 3 kW?

(b) If the pressure remains constant at 13.8 MPa, (i) what would be the effect of doubling the speed on the torque and (ii) what would be the effect of halving the speed on the torque?

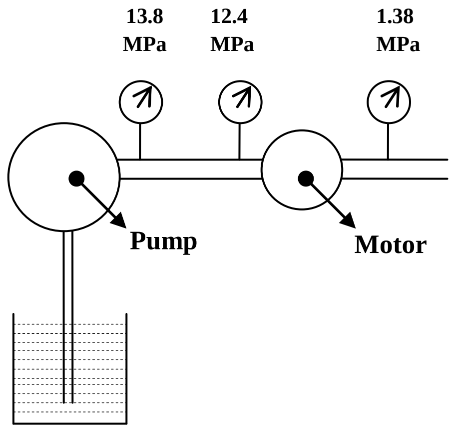


Figure 1.15

Solution: We have

$$\text{Power} = \Delta p\ Q = (12400 - 1380)\ \text{kPa} \times 0.00632\ \text{m}^3/\text{s}$$
$$= 69.6\ \text{kW}$$

Note: If the pipeline between the pump and motor is horizontal and of constant diameter, then the cause of pressure drop (12.4 − 1.38 MPa) is due to friction.

(a) We have

$$P = T \times \omega = 3000$$
$$\Rightarrow T \times \frac{2 \times \pi \times 1750}{60} = 3000$$
$$\Rightarrow T = \frac{3000 \times 60}{2 \times \pi \times 1750} = 16.37\ \text{N m}$$

(b)

(i) $T = \dfrac{pV_\text{D}}{6.28}$ .Since p and V_D are both constant, torque remains constant. This would, however, double the power.

(ii) The torque T remains constant while the power is reduced by 50%.

Example 1.7

A hydraulic motor has a displacement of 40 cm^3/rev and is used in a system with a maximum pressure of 20000 kPa. Determine the actual torque delivered by the motor assuming that it is 100% efficient.

Solution:

Displacement $V_\text{D} = 40 \times 10^{-6}\ \text{m}^3/\text{rev}$

Pressure of the system P = 20000 kPa

Theoretical torque

$$T_\text{T} = \frac{V_\text{D} \times p}{2\pi} = \frac{40 \times 10^{-6} \times 20000 \times 10^3}{2\pi} = 127.3\ \text{N m}$$

Since the motor is 100% efficient, the actual torque is equal to the theoretical torque

$$T_\text{A} = 127.3\ \text{N m}$$

Example 1.8

A motor must produce a torque of 350 Nm in a system with an operating pressure of 25000 kPa. What size motor should we select? Assume 100% efficiency.

Solution: Given $T_{\mathrm{A}} = 350\ \mathrm{N\ m}$. Since the motor has 100% efficiency,

Theoretical torque = Actual torque

$$\Rightarrow T_{\mathrm{T}} = \frac{V_{\mathrm{D}} \times P}{2\pi}$$

$$\Rightarrow 350 = \frac{V_{\mathrm{D}} \times 25000 \times 10^3}{2\pi}$$

$$\Rightarrow V_{\mathrm{D}} = 88\ \mathrm{cm^3/rev}$$

So we should select a motor having the displacement of 88 cm^3/rev.

Example 1.9

A hydraulic motor has a displacement of 164 cm^3 and operates with a pressure of 70 bar and a speed of 2000 rpm. If the actual flow rate consumed by the motor is 0.006 m^3/s and the actual torque delivered by the motor is 170 Nm, find (a) η_{v}, (b) η_{m}, (c) η_{o} and (d) actual power delivered by the motor?

Solution:

(a) We have

$$\eta_{\mathrm{v}} = \frac{\text{Theoretical flow rate the motor should consume}}{\text{Actual flow rate consumed by the motor}} = \frac{Q_{\mathrm{T}}}{Q_{\mathrm{A}}}$$

Now $Q_{\mathrm{A}} = 0.006\ \mathrm{m^3/s}$. Theoretical flow rate is

$$Q_{\mathrm{T}} = V_{\mathrm{D}} \times N = 164 \times 10^{-6}\ (\mathrm{m^3/rev}) \times \frac{2000}{60}\ (\mathrm{rev/s}) = 0.0055\ \mathrm{m^3/s}$$

So volumetric efficiency is

$$\eta_{\mathrm{v}} = \frac{0.0055}{0.006} \times 100 = 91.67\%$$

(b) Mechanical efficiency is given by

$$\eta_{\mathrm{m}} = \frac{\text{Actual torque delivered by the motor}}{\text{Theoretical torque motor should deliver}} = \frac{T_{\mathrm{A}}}{T_{\mathrm{T}}}$$

Theoretical torque,

$$T_{\mathrm{T}} = \frac{p \times V_{\mathrm{D}}}{2\pi} = \frac{70 \times 10^5 \times 164 \times 10^{-6}}{2\pi} = 182.71\ \mathrm{N\ m}$$

So mechanical efficiency,

$$\eta_{\mathrm{m}} = \frac{170}{182.71} = 93.04\%$$

(c) We have

$$\eta_{\mathrm{o}} = \eta_{\mathrm{m}} \times \eta_{\mathrm{v}} = 0.9304 \times 0.9167 = 0.853 = 85.3\%$$

So overall efficiency is 85.3 %.

(d) Actual power is

$$T_A\omega = 170 \times \left(2000 \times \frac{2 \times \pi}{60}\right) = 35600 \text{ W} = 35.6 \text{ kW}$$

Example 1.10

A hydraulic motor receives a flow rate of 72 LPM at a pressure of 12000 kPa. If the motor speed is 800 RPM and ifthe motor has a power loss of 3 kW, find the motor actual output torque and overall efficiency.

Solution: We have

$$72 \text{ LPM} = 0.0012 \text{ m}^3/\text{s}$$

Now we calculate the hydraulic power given to motor using

Hydraulic power = pQ = 0.0012 $\text{m}^3/\text{s} \times 12000$ = 14400 W = 14.4 kW

Actual power is obtained by subtracting the losses,

$$\text{Actual power} = T\omega = 14.4 - 3 = 11.4 \text{ kW}$$

$$\Rightarrow T = \frac{11400}{800 \times \frac{2\pi}{60}} = 136 \text{ N m}$$

The overall efficiency is

$$\text{Overall efficiency} = \frac{11.4}{14.4} = 0.792 = 79.2\ \%$$

Example 1.11

A hydraulic motor has a volumetric efficiency of 90% and operates at a speed of 1750 RPM and a pressure of 69 bar. If the actual flow rate consumed by the motor is 0.0047 m^3/s and the actual torque delivered by the motor is 147 Nm, find the overall efficiency of the motor.

Solution: The overall efficiency is

$$\eta_o = \frac{T_A\omega}{pQ_A} = \frac{147 \times \frac{1750 \times 2 \times \pi}{60}}{69 \times 10^5 \times 0.0047} = 0.83 = 83\%$$

Example 1.12

A hydrostatic transmission operating at 105 bar pressure has the following characteristics:

Pump	Motor
V_d = 100 cm^3	V_d = ?
η_v = 85%	η_v = 94%
η_m = 90%	η_m = 92%
N = 1000rpm	N= 600 rpm

Find the (a) displacement of motor and (b) motor output torque.

Solution:

(a) Pump theoretical flow rate

$$Q_{\text{T-pump}} = V_{\text{d}} \times N = \frac{100 \times 10^{-6} \times 1000}{60} = 1.667 \times 10^{-3}\ \text{m}^3/\text{s}$$

Actual flow rate

$$Q_{\text{A-pump}} = \eta_{\text{V}} \times Q_{\text{T}} = 1.667 \times 10^{-3} \times 0.85 = 1.42 \times 10^{-3}\ \text{m}^3/\text{s}$$

Actual flow from the pump is the actual flow to the motor. So for the motor

$$Q_{\text{A-motor}} = 1.42 \times 10^{-3}\ \text{m}^3/\text{s}$$

$$Q_{\text{T-motor}} = \eta_{\text{V}} \times Q_{\text{A}} = 1.42 \times 10^{-3} \times 0.94 = 1.332 \times 10^{-3}$$

So the theoretical flow rate, $Q_{\text{T-motor}} = 1.332 \times 10^{-3}\ \text{m}^3/\text{s}$. Now

$$Q_{\text{T-motor}} = V_{\text{D-motor}} \times N$$

$$\Rightarrow V_{\text{D-motor}} = \frac{Q_{\text{T-motor}}}{N_{\text{motor}}} = \frac{1.332 \times 10^{-3}}{600/60} = 1.332 \times 10^{-4} = 133\ \text{cm}^3/\text{rev}$$

So for the motor, the displacement is 133 cm^3/rev.

(b) Torque delivered by the motor

To calculate torque delivered by the motor, let us first calculate the actual power to motor

$$\text{Power}_{\text{actual to motor}} = p\,Q = 105 \times 10^5 \times 0.00142 = 14900\ \text{W}$$

Now

$$\text{Power}_{\text{actual by motor}} = \text{Power}_{\text{actual to motor}} \times \text{Mechanical efficiency} \times \text{volumetricefficiency}$$

$$\text{Power}_{\text{actual by motor}} = 14900 \times 0.94 \times 0.92 = 12900\ \text{W}$$

$$\text{Torque}_{\text{actual by motor}} = \frac{12900}{\frac{600 \times 2\pi}{60}} = 205\ \text{Nm}$$

1.13 Performance Curves for a Variable Displacement Motor

The following curves represent typical performance curves obtained for a 100 cm^3 variable displacement motor operating at full displacement.Figure1.16 gives the motor input flow (LPM) and motor output torque as a function of motor speed(RPM) at two pressure levels.

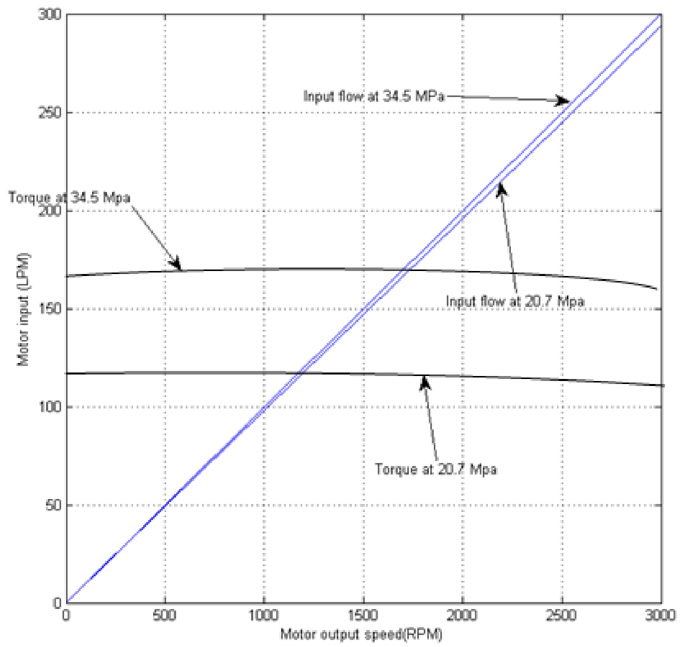


Figure 1.16 Motor input flow versus motor output torque

Figure 1.17gives the curves of overall and volumetric efficiencies as a function of motor speed(RPM) for pressure levels of 34.5 and 20.7 MPa.

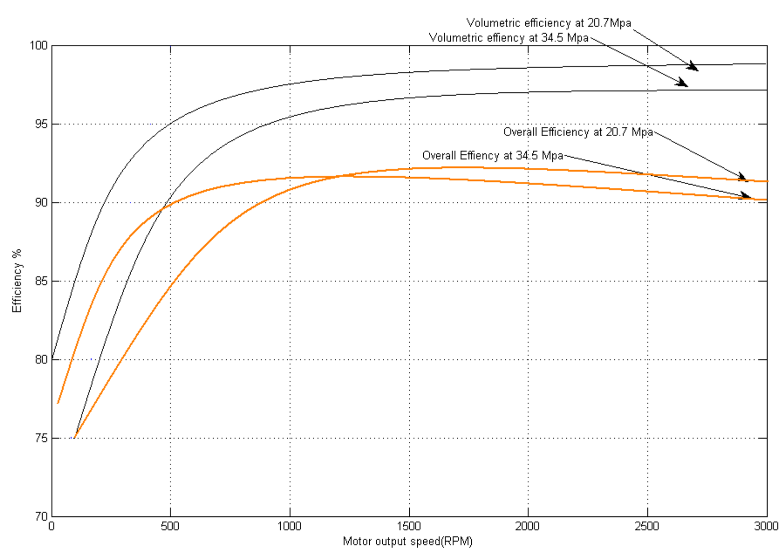


Figure1.17 Performance curves for a 100 cm^3 variable displacement motor

Objective-Type Questions

Fill in the Blanks

1. A hydraulic motor is a device which converts fluid power into _____ or converts fluid pressure into _____.
2. In an axial piston motor, the piston reciprocates _____ to the axis of the cylinder block.
3. In a radial piston-type motor, the piston reciprocates radially or _____ to the axis of the output shaft.
4. Rack and pinion rotary actuator is a commonly used design for obtaining _____ revolution actuation.
5. Gear motors typically have an overall efficiency of _____ as compared to _____ for piston motors.

State True or False

1. A hydraulic motor is a hydraulic pump which runs backward.
2. Gear motors are the most efficient and most dirt tolerant.
3. Hydraulic motors can be stalled for any length of time and their direction of rotation can be instantly reversed and their rotational speed can be infinitely varied.
4. The moment of inertial to torque ratio for a hydraulic motor is nearly 100.
5. A semi-rotary actuator allows only a partial revolution.

Review Questions

1. Differentiate between a hydraulic pump and a hydraulic motor.
2. List the advantages of a hydraulic motor over an electric motor.
3. List four important applications of hydraulic motor.
4. Explain with a neat sketch the working of gear motor.
5. Write the classification of piston motor.
6. Define volumetric efficiency, mechanical efficiency and overall efficiency of hydraulic motor.
7. Why is the actual flow rate required by a hydraulic motor higher than the theoretical flow rate?
8. Why is the actual torque output delivered by a hydraulic motor less than the calculated theoretical torque?
9. List few applications of a semi-rotary actuator.
10. Where are external gear motors used?
11. List the advantages of external gear motors.
12. What is a limited-rotation hydraulic motor? How does it differ from a hydraulic motor?
13. What are the main advantages of a gear motor?
14. Why are vane motors fixed-displacement units?
15. Name one way in which vane motors differ from vane pumps.
16. Can a piston pump be used as a piston motor?
17. Why does a hydraulic motor use more flow than it should theoretically?
18. Name four advantages of hydrostatic transmission.
19. Why does a hydraulic motor deliver less torque than it should theoretically?
20. Explain why, theoretically, the torque output from a fixed-displacement hydraulic motor operating at a constant pressure is the same regardless of changes in speed.
21. Define the displacement and torque ratings of a hydraulic motor.
22. Explain how vanes are held in contact with the cam ring in a high-performance vane motor.
23. How is torque developed in an inline-type piston?

24. If a hydraulic motor is pressure compensated, what is the effect of an increase in the working fluid?
25. Which type of hydraulic motor is generally the most efficient?

Answers
Fill in the Blanks

1.Rotary power, torque
2.Parallel
3.Perpendicular
4.Partial
5. 70–75%, 85–95%

State True or False

1.False
2.False
3.True
4.False
5.True

Lecture 12

HYDRAULIC ACTUATORS

Learning Objectives

Upon completion of this chapter, the student should be able to:

- Explain the classification of hydraulic actuators.
- Explain various types of hydraulic cylinders.
- Describe the construction and working of double-acting cylinders.
- Derive an expression for force, velocity and power for hydraulic cylinders.
- Analyze various lever systems using hydraulic cylinders.
- Explain the importance of cylinder cushioning.
- Explain various types of cylinder mountings used in fluid power.
- Evaluate the performance of hydraulic systems using cylinders.

1.1 Introduction

Hydraulic systems are used to control and transmit power. A pump driven by a prime mover such as an electric motor creates a flow of fluid, in which the pressure, direction and rate of flow are controlled by valves. An actuator is used to convert the energy of fluid back into the mechanical power. The amount of output power developed depends upon the flow rate, the pressure drop across the actuator and its overall efficiency. Thus, hydraulic actuators are devices used to convert pressure energy of the fluid into mechanical energy.
Depending on the type of actuation, hydraulic actuators are classified as follows:

1. Linear actuator: For linear actuation (hydraulic cylinders).
2. Rotary actuator: For rotary actuation (hydraulic motor).
3. Semi-rotary actuator: For limited angle of actuation (semi-rotary actuator).

Hydraulic linear actuators, as their name implies, provide motion in a straight line. The total movement is a finite amount determined by the construction of the unit. They are usually referred to as cylinders, rams and jacks. All these items are synonymous in general use, although ram is sometimes intended to mean a single-acting cylinder and jack often refers to a cylinder used for lifting. The function of hydraulic cylinder is to convert hydraulic power into linear mechanical force or motion. Hydraulic cylinders extend and retract a piston rod to provide a push or pull force to drive the external load along a straight-line path.
Continuous angular movement is achieved by rotary actuators, more generally known as a hydraulic motor. Semi-rotary actuators are capable of limited angular movements that can be several complete revolutions but 360^{o} or less is more usual.

1.2 Types of Hydraulic Cylinders

Hydraulic cylinders are of the following types:

- Single-acting cylinders.
- Double-acting cylinders.
- Telescopic cylinders.
- Tandem cylinders.

1.2.1 Single-Acting Cylinders

A single-acting cylinder is simplest in design and is shown schematically in Fig.1.1. It consists of a piston inside a cylindrical housing called barrel. On one end of the piston there is a rod, which can reciprocate**.** At the opposite end, there is a port for the entrance and exit of oil. Single-acting cylinders produce force in one direction by hydraulic pressure acting on the piston. (Single-acting cylinders can exert a force in the extending direction only.) The return of the piston is not done hydraulically. In single-acting cylinders, retraction is done either by gravity or by a spring.

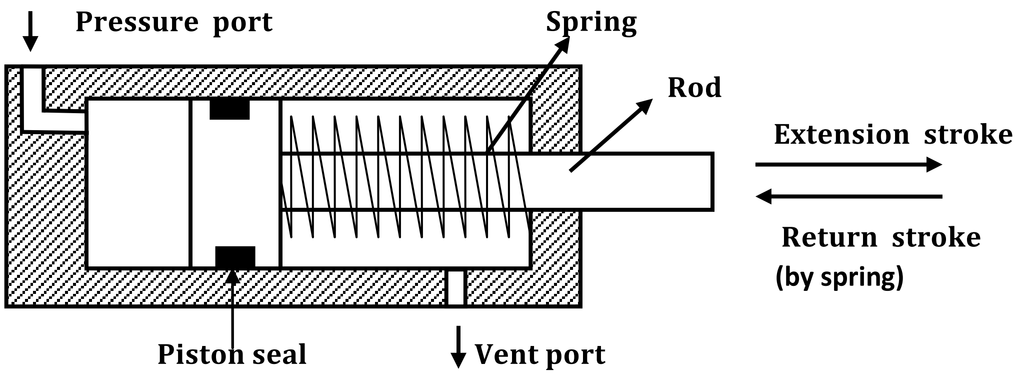


Figure 1.1 Single-acting cylinders

According to the type of return, single-acting cylinders are classified as follows:

- Gravity-return single-acting cylinder.
- Spring-return single-acting cylinder.

1.2.1.1 Gravity-Return Single-Acting Cylinder

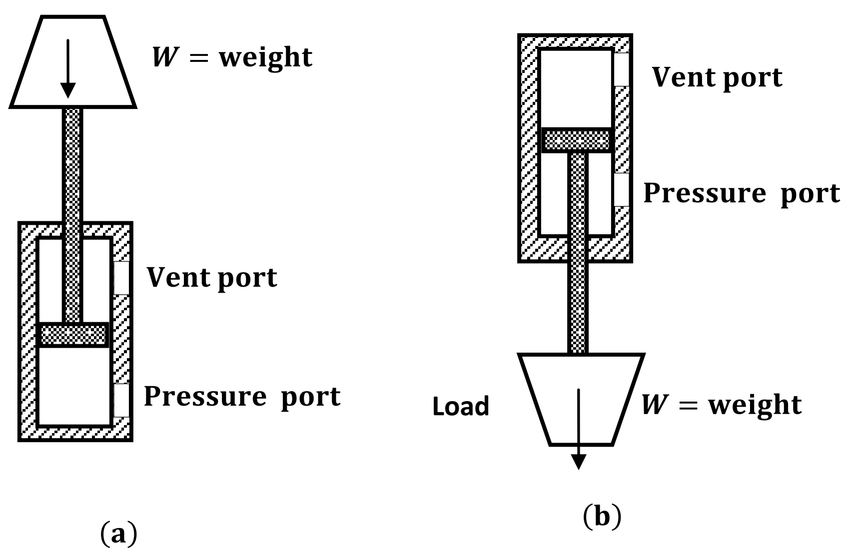


Figure 1.2 Gravity-return single-acting cylinder: (a) Push type; (b) pull type

Figure1.2 shows gravity-return-type single-acting cylinders. In the push type [Fig. 1.2(a)], the cylinder extends to lift a weight against the force of gravity by applying oil pressure at the blank end. The oil is passed through the blank-end port or pressure port. The rod-end port or vent port is open to atmosphere so that air can flow freely in and out of the rod end of the cylinder. To retract the cylinder, the pressure is simply removed from the piston by connecting the pressure port to the tank. This allows the weight of the load to push the fluid out of the cylinder back to the tank. In pull-type gravity-return-type single-acting cylinder, the cylinder [Fig. 1.2(b)] lifts the weight by retracting. The blank-end port is the pressure port and blind-end port is now the vent port. This cylinder automatically extends whenever the pressure port is connected to the tank.

1.2.1.2 Spring-Return Single-Acting Cylinder

A spring-return single-acting cylinder is shown in Fig.1.3.In push type [Fig. 1.3(a)], the pressure is sent through the pressure port situated at the blank end of the cylinder. When the pressure is released, the spring automatically returns the cylinder to the fully retracted position. The vent port is open to atmosphere so that air can flow freely in and out of the rod end of the cylinder.
Figure 1.3(b) shows a spring-return single-acting cylinder. In this design, the cylinder retracts when the pressure port is connected to the pump flow and extends whenever the pressure port is connected to the tank. Here the pressure port is situated at the rod end of the cylinder.

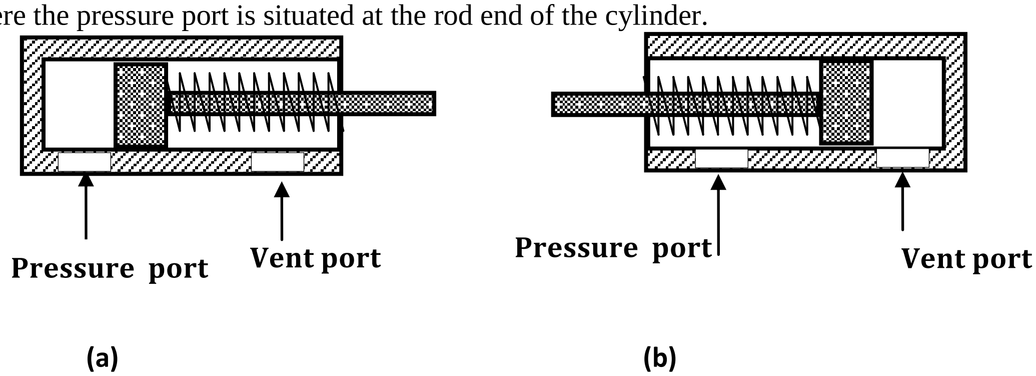


Figure 1.3 (a) Push- and (b) pull-type single-acting cylinders

1.2.2 Double-Acting Cylinder

There are two types of double-acting cylinders:

- Double-acting cylinder with a piston rod on one side.
- Double-acting cylinder with a piston rod on both sides.

1.2.2.1 Double-Acting Cylinder with a Piston Rod on One Side

Figure 1.4 shows the operation of a double-acting cylinder with a piston rod on one side. To extend the cylinder, the pump flow is sent to the blank-end port as in Fig. 1.4(a). The fluid from the rod-end port returns to the reservoir. To retract the cylinder, the pump flow is sent to the rod-end port and the fluid from the blank-end port returns to the tank as in Fig.1.4(b).

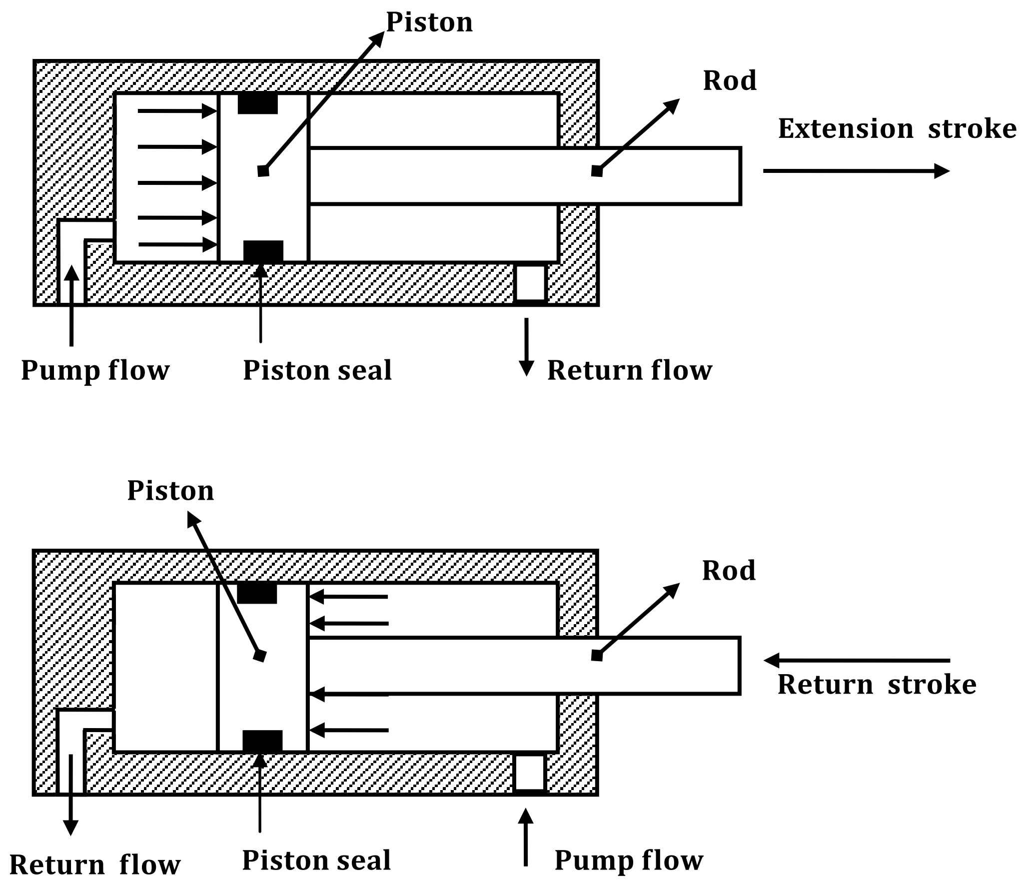


Figure 1.4 Double-acting cylinder with a piston rod on one side

1.2.2.2 **Double-Acting Cylinder with a Piston Rod on Both Sides**

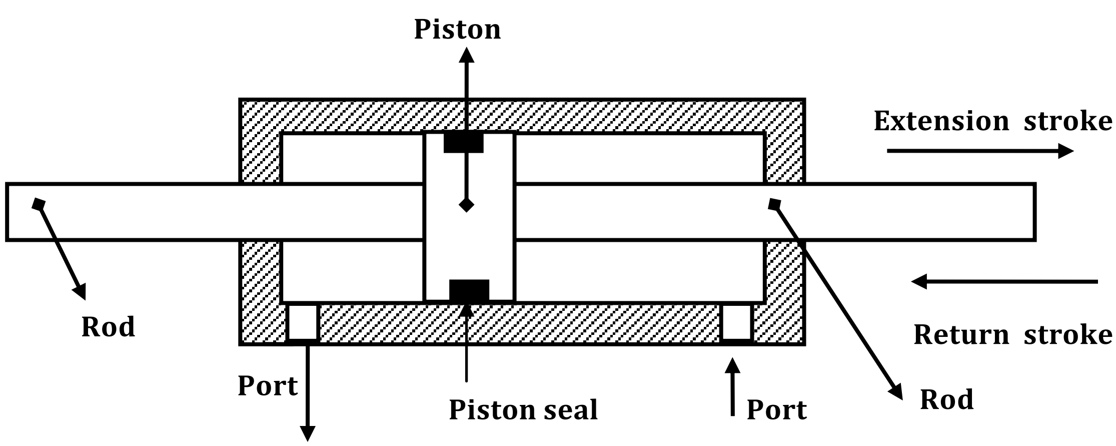


Figure 1.5 Double-acting cylinder with a piston rod on one side

A double-acting cylinder with a piston rod on both sides (Fig.1.5)is a cylinder with a rod extending from both ends. This cylinder can be used in an application where work can be done by both ends of the cylinder, thereby making the cylinder more productive. Double-rod cylinders can withstand higher side loads because they have an extra bearing, one on each rod, to withstand the loading.

1.2.3Telescopic Cylinder

A telescopic cylinder (shown in Fig. 1.6) is used when a long stroke length and a short retracted length are required. The telescopic cylinder extends in stages, each stage consisting of a sleeve that fits inside the previous stage. One application for this type of cylinder is raising a dump truck bed. Telescopic cylinders are available in both single-acting and double-acting models. They are more expensive than standard cylinders due to their more complex construction.

They generally consist of a nest of tubes and operate on the displacement principle. The tubes are supported by bearing rings, the innermost (rear) set of which have grooves or channels to allow fluid flow. The front bearing assembly on each section includes seals and wiper rings. Stop rings limit the movement of each section, thus preventing separation. When the cylinder extends, all the sections move together until the outer section is prevented from further extension by its stop ring. The remaining sections continue out-stroking until the second outermost section reaches the limit of its stroke;this process continues until all sections are extended, the innermost one being the last of all.

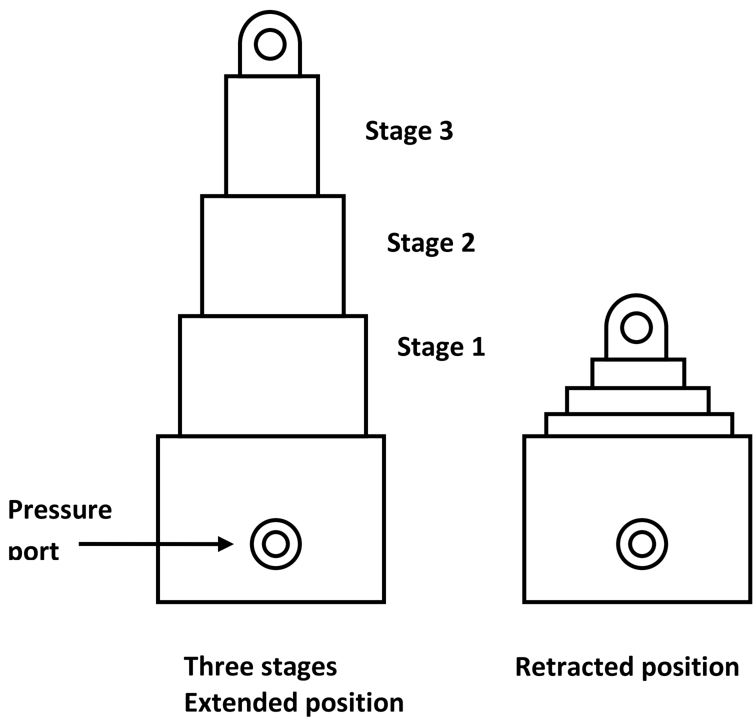


Figure 1.6 Telescopic cylinder

For a given input flow rate, the speed of operation increases in steps as each successive section reaches the end of its stroke. Similarly, for a specific pressure, the load-lifting capacity decreases for each successive section.

1.2.4 Tandem Cylinder

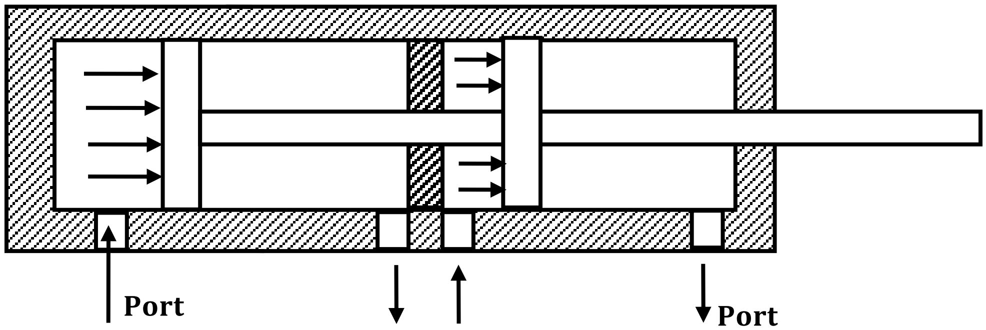


Figure 1.7 Tandem cylinder

A tandem cylinder, shown in Fig. 1.7, is used in applications where a large amount of force is required from a small-diameter cylinder. Pressure is applied to both pistons, resulting in increased force because of the larger area. The drawback is that these cylinders must be longer than a standard cylinder to achieve an equal speed because flow must go to both pistons.

1.2.4.1 Through-Rod Cylinders

These are similar in construction to the standard double-acting cylinders, but have a cylinder rod extending through both cylinder end caps. Although it is possible to have both the piston rods with different diameters at each end of the cylinder, generally the rods have the same diameters. The main applications of through-rod cylinders are as follows: the same speed is required in both the directions, both ends of the rod can be utilized to do work and the non-working end is used to indicate or signal the position of the load. In some applications, the rod is fixed at both the ends and the cylinder body carrying the load moves on the rod.

A major problem in the manufacture of through-rod cylinders is achieving the correct alignment and concentricity of cylinder bore, piston, end caps and rods. Any misalignment can result in excessive seal wear and premature cylinder failure.

1.2.4.2 Displacement Cylinders

A displacement-type hydraulic cylinder shown in Fig. 1.8 consists of a rod that is displaced from inside a tube by pumping hydraulic fluid into the tube. The volume of the rod leaving the tube is equal to the volume of fluid entering the tube, hence the name “displacement cylinder.”

The rod of the displacement cylinder is guided by bearings in the nose or neck of the cylinder body. A collar on the end of the rod prevents it from being ejected and limits the stroke of the cylinder. Elastomer seals in the neck prevent any leakage of fluid along the outside of the rod. This design is a single-acting “push” or extension cylinder, which has to be retracted by gravity, a spring or some external force. The bore of the cylinder body does not require machining other than that for the neck bearing and the inlet port; the manufacturing cost is, therefore, low when compared with other types or hydraulic cylinders.

The maximum thrust exerted by a displacement cylinder is given by

$$\text{Maximum thrust} = \text{Pressure} \times \text{Rod area} = p \times \frac{\pi d^2}{4}$$

where *d* is the diameter of the rod. The extending speed of the rod is given by

$$\text{Rod speed} = \frac{\text{Flow rate of fluid entering the cylinder}}{\text{Area of cylinder rod}}$$

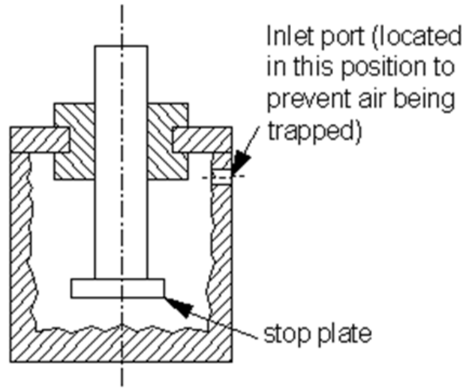


Figure 1.8 Displacement cylinders

Example 1.1
A displacement-type cylinder has a rod of 65 mm diameter and is powered by a hand pump with a displacement of 5 mL per double stroke. The maximum operating pressure of the system is to be limited to 350 bar. (a) Draw a suitable circuit diagram showing the cylinder, pump and any additional valving required. (b) Calculate the number of double pumping strokes needed to extend the cylinder rod by 50 mm. (c) Calculate the maximum load that could be raised using this system.

Solution:
(a) The circuit diagram is given in Fig. 1.9.

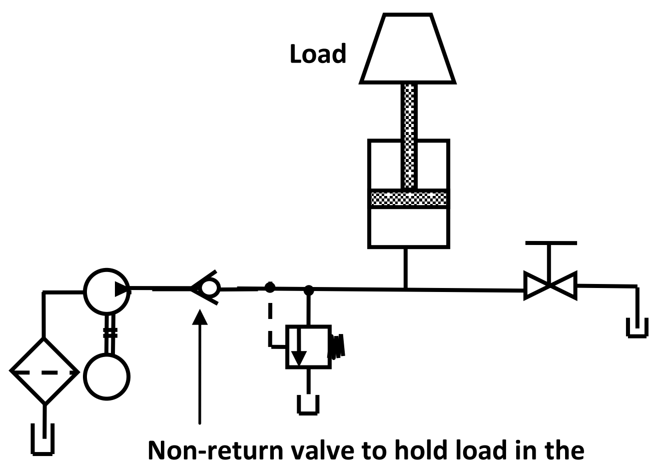


Figure 1.9
(b) The volume of rod displaced is equal to the volume of fluid entering the cylinder. Let the rod diameter be d, the distance rod extends be L, the displacement per double stroke of pump be V and the number of double pump strokes be S. Then

Rod volume displaced = Fluid volume entering

$$\Rightarrow \frac{d^2}{4} \times L = V \times S$$

Substituting values given in the problem and showing units for each value we get

$$\frac{65^2}{4}\ \mathrm{mm}^2 \times 50\ \mathrm{mm} = 5\ \mathrm{mL} \times S$$

$$\Rightarrow \frac{65^2}{4} \times 50\ \mathrm{mm}^3 = 5S\ \mathrm{mL}$$

The units on both sides of the equation must be the same.

Note:

$1\ \mathrm{mL} = 1 \times 10^{-3}\ \mathrm{L}$

$1\ \mathrm{L} = 1 \times 10^{-3}\ \mathrm{m}^3$

$1\ \mathrm{mL} = 1 \times 10^{-6}\ \mathrm{m}^3$

$1\ \mathrm{mm}^3 = 1 \times 10^{-9}\ \mathrm{m}^3$

Thus, for the dimensional equality

$$\frac{65^2}{4} \times 50 \times 10^{-9}(\mathrm{m}^3) = 5\times 10^{-6}(\mathrm{m}^3) \times S$$

or $$\frac{65^2}{4} \times 50\ (\mathrm{m}^3) = 5\times 10^{3}\ (\mathrm{m}^3) \times S$$

Therefore,

$$S = \frac{65^2 \times 50}{4\times 5\times 10^3} = 33.17 \text{ double strokes}$$

(c) We have

$$\text{Maximum thrust} = \text{Pressure} \times \text{Rod area}$$

Substituting the values given into the problem and showing units we get

$$\text{Maximum thrust} = 350 \times 10^5 \times \frac{65^2}{4} \times 10^{-6}\left(\frac{\mathrm{N}}{\mathrm{m}^2}\times \mathrm{m}^2\right)$$

$$= 35 \times \frac{65^2}{4}\ \mathrm{N} = 116080\ \mathrm{N} = 116.080\ \mathrm{kN}$$

Example 1.2

A three-stage displacement-type telescopic cylinder is used to tilt the body of a lorry (Fig. 1.10). When the lorry is fully laden, the cylinder has to exert a force equivalent to 4000 kg at all points in its stroke. The outside diameters of the tubes forming the three stages are 60, 80 and 100 mm. If the pump powering the cylinder delivers 10 LPM, calculate the extend speed and pressure required for each stage of the cylinder when tilting a fully laden lorry.

Solution:

First-stage

$$\text{First-stage diameter} = 100\ \mathrm{mm}$$

$$\text{First-stage speed} = \frac{\text{Quantity flowing}}{\text{Area}}$$

$$= \frac{10\times 10^{-3}}{(\pi/4)\times(0.1)^2}\left(\frac{\mathrm{m}^3}{\mathrm{min}\times \mathrm{m}^2}\right) = \frac{4}{\pi} = 1.27\ \mathrm{m/min}$$

$$\text{First-stage pressure} = \frac{\text{Load}}{\text{Area}}$$

$$= \frac{4000 \times 9.81}{(\pi/4) \times (0.1)^2}\ \text{N/m}^2 = 5 \times 10^6\ (\text{N/m}^2) = 50\ \text{bar}$$

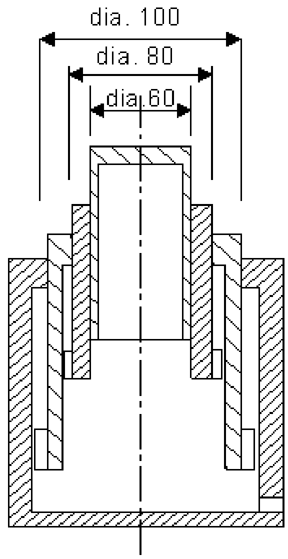


Figure 1.10

Second-stage:

Second-stage diameter = 80 mm

$$\text{Second-stage speed} = \frac{\text{Quantity flowing}}{\text{Area}}$$

$$= \frac{10 \times 10^{-3}}{(\pi/4) \times (0.08)^2} \left(\frac{\text{m}^3}{\text{min} \times \text{m}^2} \right) = 1.99\ \text{m/min}$$

$$\text{Second-stage pressure} = \frac{\text{Load}}{\text{Area}}$$

$$= \frac{4000 \times 9.81}{(\pi/4) \times (0.08)^2}\ \text{N/m}^2 = 7.81 \times 10^6 (\text{N/m}^2) = 78.1 \text{bar}$$

Third-stage:

Third-stage diameter = 60 mm

$$\text{Third-stage speed} = \frac{\text{Quantity flowing}}{\text{Area}}$$

$$= \frac{10 \times 10^{-3}}{(\pi/4) \times (0.06)^2} \left(\frac{\text{m}^3}{\text{min} \times \text{m}^2} \right)$$

$$= 3.54\ \text{m/min}$$

$$\text{Third-stage pressure} = \frac{\text{Load}}{\text{Area}}$$

$$= \frac{4000 \times 9.81}{(\pi/4) \times (0.06)^2}\ \text{N/m}^2 = 13.9 \times 10^6 = 139\ \text{bar}$$

Telescopic cylinders are made in a standard range for vehicle applications. Although non-standard cylinders can be obtained, they tend to be very expensive if ordered as a single piece.

1.3Standard Metric Cylinders

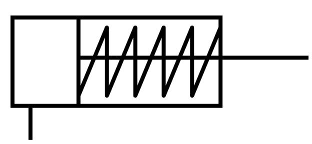
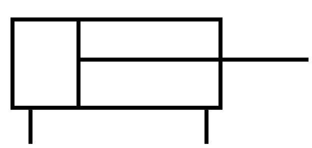
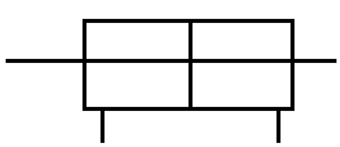
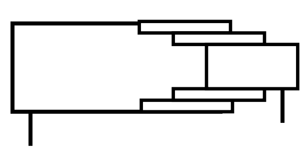
Table 1.1gives preferred sizes for the cylinder bore and rod diameter of metric cylinders. Most cylinder manufacturers have based their standard range of metric cylinders on these recommendations, offering two rod sizes for each cylinder bore.

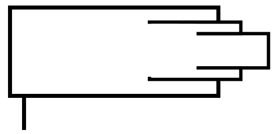
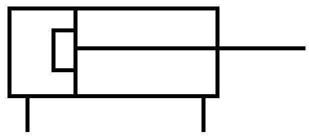
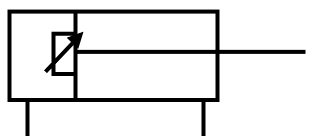
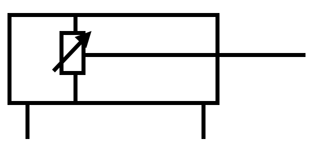
A number of combinations have a piston rod to piston diameter ratio in the region of 0.7, which gives an annulus area of approximately one-half of the full bore area. This area ratio is of use in regenerative circuits to give similar values of speed and thrust on both the extension and retraction strokes. Table 1.2 gives the graphical symbols for various kinds of cylinders.

Table 1.1 Recommended cylinder bore and rod sizes

Piston diameter (mm)		40	50	63	80	100	125	140	160	180	200	220	250	280	320
Piston rod diameter (mm)	Small	20	28	36	45	56	70	90	100	110	125	140	160	180	200
	Large	28	36	45	56	70	90	100	110	125	140	160	180	200	220

Table 1.2 Graphical symbols of different linear actuators

S. No.	**Graphical Symbols**	**Explanation**
1.		Single-acting cylinder with unspecified return
2.		Single-acting cylinder with spring return
3.		Double-acting cylinder –single piston rod
4.		Double-acting cylinder –doublepiston rod
5.		Telescopic cylinder–double acting

6.		Telescopic cylinder–single acting
7.		Double-acting cylinder– fixed cushion on one side
1.		Double-acting cylinder–variable cushion on one side
9.		Double-acting cylinder–variable cushion on both sides

1.4Cylinder Force, Velocity and Power

The output force (*F*) and piston velocity (*v*) of double-acting cylinders are not the same for extension and retraction strokes.

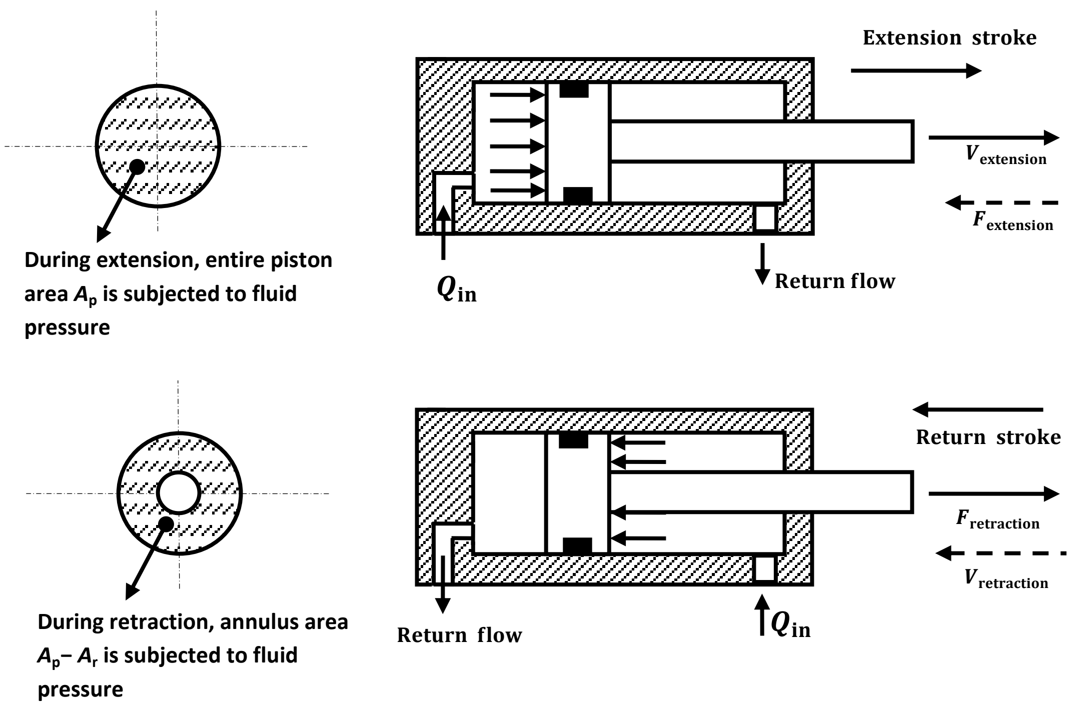


Figure 1.11 Effective area during (a) extension strokes and (b)retraction strokes

During the extension stroke shown in Fig.1.11(a),the fluid pressure acts on the entire circular piston area A_p. During the retraction stroke, the fluid enters the rod-end side and the fluid pressure acts on the smaller annular area between the rod and cylinder bore (A_p-A_r) as shown by the shaded area in Fig.1.11(b) (A_r is the area of the piston rod). Due to the difference in the cross-sectional area, the velocity of the piston changes.Because $A_p$is greater than (A_p-A_r), the retraction velocity (v_{ret}) is greater than the extension velocity (v_{ext}) for the same flow rate.

During the extension stroke, the fluid pressure acts on the entire piston area (A_r), while during the retraction stroke, the fluid pressure acts on the annular area (A_p-A_r). This difference in area accounts for the difference in output forces during extension and retraction strokes. Because A_r is greater than A_p-A_r), the extension force is greater than the retraction force for the same operating pressure.
Force and velocity during extension stroke

Velocity $$v_{ext} = \frac{Q_{in}}{A_p}$$

Force
$$F_{ext} = p \times A_p$$

Force and velocity during retraction stroke
Velocity

$$v_{ext} = \frac{Q_{in}}{A_p - A_r}$$

Force,

$$F_{ext} = p \times (A_p - A_r)$$

Power developed by a hydraulic cylinder (both in extension and retraction) is

$$\text{Power} = \text{Force} \times \text{Velocity} = F \times V$$

In metric units, the kW power developed for either extension or retraction stroke is

$$\begin{aligned} \text{Power (kW)} &= v_p\,(\text{m/s}) \times F\ (\text{kN}) \\ &= Q_{in}\,(\text{m}^3/\text{s}) \times p\ (\text{kPa}) \end{aligned}$$

Power during extension is

$$P_{ext} = F_{ext} \times v_{ext} = p \times A_p \times \frac{Q_{in}}{A_p} = p \times Q_{in} \tag{1.1}$$

Power during retraction is

$$\begin{aligned} P_{ret} &= F_{ret} \times v_{ret} \\ &= p \times (A_p - A_r) \times \frac{Q_{in}}{A_p - A_r} \\ &= p \times Q_{in} \end{aligned} \tag{1.2}$$

Comparing Equation. (1.1) and (1.2), we can conclude that the powers during extension and retraction strokes are the same.

Lecture 13

HYDRAULIC ACTUATORS[CONTINUED]

1.5Acceleration and Deceleration of Cylinder Loads

Cylinders are subjected to acceleration and deceleration during their operation. Cylinders are decelerated to provide cushioning and cylinders are accelerated to reduce the cycle time of the operation.

1.5.1 Acceleration

To calculate the acceleration of cylinder loads, the equations of motion must be understood.Let u be the initial velocity, v the velocity after a time t,s the distance moved during the time t and$a$ the acceleration during the time t.The standard equations of motion are as follows:

$$v = u + at$$

$$v^2 = u^2 + 2as$$

$$s = ut + \frac{1}{2}at^2$$

and

$$s = \frac{1}{2}(u + v)t$$

The force F to accelerate a weight W horizontally with an acceleration a is given by

$$\text{Force} = \text{Mass} \times \text{Acceleration}$$

$$F = \frac{W}{g}a$$

where g is the acceleration due to gravity and is 9.81 m/s^2. The force P required to overcome friction is given by $P = \mu W$, where μ is the coefficient of friction.

Note: Dynamic cylinder thrust

In dynamic applications, the load inertia, seal friction, load friction, etc., must be allowed for calculating the dynamic thrust.As a first approximation, the dynamic thrust can be taken as 0.9 times the static thrust. Cylinder seal friction varies with seal and cylinder design. The pressure required to overcome seal friction is not readily available from the majority of cylinder manufacturers. The seal friction breakout pressure can be taken as 5 bar for calculation purposes. It reduces when the piston starts to move. The pressure required to overcome seal friction reduces as the cylinder bore size increases and varies according to the seal design.

Example 1.3

A cylinder is required to move a 10 kN load 150 mm in 0.5 s. What is the output power?

Solution: The velocity is given by

$$v = \frac{d}{t} = \frac{0.15}{0.5} = 0.3 \text{ m/s}$$

Power is given by

$$P = F \times v = 10 \times 10^3 \times 0.3 = 3 \text{ kW}$$

Example 1.4

A cylinder is required to extend at a minimum speed of 0.75 m/s in a system with a flow rate of 60 LPM. What cylinder size is required?

Solution:
Let us first convert the LPM to m^3/s:

$$Q = 60\text{ LPM} = \frac{60}{1000}\text{ m}^3/\text{min} = \frac{60}{1000\times 60} = 10^{-3}\text{ m}^3/\text{s}$$

Now we know flow rate and velocity, so we can calculate the diameter,

$$Q = A_p \times v$$

$$\Rightarrow 10^{-3} = \frac{\pi}{4} D_p^2 \times 0.75$$

Therefore,

$$D_p = \sqrt{\frac{4\times 10^{-3}}{\pi \times 0.75}} = 41.2\text{ mm}$$

Example 1.5
An 8 cm diameter hydraulic cylinder has a 4 cm diameter rod. If the cylinder receives flow at 100 LPM and 12 MPa, find the (a) extension and retraction speeds and (b) extension and retraction load carrying capacities.

Solution:
Let us first convert the flow in LPM to m^3/s before we calculate forward velocityQ_{in}=100 LPM = 100/(1000 × 60) =1/600 m^3/s
Now
D_C = Diameter of cylinder = 8 cm = 8 × 10^{-2} m
d_r = Diameter of piston rod = 4 cm = 4 × 10^{-2} m
p = 12 MPa = 12 × 10^6 N/m^2 or Pa

(a) Forward velocity is given by

$$v_{ext} = \frac{Q_{in}}{A_p} = \frac{1/600}{\pi d^2/4} = 0.3315\text{ m/s}$$

Return velocity is given by

$$v_{ret} = \frac{Q_{in}}{(A_p - A_r)} = \frac{\frac{1}{600}}{\frac{\pi(d_C^2 - d_r^2)}{4}} = 0.442\text{ m/s}$$

(b) Force during extension is given by

$$F_{ext} = p \times a_p = 12\times 10^6\,\frac{\pi(8\times 10^{-2})^2}{4} = 60318.57\text{ N}$$

Force during retraction is given by

$$
\begin{aligned}
F_{\text{ret}} &= p \times (A_{\text{p}} - A_{\text{r}}) \\
&= 12 \times \frac{10^6 \times \pi[(8\times 10^{-2})^2 - (4\times 10^{-2})^2]}{4} \\
&= 42238.9\ \text{N} = 45.24\ \text{kN}
\end{aligned}
$$

Example 1.6

A pump supplies oil at 0.0016 m^3/s to a 40 mm diameter double-acting hydraulic cylinder. If the load is 5000 N (extending and retracting) and the rod diameter is 20 mm, find the

(a) Hydraulic pressure during the extending stroke.
(b) Piston velocity during the extending stroke.
(c) Cylinder kW power during the extending stroke.
(d) Hydraulic pressure during the retracting stroke,
(e) Piston velocity during the retracting stroke.
(f) Cylinder kW power during the retracting stroke.

Solution: We have $Q_{\text{in}} = 0.016\ \text{m}^3/\text{s}$, $F_{\text{ext}} = F_{\text{ret}} = 5000$ N, $d_{\text{C}} = 40$ mm = 0.04 m, $d_{\text{r}} = 20$ mm = 0.02 m.

(a) Hydraulic pressure during the extending stroke is

$$
p_{\text{ext}} = \frac{F_{\text{ext}}}{A_{\text{p}}} = \frac{5000}{\frac{\pi d_{\text{C}}^2}{4}} = \frac{5000}{\frac{\pi (0.04)^2}{4}} = 3978.8\,\text{kPa}
$$

(b) Piston velocity during the extending stroke is

$$
v_{\text{ext}} = \frac{Q_{\text{in}}}{A_{\text{p}}} = \frac{Q_{\text{in}}}{\frac{\pi d_{\text{C}}^2}{4}} = \frac{0.016}{\frac{\pi (0.04)^2}{4}} = 1.273\ \text{m/s}
$$

(c) Cylinder kW power during the extending stroke is

$$
P_{\text{ext}} = F_{\text{ext}} \times v_{\text{ext}} = 5000 \times 1.273 = 6.366\,\text{kW}
$$

(d) Hydraulic pressure during the retracting stroke is

$$
p_{\text{ret}} = \frac{F_{\text{ret}}}{A_{\text{p}} - A_{\text{r}}} = \frac{5000}{\frac{\pi (d_{\text{C}}^2 - d_{\text{r}}^2)}{4}} = \frac{5000}{\frac{\pi [(0.04)^2 - (0.02)^2]}{4}} = 5305.16\ \text{kPa}
$$

(e) Piston velocity during the retracting stroke is

$$
v_{\text{ret}} = \frac{Q_{\text{in}}}{A_{\text{p}} - A_{\text{r}}} = \frac{0.016}{\frac{\pi (0.04^2 - 0.02^2)}{4}} = 1.697\ \text{m/s}
$$

(f) Cylinder kW power during the retracting stroke is

$$
P_{\text{ret}} = F_{\text{ret}} \times v_{\text{ret}} = 5000 \times 1.697 = 8.488\ \text{kW}
$$

Example 1.7

A hydraulic cylinder has a rod diameter equal to one half the piston diameter. Determine the difference in load-carrying capacity between extension and retraction stroke if pressure is constant. What would happen if the pressure were applied to both sides of the cylinder at the same time?

Solution: Forward or extending stroke is

$$
F_{\text{ext}} = p \times A_{\text{p}}
$$

Retracting stroke is

$$F_{\text{ret}} = p \times (A_{\text{p}} - A_{\text{r}})$$

Also, given that

$$d_{\text{p}} = 2d_{\text{r}}$$

Therefore

$$A_{\text{p}} = 4A_{\text{r}}$$

Now

$$\frac{F_{\text{ext}}}{F_{\text{ret}}} = \frac{p \times A_{\text{p}}}{p \times (A_{\text{p}} - A_{\text{r}})}$$

$$= \frac{A_{\text{p}}}{A_{\text{p}} - A_{\text{r}}}$$

$$= \frac{A_{\text{p}}}{A_{\text{p}} - 0.25A_{\text{p}}}$$

Therefore,

$$\frac{F_{\text{ext}}}{F_{\text{ret}}} = \frac{4}{3}$$

Again

$$F_{\text{ext}} - F_{\text{ret}} = pA_{\text{p}} - p(A_{\text{p}} - 0.25A_{\text{p}})$$

$$= pA_{\text{p}} - p\left\{A_{\text{p}} - \frac{A_{\text{p}}}{4}\right\}$$

$$\Rightarrow F_{\text{ext}} - F_{\text{ret}} = \frac{pA_{\text{p}}}{4}$$

Therefore,

$$\text{Difference} = \frac{\text{Pressure} \times \text{Piston area}}{4}$$

If the pressure were applied to both sides of the cylinder at the same time, there would be a net force to extend the cylinder. This net force will be the same as obtained above:

$$F_{\text{net-extending}} = \frac{\text{Pressure} \times \text{Piston area}}{4}$$

Example 1.8

A cylinder with a bore of 150 mm and a piston rod diameter of 105 mm, has to extend with a speed of 7 m/s, pressure applied is 150 bar. Calculate

(a) The flow rate in LPM of oil to extend the cylinder

(b) The flow rate in LPM from annulus side to extend the cylinder.

(c) The retract speed in m/min using (a).

(d) The flow rate from full bore end on retract.

Solution: Area of piston is given by

$$A_{\text{p}} = \frac{\pi}{4} \times D_{\text{p}}^2 = \frac{\pi}{4} \times (0.15)^2 = 0.01767 \text{ m}^2$$

Area of rod is

$$A_{\text{r}} = \frac{\pi}{4} \times D_{\text{r}}^2 = \frac{\pi}{4} \times (0.105)^2 = 0.008659 \text{ m}^2$$

Hence, $A_\mathrm{p} - A_\mathrm{r} = 0.00901\,\mathrm{m}^2$.

Given, v_{ext} = 7 m/min and p = 150 bar.

(a) Flow rate of oil to extend in LPM:

$$\begin{aligned}(Q_{\mathrm{ext}})_\mathrm{p} &= A_\mathrm{p} \times v_{\mathrm{ext}} \\ &= 0.01767 \times 7 \\ &= 0.12369\,\mathrm{m}^3/\mathrm{min} = 123.69\,\mathrm{LPM}\end{aligned}$$

(b) Flow rate of oil to extend from annulus in LPM:

$$\begin{aligned}(Q_{\mathrm{ext}})_\mathrm{a} &= (A_\mathrm{p} - A_\mathrm{r}) \times v_{\mathrm{ext}} \\ &= 0.00901 \times 7 \\ &= 0.06307\,\mathrm{m}^3/\mathrm{min} = 63\,\mathrm{LPM}\end{aligned}$$

(c) Retract speed in m/min using (a):

$$Q_{\mathrm{ret}} = (Q_{\mathrm{ext}})_\mathrm{p} = (A_\mathrm{p} - A_\mathrm{r}) \times v_{\mathrm{ret}}$$

$$\Rightarrow 0.12369 = 0.00901 \times v_{\mathrm{ret}}$$

$$\Rightarrow v_{\mathrm{ret}} = 13.728\,\mathrm{m/min}$$

(d) Flow rate from full bore end on retract:

$$\begin{aligned}Q_{\mathrm{ret}} &= A_\mathrm{p} \times v_{\mathrm{ret}} \\ &= 0.01767 \times 13.728 \\ &= 0.06587\,\mathrm{m}^3/\mathrm{min} = 65.87\,\mathrm{LPM}\end{aligned}$$

1.6Various Methods of Applying Linear Motion Using Hydraulic Cylinders

A cylinder must produce a force equal to the load the cylinder is required to overcome. A cylinder may be placed with its axis vertical, horizontal or inclined depending on the load to be actuated.

1. **Vertical cylinder:** Ina vertical cylinder, the load to be actuated is in the vertical direction as shown in Fig. 1.12. Then the cylinder load F is equal to the weight W of the object, acting in the vertical direction.

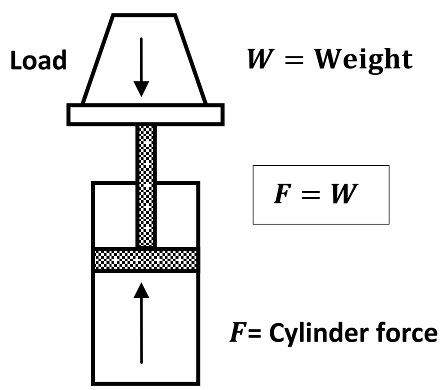


Figure 1.12 Cylinder load– vertical cylinder

2. **Horizontal cylinder:**The schematic diagram of horizontal cylinder is shown in Fig. 1.13. Ina horizontal cylinder, the cylinder load is theoretically zero,because no component of the object's weight acts along the axis of the cylinder. However, when the object slides across the horizontal surface, the cylinder must overcome the frictional force created between the object and the horizontal surface.

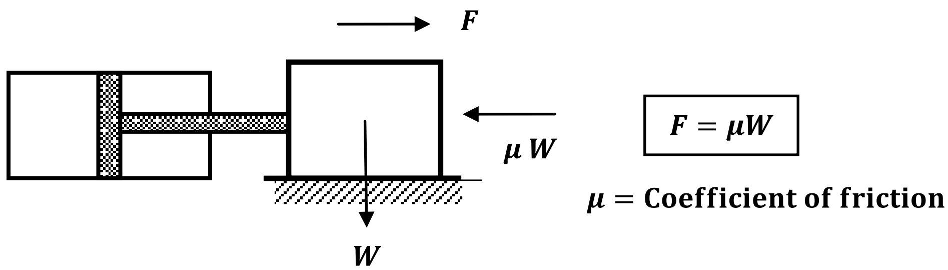


Figure 1.13 Cylinder load –horizontal cylinder

3. **Inclined cylinder:**In an inclined cylinder as shown in Fig. 1.14, the cylinder load equals the component of the object's weight acting along the axis of the cylinder and frictional force

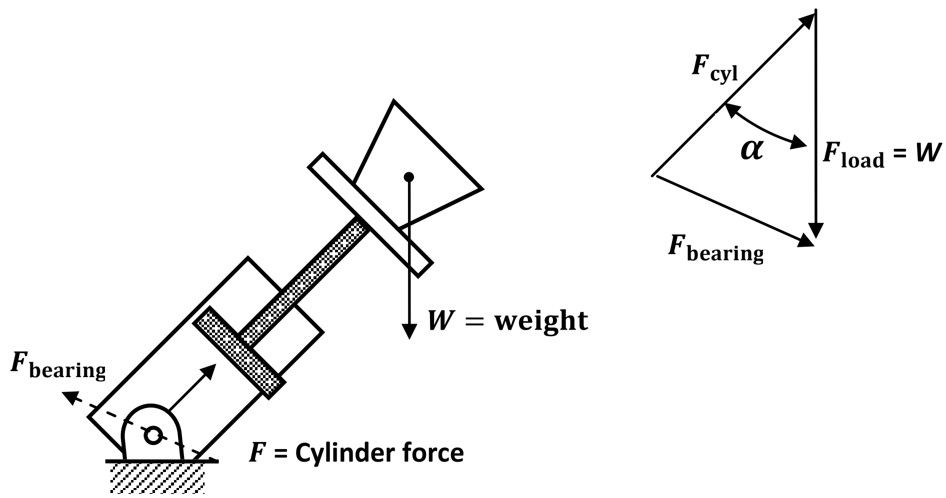


Figure 1.14 Cylinder load –inclined cylinder

For an inclined cylinder, the load the cylinder must overcome is less than the weight of the object to be moved if the object does not slide on an inclined surface. The cylinder loads calculated as above are based on moving an object at a constant velocity. But when the object has to be accelerated from zero velocity to a steady-state velocity, an additional force called inertia force must be added to the weight component and any frictional force involved.

Let$F_{load} = W$ = weight or load acting vertically downward, F_{cyl} = load acting on the cylinder, F_{bear} = force on the bearings and α = angle between the load W and the axis of the cylinder.Then,

$$F_{cyl} = F_{load}\cos(a)$$
$$F_{bear} = F_{load}\sin(a)$$

Example 1.9

Find the cylinder force required to move a 6000 N weight along a horizontal surface at a constant velocity (Fig. 1.15). The coefficient of friction between the weight and horizontal support surface is 0.14.

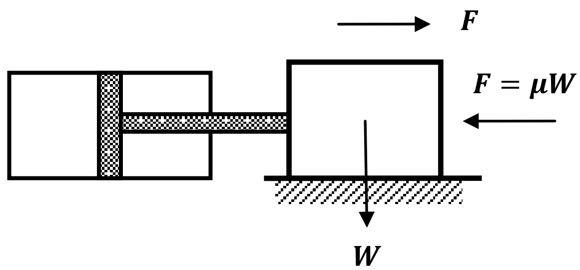


Figure 1.15

Solution: The cylinder force is given by

$$F_{\text{cyl}} = \text{Frictional force } (F_{\text{f}}) = \mu W$$
$$\Rightarrow F_{\text{cyl}} = 0.14 \times 6000 = 840 \text{ N}$$

Example 1.10

Find the cylinder force required to lift a 6000 N weight along a direction that is 30° from the horizontal direction as shown in Fig. 1.16. The weight is moved at a constant velocity.

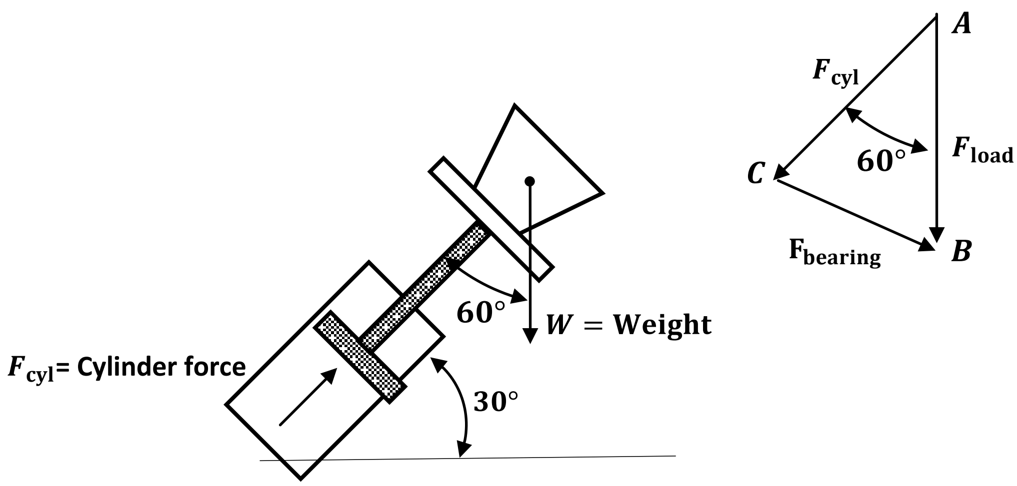


Figure 1.16

Solution: Let

$F_{\text{load}} = W$ = weight or load acting vertically downward

F_{cyl} = load acting on the cylinder

F_{bear} = force on the bearings

α= angle between the load W and the axis of the cylinder

Then

$$F_{\text{cyl}} = F_{\text{load}}\cos\alpha$$
$$F_{\text{bear}} = F_{\text{load}}\sin\alpha$$

Hence,

$$F_{\text{cyl}} = 6000 \times \cos 60° = 3000 \text{ N}$$

Example 1.11

A 6000 N weight is to be lifted upward in a vertical direction for the system shown in Fig. 1.17. Find the cylinder force required to

(a) Move the weight at a constant velocity of 1.75 m/s.

(b) Accelerate the weight from zero velocity to 1.75 m/s in 0.5 s.

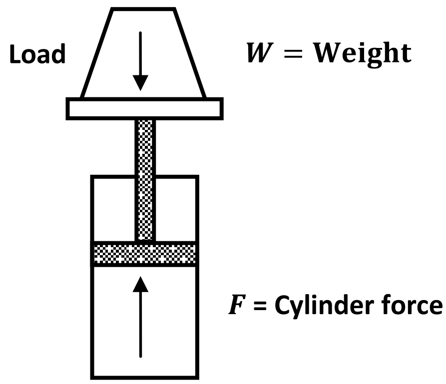


Figure 1.17

Solution:

(a) For a constant velocity, the cylinder force to move weight at a constant velocity of 1.75 m/s

$$F_{\text{cyl}} = F_{\text{a}} = W = 6000 \text{ N}$$

(b) Force required to accelerate the weight:

First we shall calculate acceleration

$$a = \frac{1.75 - 0}{0.5} = 3.5 \text{ m/s}^2$$

Force required to accelerate the weight is

$$F_{\text{a}} = m \times a = \frac{6000 \times 3.5}{9.81} = 2140.67 \text{ N}$$

The cylinder force F_{cyl} required is equal to the sum of the weight and acceleration force

$$F_{\text{cyl}} = 6000 + 2140.67 = 8140.67 \text{ N}$$

Example 1.12

A 10000 N weight is to be lowered by a vertical cylinder as shown in Fig. 1.18. The cylinder has a 75 mm diameter piston and 50 mm diameter rod. The weight is to decelerate from 100 m/min to a stop in 0.5 s. Determine the required pressure in the rod end of the cylinder during the deceleration motion.

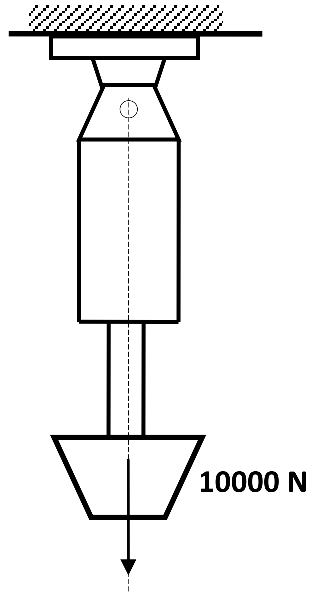


Figure 1.18

Solution: As per Newton's law of motion, we have

$$\sum F = m\ a$$

where

$$a = \frac{100\dfrac{\text{m}}{\text{min}} \times \dfrac{1\,\text{min}}{60\,\text{s}}}{0.5\,\text{s}} = \frac{1.67\,\text{m/s}}{0.5\,\text{s}} = 3.34\,\text{m/s}^2$$

Summing forces on the 10000 N weight, we have

$$p(A_\text{p} - A_\text{r}) - 10000\,\text{N} = \frac{10000\,\text{N}}{9.81\,\text{m/s}^2} \times 3.34\ \text{m/s}^2$$

$$\Rightarrow p\ (\text{N/m}^2) \times \frac{\pi}{4}(0.075^2 - 0.050^2)\ \text{m}^2 - 10000\,\text{N} = 3408\,\text{N}$$

Solving we get

$$p = 5450000\ (\text{N/m}^2) = 5450000\,\text{Pa} = 5450\,\text{kPa}$$

Example 1.13

A 27000 N weight is being pushed up on an inclined surface at a constant speed by a cylinder, as shown in Fig. 1.19. The coefficient of friction between the weight and the inclined surface equals 0.15.

(a) Determine the required cylinder piston diameter for the pressure of 6894 kPa,

(b) Determine the required cylinder piston diameter, if the weight is to accelerate from a 0 mm/s to a 1524 mm/s in 0.5 s.

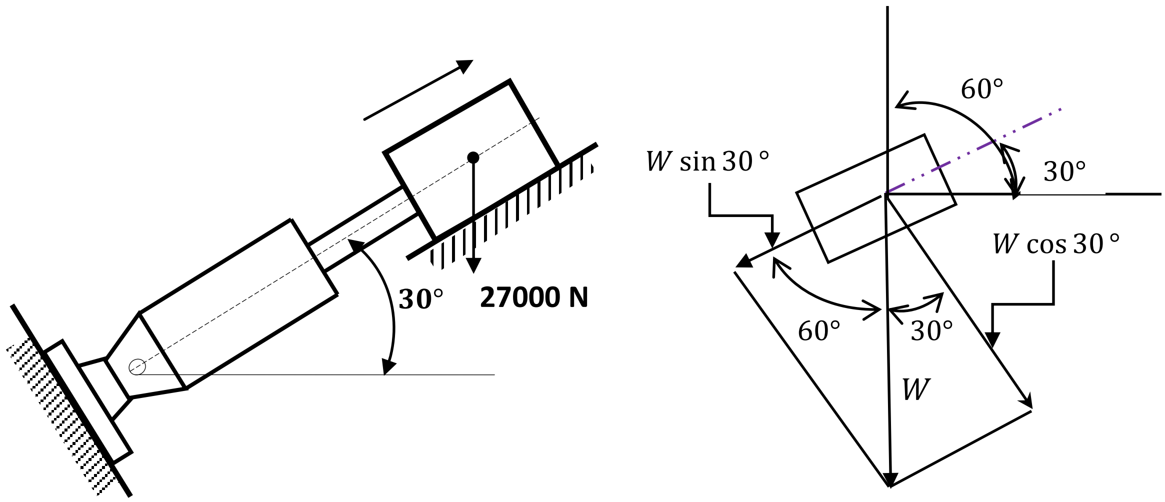


Figure 1.19

Solution:

(a) Cylinder piston diameter for the pressure of 6894 kPa

The component of the weight W acting along the axis of the cylinder is $W \sin 30°$. The component of weight W acting along normal to the incline surface is $W \cos 30°$. The frictional force equals the coefficient of friction times the force normal to sliding surfaces.

Therefore, the frictional force f acting along the axis of the cylinder is

$$\begin{aligned} f &= \mu \times W \cos 30° \\ &= 0.15 \times 27000 \times \cos 30° \\ &= 3507\ \text{N} \end{aligned}$$

Total force on the cylinder is frictional force and vertical component:

$$\begin{aligned} F_{\text{cylinder}} &= f + W \sin 30° \\ &= 3507 + 27000 \times \sin 30° \\ &= 17007\ \text{N} \end{aligned}$$

$$F_{\text{cylinder}} = pA_{\text{p}} = 17007\ \text{N}$$

Diameter to resist 17007 N is given by

$$F_{\text{cylinder}} = 6894000 \times \frac{\pi}{4} D^2 = 17007\ \text{N}$$

$$\Rightarrow D = 0.05605\ \text{m} = 56\ \text{mm}$$

(b) Cylinder piston diameter if the weight is to accelerate from a 0 mm/s to a 1524 mm/s in 0.5 s

As per Newton's law of motion, we have calculate the acceleration

$$\sum F = m\ a$$

where

$$a = \frac{1524\,\text{mm/s}}{0.5\text{s}} = 3048\ \text{mm/s}^2 = 3.048\ \ \text{m/s}^2$$

Summing forces on the 27000 N weight using values determined in (a), we have

$$pA_p - 17007 = \frac{27000 \times 3.048}{9.81}$$

$$\Rightarrow \qquad 6894000 \times \frac{\pi}{4} D^2 - 17007 = \frac{27000 \times 3.048}{9.81}$$

$$\Rightarrow \qquad 5414535 D^2 - 17007 = 8389$$

$$\Rightarrow D = 0.0685\ \text{m} = 68.5\ \text{mm}$$

Example 1.14

A hydraulic cylinder has a bore of 200 mm and a piston rod diameter of 140 mm. For an extend speed of 5 m/min, calculate

(a) The supply flow rate.
(b) The flow rate from the annulus side on extend.
(c) The retract speed using Q_E.
(d) The flow rate from the full bore end on retract.

Also, if the maximum pressure applied to the cylinder is 100 bar, calculate the (e) dynamic extend thrust and the (f) dynamic retract thrust assuming that dynamic thrust = 0.9 $\times$ static thrust. Moreover, the hydraulic cylinder having a bore of 200 mm diameter and a rod of 140 mm diameter are connected regeneratively. (g) If the same flow rate of 157 L/min is used, calculate the extend speed. (h) If the maximum system pressure is 100 bar, calculate the dynamic extend thrust.

Solution:

(a) Flow rate of oil to extend cylinder at 5 m/min:

Q_E = Area of piston × Velocity

$$= \frac{\pi}{4} \times (200/1000)^2 \times \frac{5}{60}$$

$= 0.00262\ \text{m}^3/\text{min}$

$= 0.00262 \times 60 \times 1000$

$= 157$ L/min

(b) Flow of oil leaving cylinder Q_E is given by

$Q_E =$ Annulus area × Velocity

$$= \frac{\pi}{4} \times [(200/1000)^2 - (140/1000)^2] \times \frac{5}{60}$$

$= 80$ L/min

(c) The same fluid flow rate used to extend the cylinder (157 LPM) is used to retract the cylinder. The retract cylinder velocity v is given by

$$v = \frac{Q_E}{A - a}$$

Now

Q_E= 157 L/min = 0.00262 m^2

$(A - a)$ = Annulus area= 0.01602 m^2

$$v = \frac{0.00262}{0.01602} = 0.614 \text{ m/s} = 9.8 \text{ m/min}$$

(d) Flow from the full bore end of cylinder Q_R is given by
$Q = A \times v = 0.03142 \times 0.164 = 0.00515 \text{ m}^3/\text{s} = 309$ LPM

(e) We have

$$\text{Full bore area} = \frac{\pi \times 0.2^2}{4} = 0.0314 \text{ m}^2$$

Dynamic extend thrust = 0.9 × Pressure × Full bore area

$$= 0.9 \times 100 \times 10^5 \times 0.0314 \left(\frac{\text{N}}{\text{m}^2} \times \text{m}^2\right)$$

$$= 283 \text{ kN}$$

(f) Annulus area $= \dfrac{\pi \times (0.2^2 - 0.14^2)}{4}$

$$= 0.016 \text{ m}^2$$

$$\text{Dynamic retract thrust} = 0.9 \times 100 \times 10^5 \times 0.016 \left(\frac{\text{N}}{\text{m}^2} \times \text{m}^2\right)$$

$$= 144 \text{ kN}$$

(g) Piston rod area $= \dfrac{\pi \times 0.14^2}{4} = 0.0154 \text{ m}^2$

$$\text{Extend speed} = \frac{\text{Flow rate}}{\text{Piston rod area}}$$

$$= \frac{157 \times 10^{-3}}{0.0154} \left(\frac{\text{L}}{\text{min}} \times \frac{\text{m}^3}{\text{L} \times \text{m}^2}\right)$$

$$= 10.2 \text{ m/min}$$

This compares with 5 m/min when connected conventionally.

(h) For a regenerative system

$$\text{Dynamic extend thrust} = 0.9 \times 100 \times 10^5 \times 0.0154 \left(\frac{\text{N}}{\text{m}^2} \times \text{m}^2\right)$$

$$= 138.6 \text{ kN}$$

As the area of the annulus is almost equal to that of the rod, the regenerative extend and conventional retract thrusts and speeds are almost the same.

Example 1.15

A mass of 2000 kg is to be accelerated horizontally up to a velocity of 1 m/s from the rest over a distance of 50 mm (Fig. 1.20). The coefficient of friction between the load and guide is 0.15. Calculate the bore of the cylinder required to accelerate this load if the maximum allowable pressure at the full bore end is 100 bar (take seal friction to be equivalent to a pressure drop of 5 bar). Assume that the back pressure at the annulus end of the cylinder is zero.

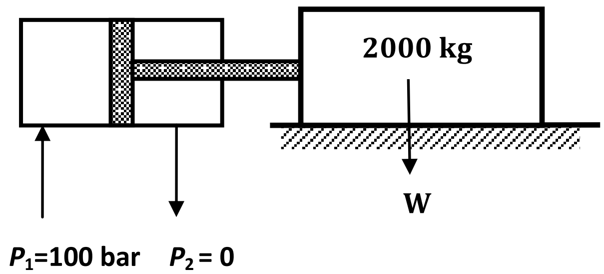


Figure 1.20

Solution: In this case, u = 0, v = 1 m/s, s = 0.05 m and a is unknown. Using the equation

$$v^2 = u^2 + 2as$$

We have,

$$1^2 = 0^2 + 2a \times 0.05 \Rightarrow a = 10\ \mathrm{m/s^2}$$

The force to accelerate the load is given by

$$F = \left(\frac{w}{g}\right)a = \left(\frac{2000 \times 9.81}{9.81}\right)10 = 20000\ \mathrm{N}$$

The force P to overcome the load friction is given by

$$P = \mu W = 0.15 \times 2000 \times 9.81 = 2943\ \mathrm{N}$$

The total force to accelerate the load and overcome friction is

$$F + P = 20000 + 2943 = 22943\ \mathrm{N}$$

The cylinder area required for a given thrust is calculated from

Thrust = Force × Area

The pressure available is pressure at the full bore end of the cylinder minus the equivalent seal break-out pressure.

$$\text{Pressure available} = 100-5 = 95\ \text{bar} = 95 \times 10^5\ \text{bar}$$

Area is given by

$$\text{Area} = \frac{22943}{195 \times 10^5}$$
$$= 0.002415\ \mathrm{m^2} = 2415\ \mathrm{mm^2}$$

Now area is also given by

$$\text{Area} = \frac{\pi D^2}{4}$$

where D is the diameter. Comparing the two equations , we get D = 55.4 mm.
The cylinder diameter is thus 55.4 mm. This neglects the effect of any back pressure. The nearest standard cylinder above has a 63 mm diameter bore.

Lecture 14

HYDRAULIC ACTUATORS [CONTINUED]

1. 7 First-, Second- and Third-Class Lever Systems

Many mechanisms use hydraulic cylinders to transmit motion and power. Among these, lever mechanisms such as toggles, the rotary devices and the push--pull devices use a hydraulic cylinder. In this section, the mechanics of cylinder loading used in first-class, second-class and third-class lever systems is being discussed.

1. 7.1 First-Class Lever System

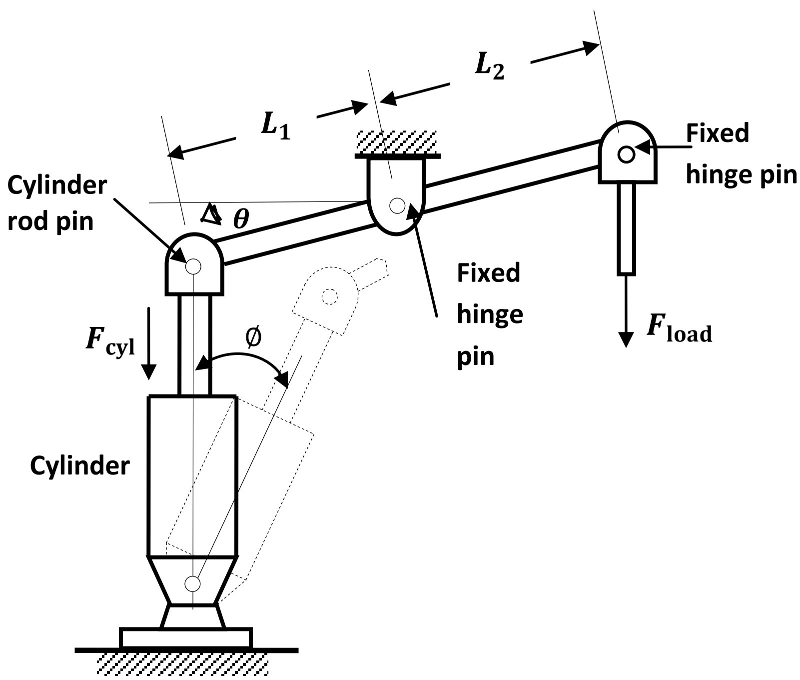


Figure 1. 21 First-class lever system

In this lever system, the fixed-hinge point is located in between the cylinder and the loading point. The schematic arrangement of a first-class lever system with a hydraulic cylinder is shown in Fig.1. 21. In this system, the downward load acts at the lever end. The cylinder has to apply a downward force to lift the load. The cylinder has a clevis mounting arrangement; it pivots about its eye-end center through an angle. However, the effect of this angle (around 10° to 15°) is negligible on the force and hence cannot be considered.

Here, F_{load} = load to be operated, F_{cyl} = load to be exerted by a hydraulic cylinder, L_1 = distance from the rod end to the pivot point, L_2 = distance from the pivot point to the loading point and θ = inclination of the lever measured with respect to the horizontal line at the hinge.

When the load is being lifted, the cylinder force rotates the lever in an anticlockwise direction about the pivot point. Due to this, a moment acts in the anticlockwise direction. At the same time, the force due to the load acting causes a clockwise moment. At equilibrium, the two moments are equal

$$F_{cyl} \times (L_1 \cos\theta) = F_{load} \times L_2 \cos\theta$$

$$F_{cyl} = \frac{L_2}{L_1} F_{load} \qquad (1.\ 3)$$

Suppose the centerline of the hydraulic cylinder tilts by an offset angle ϕ from the vertical; the relationship becomes

$$F_{\text{cyl}} \cos\phi \times (L_1)(\cos\theta) = F_{\text{load}} \times L_2 \cos\theta$$

$$F_{\text{cyl}} = \frac{L_2}{(L_1)\cos\phi} F_{\text{load}} \qquad (1.\ 4)$$

Note 1: If$L_1 > L_2$, the cylinder force is less than the load force and the cylinder stroke is greater than the load stroke.
Note 2: Ifthe inclination of cylinder (ϕ) is less than 10^{o},its effect can be ignored in equation (2).

1. 7.2 Second-Class Lever System

In this lever system, the loading point is in between the cylinder and the hinge point as shown in Fig.1. 22.

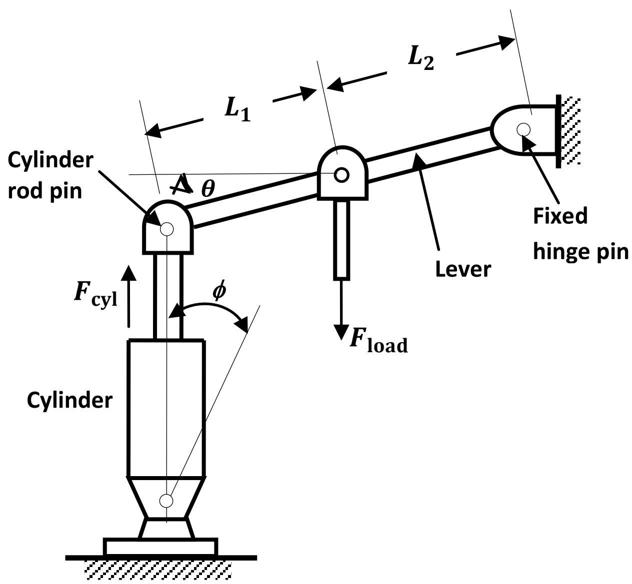


Figure 1. 22 Second-class lever system

Using the same nomenclature discussed under the previous lever systems and equating moments about the fixed-hinge pin, we can write

$$F_{\text{cyl}} \cos\phi \times (L_1 + L_2)(\cos\theta) = F_{\text{load}} \times L_2 \cos\theta$$

$$F_{\text{cyl}} = \frac{L_2}{(L_1 + L_2)\cos\phi} F_{\text{load}} \quad (1.\ 5)$$

Note 3: Compared to the first-class lever, the second-class lever requires smaller cylinder force to drive the given load force for same L_1 and L_2 and load force. In other words, if we use a second-class lever cylinder, a smaller size cylinder can be used.
Note 4: Compared to the first-class lever, the second-class lever also results ina smaller load stroke for a given cylinder stroke.

1. 7.3 Third-Class Lever System

For a third-class lever system shown in Fig. 1.23, the cylinder rod pin lies between the load road pin and the fixed-hinge pin of the lever.

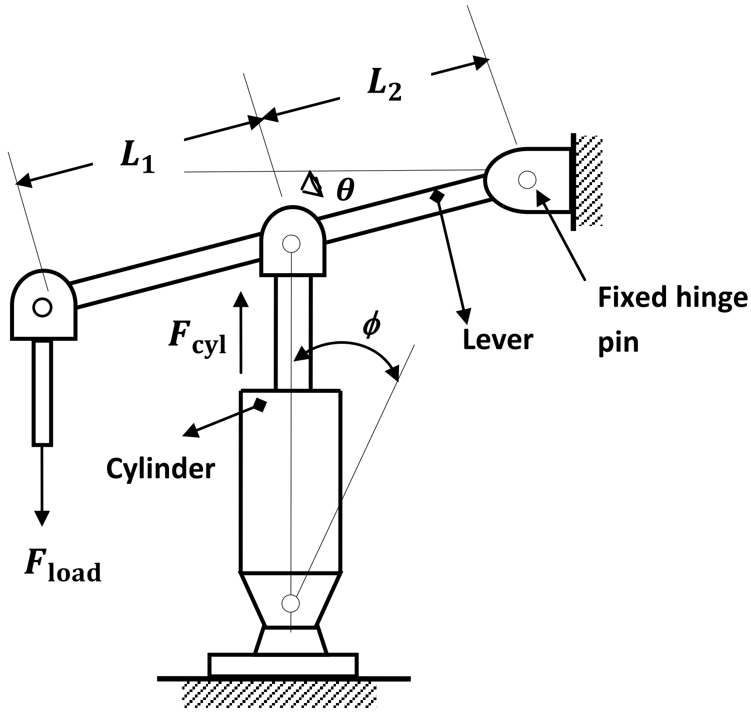


Figure 1. 23 Third-class lever system

Equating moments about the hinge point, we can write

$$F_{\text{cyl}} \cos\phi \times (L_2 \cos\theta) = F_{\text{load}} \times (L_1 + L_2) \cos\theta$$

$$F_{\text{cyl}} = \frac{L_1 + L_2}{L_2 \cos\phi} F_{\text{load}} \qquad (1.\ 6)$$

Note 4: In a third-class lever system**,** cylinder force is greater than load force.
Note 5: In a third-class lever system, load stroke is greater than the cylinder stroke and therefore requires a larger cylinder.

Example 1. 16

Following data are given for the first-, second- and third-class lever systems: $L_1 = L_2 = 25.4$ cm, $\phi = 10°$, $F_{\text{load}} = 4444$ N. Compare the cylinder force needed in each case to overcome the load force. Repeat this with $\phi = 5°$ and $10°$.

Solution: First-class lever system

$$F_{\text{cyl}} = \left(\frac{L_2}{L_1}\right) \frac{F_{\text{load}}}{\cos\phi}$$

$$= \frac{25}{25\cos 0°} 4444 \text{ N}$$

$$= 4444 \text{ N}$$

Second-class lever system

$$F_{cyl} = \frac{L_2}{(L_1 + L_2)\cos\phi} F_{load}$$
$$= \frac{2}{(25+25)\cos 10^\circ} 4444$$
$$= 2222\,\text{N}$$

Third-class lever system

$$F_{cyl} = \frac{L_1 + L_2}{L_2 \cos\phi} F_{load}$$
$$= \frac{25+25}{25\cos 10^\circ} 4444$$
$$= 8888\,\text{N}$$

As seen, the cylinder force in the second-class lever system is half of that in the first-class lever system and one-fourth of that in the third-class lever system.

When $\phi = 5^\circ$ and 10°:

First-class lever

$$F_{cyl}\left(\phi = 5^\circ\right) = \frac{4444}{\cos 5^\circ} = 4461\ \text{N}$$
$$F_{cyl}\left(\phi = 10^\circ\right) = \frac{4444}{\cos 10^\circ} = 4729\ \text{N}$$

Second-class lever

$$F_{cyl}\left(\phi = 5^\circ\right) = \frac{2222}{\cos 5^\circ} = 2231\ \text{N}$$
$$F_{cyl}\left(\phi = 10^\circ\right) = \frac{2222}{\cos 10^\circ} = 2365\ \text{N}$$

Third-class lever

$$F_{cyl}(\phi = 5^\circ) = \frac{8888}{\cos 5^\circ} = 8922\,\text{N}$$
$$F_{cyl}\left(\phi = 10^\circ\right) = \frac{8888}{\cos 10^\circ} = 9458\ \text{N}$$

Example 1. 17

For the system given in Fig. 1. 24 determine the force required to drive a 1000 N load.

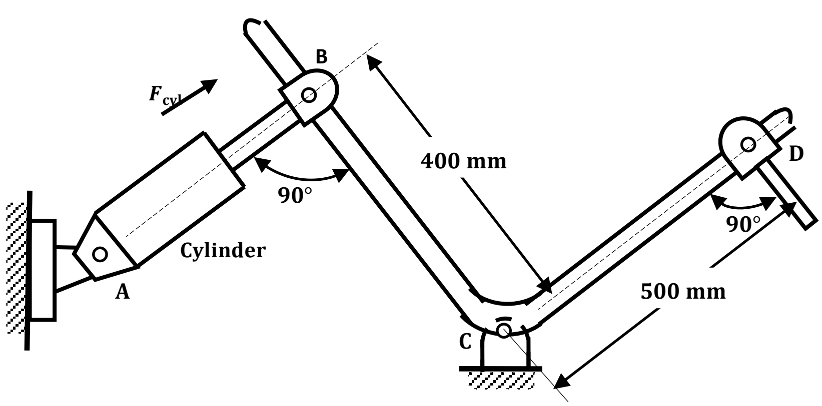


Figure 1. 24

Solution: As seen, it is a first-class lever system. Taking moments about the hinge, we get

$$F_{\text{cyl}} \times \text{BC} = F_{\text{load}} \times \text{CD}$$

$$\Rightarrow F_{\text{cyl}} = F_{\text{load}} \times \frac{\text{CD}}{\text{BC}} = 1000 \times \frac{500}{400} = 1250 \text{ N}$$

Example 1. 18

For the crane system, determine the hydraulic cylinder force required to lift a 2000 N load (Figs. 1. 25 and 1. 26).

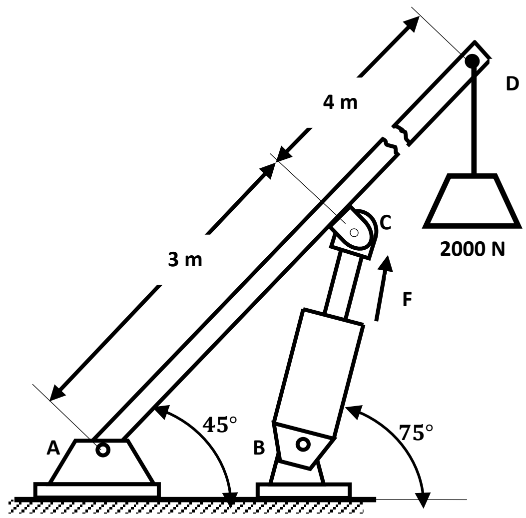


Figure 1. 25

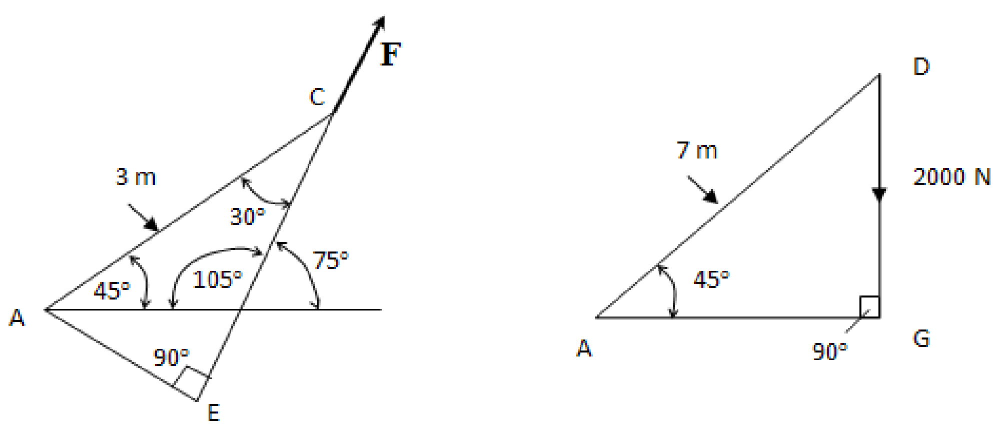


Figure 1. 26

Solution: The given system is a third-class lever system as the cylinder pin lies in between the load rod pin and fixed-hinge pin of the lever. Equating moments about fixed pin A due to the cylinder force *F* and the 2000 N force we get

$$2000 \times \text{Perpendicular dist. AG} = F \times \text{Perpendicular dist. AE}$$

From trigonometry of right-angled triangles, we have

$$\cos 45° = \frac{\text{AG}}{7} \Rightarrow \text{AG} = 7\cos 45° = 4.95 \,\text{m}$$

$$\sin 30° = \frac{\text{AE}}{3} \Rightarrow \text{AE} = 3\sin 30° = 1.5\,\text{m}$$

$$2000 \times 4.95 = F \times 1.5$$

$$\Rightarrow F = 6600\ \text{N}$$

Example 1. 19

Figure 1. 27 shows a toggle mechanism. Find the output load force for a hydraulic cylinder force of 1000 N.

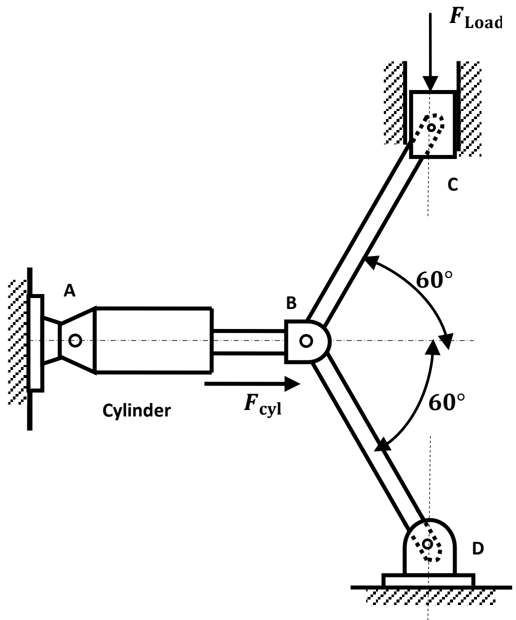


Figure 1. 27

Solution: Setting the sum of the forces on pin C equal to zero (from Newton's law of motion), force (F) = ma = 0 because a = 0 for constant velocity motion, yields the following for the x- and y-axes:

y-axis: $F_{\text{BC}} \sin 60° - F_{\text{BD}} \sin 60° = 0 \Rightarrow F_{\text{BC}} = F_{\text{BD}}$

x-axis: $F_{\text{cyl}} - F_{\text{BC}} \cos 60° - F_{\text{BD}} \cos 60° = 0$

$$F_{\text{cyl}} - 2F_{\text{BC}} \cos 60° = 0$$

$$\Rightarrow F_{\text{BC}} = \frac{F_{\text{cyl}}}{2\cos 60°}$$

Similarly, setting the sum of forces on pin C equal to zero for the y-axis direction yields

$$F_{\text{BC}} \sin 60° - F_{\text{load}} = 0$$

Therefore, we have

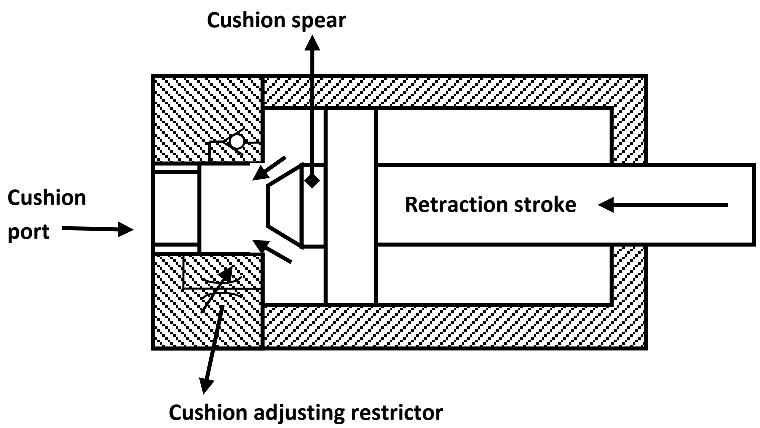
$$F_{\text{load}} = F_{\text{BC}} \sin 60°$$

$$= \frac{\sin 60°}{2\cos 60°} \times F_{\text{cyl}}$$

$$= \frac{\tan 60°}{2} \times 1000 = 866\,\text{N}$$

1. 8 Cylinder Cushions

For the prevention of shock due to stopping loads at the end of the piston stroke, cushion devices are used. Cushions may be applied at either end or both ends. They operate on the principle that as the cylinder piston approaches the end of stroke, an exhaust fluid is forced to go through an adjustable needle valve that is set to control the escaping fluid at a given rate. This allows the deceleration characteristics to be adjusted for different loads. When the cylinder piston is actuated, the fluid enters the cylinder port and flows through the little check valve so that the entire piston area can be utilized to produce force and motion. A typical cushioning arrangement is shown in Fig. 1. 28.



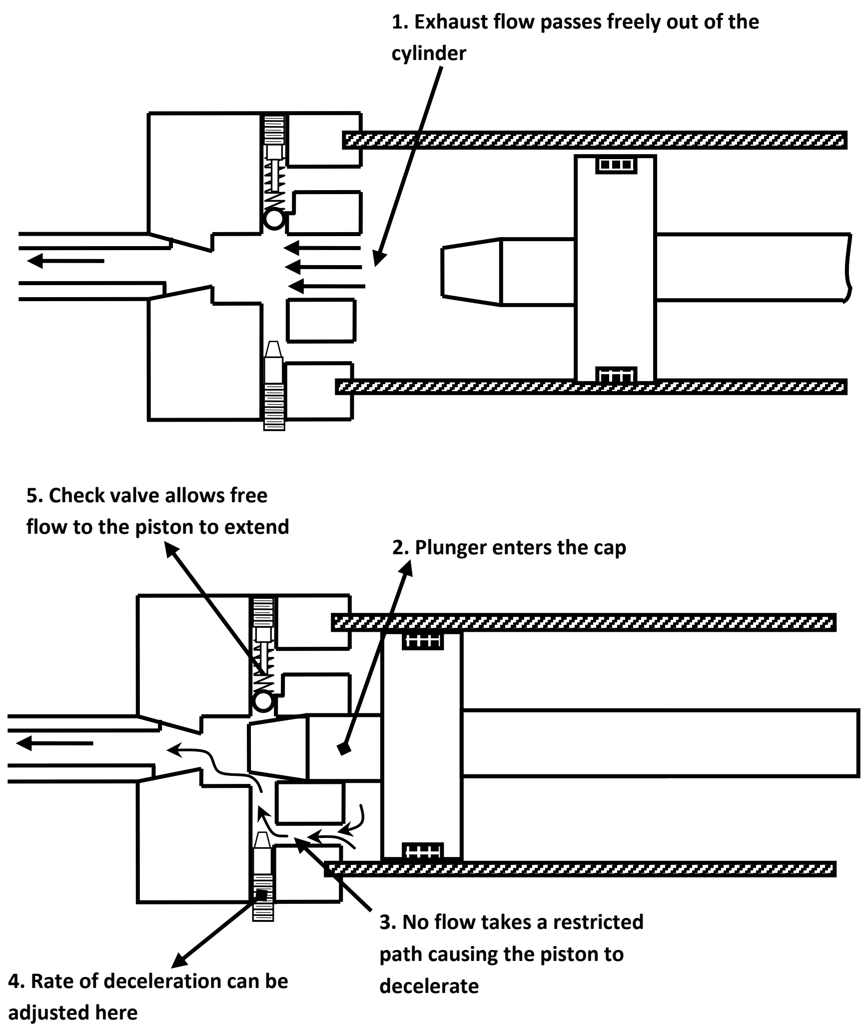


Figure 1. 28 Operation of cylinder cushions

1. 8.1 Cushioning Pressure

During deceleration, extremely high pressure may develop within a cylinder cushion. The action of the cushioning device is to set up a back-pressure to decelerate the load.

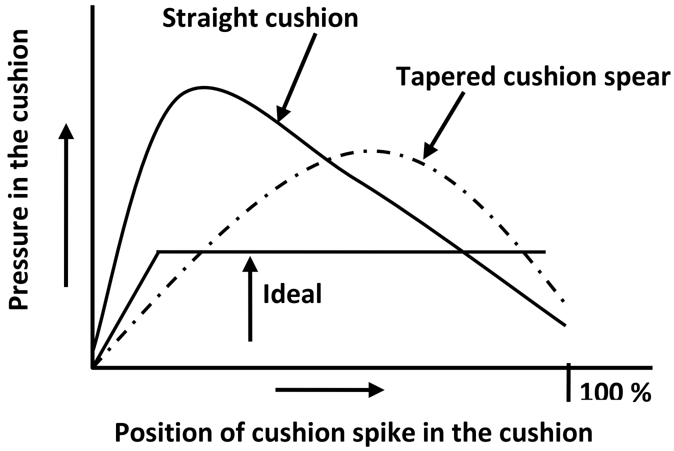


Figure 1. 29 Pressure distribution in cushioning

Ideally, the back-pressure is constant over the entire cushioning length to give a progressive load deceleration. In practice, cushion pressure is the highest at the moment when the piston rod enters the cushion(Fig. 1. 29).Some manufacturers have improved the performance of their cushioning devices by using a tapered or a stepped cushion spear. Wherever high inertia loads are encountered, the cylinder internal cushions may be inadequate but it is possible for the load to be retarded by switching in external flow controls. Deceleration can then take place over a greater part of the actuator stroke.

1. 8.2 Maximum Speeds in Cushioned Cylinders

The maximum speed of a piston rod is limited by the rate of fluid flow into and out of the cylinder and the ability of the cylinder to withstand the impact forces that occur when the piston movement is arrested by the cylinder end plate.

In an uncushioned cylinder, it is normal to limit the maximum piston velocity to 8m/min. This value is increased to 12 m/min for a cushioned cylinder, and 30 m/min is permissible with high-speed or externally cushioned cylinders. Oversize ports are necessary in cylinders used in high-speed applications. In all cases, the maximum speed depends upon the size and type of load. It is prudent to consult the manufacturer if speeds above 12 m/min are contemplated.

When only a part of the cylinder stroke is utilized, cushions cannot be used to decelerate the load. In such cases, it may be necessary to introduce some form of external cushioning especially where high loads or precise positioning is involved.

Example 1. 20
A cylinder has a bore of 125 mm diameter and a rod of 70 mm diameter. It drives a load of 2000 kg vertically up and down at a maximum velocity of 3 m/s. The lift speed is set by adjusting the pump displacement and the retract speed by a flow control valve. The load is slowed down to rest in the cushion length of 50 mm. If the relief valve is set at 140 bar, determine the average pressure in the cushions on extend and retract. (Neglect pressure drops in pipe work and valves.)

Solution: Kinetic energy of load

$$\text{Kinetic energy} = (1/2)\ \text{Mass} \times \text{Velocity}^2$$

$$= (1/2)\,(2000) \times 3^2 = 9000 \text{ N m}$$

Average force to retard load over 50 mm is

$$\frac{\text{Kinetic energy}}{\text{Distance}} = \frac{9000 \times 10^3}{50} = 180 \text{ kN}$$

The force acting on the load is

$$\text{Load} = 2000 \text{ kg} = 2000 \times 9.81 = 19.6 \text{ kN}$$

Annulus area = $\frac{\pi}{4}(0.125^2 - 0.07^2) = 0.0084 \text{ m}^2$

Full bore area = $\left(\frac{\pi}{4}\right) \times (0.125^2) = 0.0123 \text{ m}^2$

The kinetic energy of the load is opposed by the cushion force and the action of gravity on the load.
Cushion pressure to absorb the kinetic energy of load when extending is

$$\frac{(180 \times 10^3) - (19.6 \times 10^3)}{(8.4 \times 10^{-3})} \text{ (N/m}^2\text{)} = 19.1 \times 10^6 = 191 \text{ bar}$$

When the piston enters the cushion, the pressure on the full bore side of the piston rises to relief valve pressure. This pressure on the full bore side drives the piston into the cushions, and so increases the cushion pressure needed to retard the load. The cushion pressure to overcome the hydraulic pressure on the full bore end is

$$\text{Pressure} \times \frac{\text{Full bore area}}{\text{Annulus area}} = 140 \times \frac{12.3 \times 10^{-3}}{8.4 \times 10^{-3}} = 205 \text{ bar}$$

Thus, the average pressure in the cushion on the extend stroke is (190 + 205) = 395 bar.
During cushioning, the effective annular area is reduced as the cushion sleeve enters the cushion. This has been neglected in the calculation, and in practice, the cushion pressure is even greater.
When the load is retracted, forces act on the load. The back pressure owing to the flow control valve in the circuit is minimal once the piston enters the cushion and is neglected in this calculation.
The force in the cushion has to overcome the kinetic energy of the load, the weight of the load and the force due to the hydraulic pressure. The force owing to the hydraulic pressure is

$$\text{Force} = \text{Pressure} \times \text{Annulus area}$$
$$= (140 \times 10^5) \times (8.4 \times 10^{-3}) \text{ N}$$
$$= 117.6 \text{ kN}$$

Also

$$\text{Cushion force} = 180 + 19.6 + 117.6 = 317.2 \text{ kN}$$

After knowing the force, we can find cushion pressure =

$$\text{Cushion pressure} = \frac{\text{Force}}{\text{Area}} = \frac{317.2}{0.0123} \text{ (kN/m}^2\text{)} = 25800 = 258 \text{ bar}$$

The average pressure in the cushion retracting is 258 bar. Again this value is somewhat higher as the cushion spike reduces the effective cushion area below that used.

Example 1. 21

A pump delivers oil at a rate of 1.15 LPS into the blank end of the 76.2 mm diameter hydraulic cylinder shown in Fig. 1. 30. The pistons decelerate over a distance of 19.05 mm at the end of its extension stroke. The cylinder drives a 6672 N weight which slides on a flat horizontal surface having a coefficient of friction (CF) equal to 0.12. The pressure relief valve setting equals 51.7125 bar. Therefore, the maximum pressure (p_1) at the blank end of the cylinder equals 51.7125 bar while the cushion decelerates the piston. Find the maximum pressure (p_2) developed by the cushion.

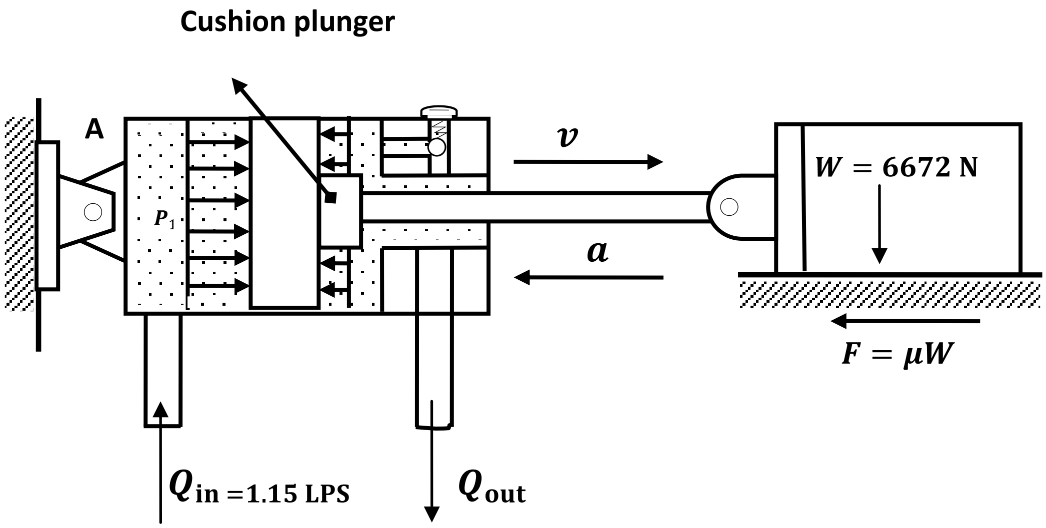


Figure 1. 30

Solution:

Step 1: Calculate the steady piston velocity v prior to deceleration:

$$v = \frac{Q_{\text{pump}}}{A_{\text{piston}}} = \frac{\frac{1.15}{1000}}{\frac{\frac{\pi}{4}(0.0762^2)}{1}} = \frac{1.15}{4.5} = 0.255 \text{ m / s}$$

Step 2: Calculate the deceleration a of the piston during the 19.05 mm displacement S using the constant acceleration or deceleration equation:

$$v^2 = 2as$$

Substituting the values and solving for deceleration we get

$$a = \frac{v^2}{2s} = \frac{0.255^2}{2(19.05 \times 10^{-3})} = 17.06 \text{ m/s}^2$$

Step 3: Using Newton's laws of motion, the net force acting on the system is equated as

$$\sum F = ma$$

Consider the forces that tend to slow down the system as positive forces as we are solving for deceleration. The mass under consideration m is equal to the sum of all the masses of moving members (piston, road and load). Because the weight of the piston and road is small compared to the weight of the load, the weight of the piston and rod is ignored. The frictional forces acting between the weight W and the horizontal support surface equal *coefficient of friction* (CF) times W. This frictional force is the external force acting on the cylinder while it moves the weight.

Substituting into Newton's equations yields

$$p_2(A_{\text{piston}} - A_{\text{cushion}}) + \text{CF} \times W - p_1(A_{\text{piston}}) = \frac{W}{g}$$

$$p_2 = \frac{\left[(6672)\left(\frac{17.06}{9.81}\right)\right] + 51.7125\left(\frac{\pi}{4}\right)(76.2\times10^{-3})^2 - (0.12)(6672)}{\left[\frac{\pi}{4}(76.2\times10^{-3})^2 - \frac{\pi}{4}\left(25.4\times10^{-3}\right)^2\right]} = 59.0212\,\text{bar}$$

Thus, the hydraulic cylinder must be designed to withstand an operating pressure of $59.0212\,\text{bar}$ rather than the pressure relief setting of $51.7125\,\text{bar}$.

1. 9 Cylinder Mountings and Strength Calculations

The types of mounting on cylinders are numerous, and they can accommodate a wide variety of applications. One of the important considerations in selecting a particular mounting is whether the force applied is tensile or compressive. As far as possible, bucking load must be avoided. The ratio of rod length to diameter should not exceed 6:1 to prevent bucking. Alignment of the rod with the resistive load is another important consideration while selecting cylinder mounts. The various kinds of mountings normally used in industries are as follows (for various mounting, refer Fig. 1. 31):

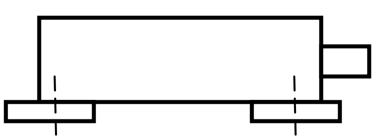
1. **Foot mounting:** It should be designed to give a limited amount of movement on one foot only to allow for thermal or load expansion. That is, the cylinder should be positively located or dowelled at one end only.

2. **Rod-end flange or front flange mounting:** During the extend stroke, pressure in the hydraulic fluid acts on the cylinder-end cap, the force set up being transmitted to the front mounting flange through the cylinder body.

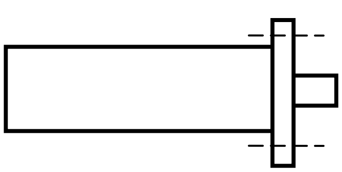
3. **Rear flange, back flange or head-end flange mounting:** No stress is present in the cylinder owing to load on the extend stroke; only hoop stress is present. The load acts through the fluid onto the rear flange.

4. **Trunnion mounting:** It allows angular movement. It is designed to take shear load only. Bearing should be as close to the cylinder body as possible.

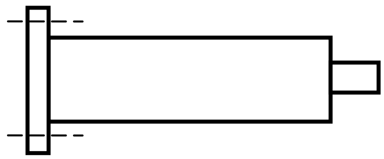
5. Eye or clevis mounting: There is a tendency for the cylinder to jack knife under load. Side loading of bearing must be carefully considered.



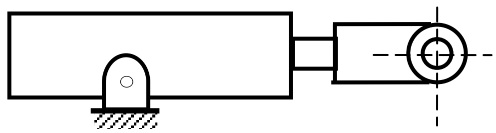
(a)



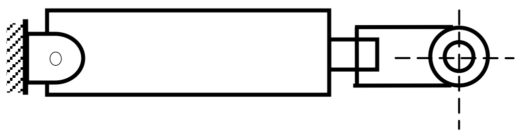
(b)



(c)



(d)



(e)

Figure 1. 31 (a) Foot mounting; (b) rod-end flange or front flange mounting; (c) rear flange, back flange or head-end flange mounting; (d) trunnion mounting; (e) eye or clevis mounting.

1. 9.1 Piston Rod Ends

The piston rod ends can be supplied with a male or female thread according to the manufacturer's specification. Rod-end eyes with spherical bearings are available from some suppliers.

1. 9.2 Protective Covers

These are fitted to protect the piston rod when the piston works in an abrasive environment or when the cylinder is not used for long periods and a heavy deposit of dust accumulates on the rod. The protective covers are of the form of telescope or bellows and completely enclose the rod at all times in the cylinder movement.

Bellows may either be molded or fabricated. Molded bellows are manufactured from rubber or plastic and owing to their construction; they are limited to a contraction ratio of about 4:1. An extended piston rod is required to accommodate the closed length of the bellows. This increases the overall cylinder length and tends to restrict their use to relatively short stroke cylinders.

Fabric covers made of plastic, leather, impregnated cloth or canvas can have a contraction ratio greater than 15:1. When a fabricated cover is used on a horizontal cylinder, it must be supported externally to prevent the cylinder rod from rubbing the cover.

Telescopic covers are made of a rigid material, normally metal, and are used under conditions where fabric covers are inadequate.

1. 9.3 Piston Rod Buckling

A piston rod in a hydraulic cylinder acts as a strut when it is subjected to a compressive load or it exerts a thrust. Therefore, the rod must be of sufficient diameter to prevent buckling. Euler's strut theory is used to calculate a suitable piston rod diameter to withstand buckling. Euler's formula states that

$$F_{\mathrm{b}} = \frac{\pi^2 EI}{L^2}$$

where F_{b} is the buckling load (kg), E is the modulus of elasticity (kg/cm^2; 2.1×10^6 kg/cm^{2}for steel), I is the second moment of inertia of the piston rod (cm^4; $\pi d^2 / 64$ for a solid rod of diameter d cm) and L is the free (equivalent) buckling length (cm) depending on the method of fixing the cylinder and piston rod and is shown in Fig.1. 32. The maximum safe working thrust or load F on the piston rod is given by

$$F = \frac{F_{\mathrm{b}}}{S}$$

where S is the factor of safety that is usually taken as 3.5. The free or equivalent buckling length L depends on the method of fixing the piston rod end and the cylinder, and on the maximum distance between the fixing points, that is, the cylinder fully extended. In cases where the cylinder is rigidly fixed or pivoted at both ends, there is a possibility of occurrence of excessive side loading. The effect of side loading can be reduced by using a stop tube inside the cylinder body to increase the minimum distance between the nose and the piston bearings. Refer Fig. 1. 33 for use of a stop tube to minimize side loading. The longer the stop tube, the lower the reaction force on the piston owing to the given value of the side load. Obviously, the stop tube reduces the effective cylinder stroke.

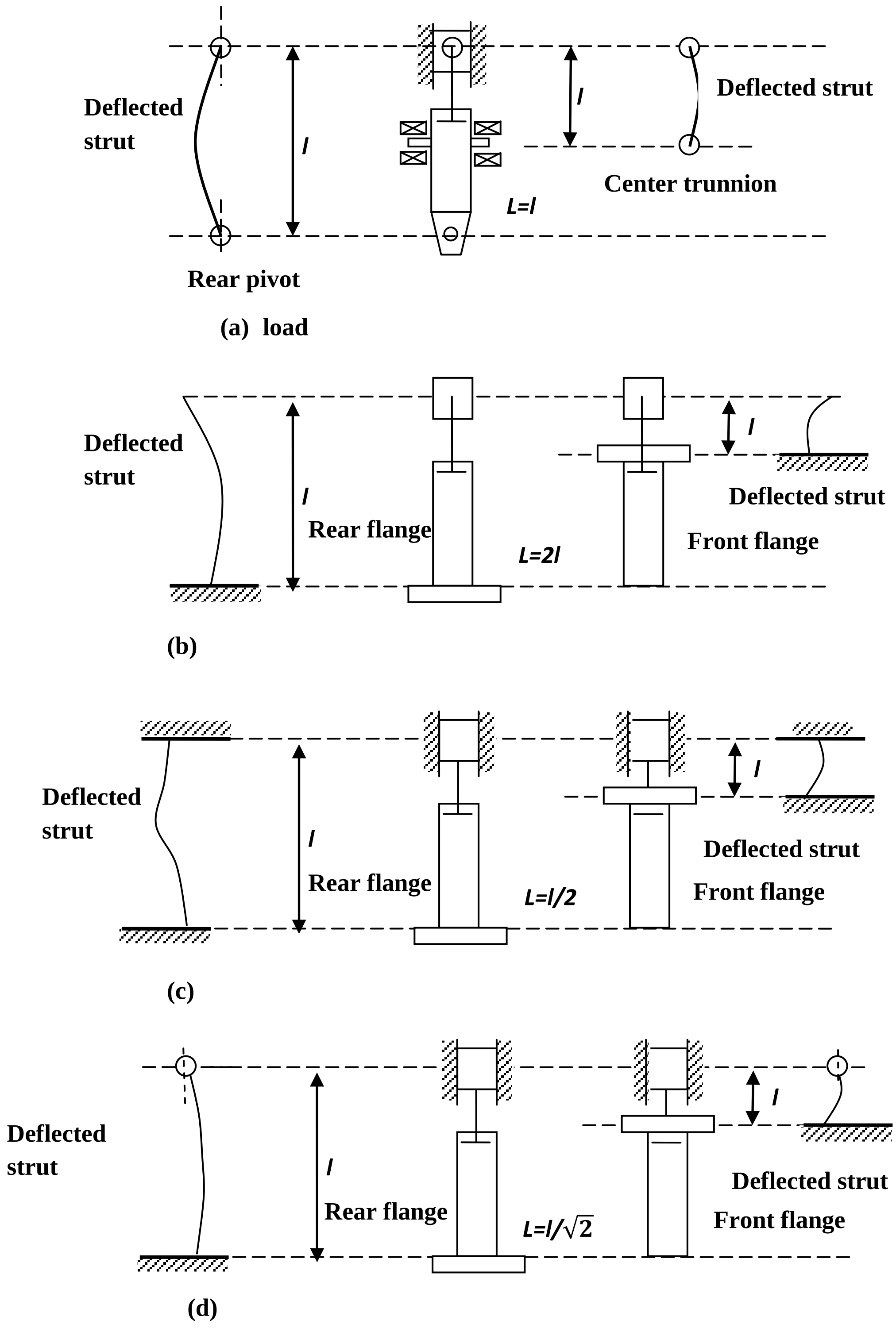


Figure 1. 32 Relationship between the piston rod, free buckling length and method of fixing. (a) Rear pivot and center trunnion mounted, guided pivoted. (b) One end rigidly fixed, free load.(c) One end rigidly fixed, guided load. (d) One end rigidly fixed, pivoted and guided load

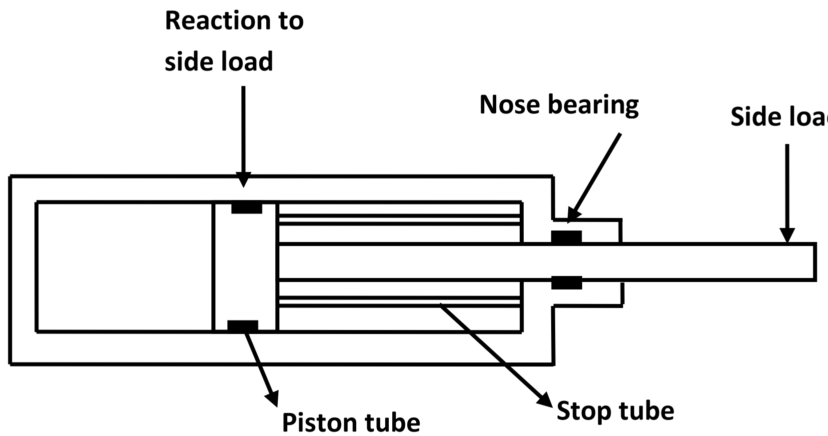


Figure 1. 33 Use of a stop tube to minimize side loading

Objective-Type Questions
Fill in the Blanks

1. An actuator is used to convert the _______ back into the _______.
2. A telescopic cylinder is used when _______ stroke length and _______ retracted length are required.
3. In a push type, the cylinder _______ to lift a weight against the force of gravity by applying oil pressure at the _______.
4. The drawback of tandem cylinder is that it is _______ than a standard cylinder, achieves an equal speed because flow must go to _______.
5. A major problem in the manufacture of through-rod cylinders is achieving _______ and concentricity of cylinder bore.

State True or False

1. Hydraulic actuators are devices used to convert the pressure energy of the fluid into mechanical energy.
2. A telescopic cylinder is used in applications where a large amount of force is required from a small-diameter cylinder.
3. Semi-rotary actuators are capable of limited angular movements that can be several complete revolutions.
4. Single-acting cylinders can exert a force in both the extending and retracting directions.
5. In pull-type gravity, return-type single-acting cylinder, the cylinder lifts the weight by extending.

Review Questions

1. What is the function of a hydraulic cylinder in a hydraulic system?
2. When is a telescoping cylinder used?
3. Explain the operation of tandem-type cylinder and mention its applications.
4. Explain the function of cushioning in cylinders.
5. Why are wiper rings used on cylinder rods?
6. Mention two applications of single-acting cylinders.

7. How does a welded type of cylinder differ from a tie-rod type? Mention the major parts of a tie-rod cylinder.
8. What are the technical specifications of a hydraulic cylinder?
9. Name the materials that are commonly used to manufacture (a) cylinder covers,(b) piston rods,(c) pistons and (d) tie-rods.
10. What is a hydraulic ram?
11. Mention the different types of mountings used in fixing the hydraulic cylinders.
12. What is the difference between a single-acting and a double-acting hydraulic cylinder?
13. Name four different types of hydraulic cylinder mountings.
14. What is a cylinder cushion? What is its purpose?
15. What is a double-rod cylinder? When would it normally be used?
16. What is a telescoping rod cylinder? When would it normally be used?
17. Differentiate between first-, second- and third-class lever systems used with hydraulic cylinders to drive loads.
18. When using a lever system with hydraulic cylinders, why must the cylinder be clevis mounted?
19. What is the purpose of a hydraulic shock absorber? Name two applications.
20. What is a hydraulic actuator?
21. How is a single-acting cylinder retracted?
22. What are the advantages of a double-acting cylinder over the single-acting cylinder?
23. For which applications, a double-rod cylinder is best suited?
24. What are the advantages and disadvantages of a tandem cylinder?
25. Name the types of cylinder mounting.

Answers
Fill in the Blanks

1.Fluid energy, mechanical power
2.Long, short
3.Extends, blank end
4.Longer, both pistons
5.Correct alignment

State True or False

1. True
2. False
3. True
4. False
5. False

Lecture 15

DIRECTIONAL CONTROL VALVES

Learning Objectives

Upon completion of this chapter, the student should be able to:

- ☐ List different types of valves used in fluid power.
- ☐ Explain various classifications of directional control valves.
- ☐ Describe the working and construction of various direction control valves.
- ☐ Identify the graphic symbols for various types of direction control valves.
- ☐ Explain the different applications of direction control valves.
- ☐ Explain the working principle of solenoid-actuated valves.
- ☐ Define valve overlap.
- ☐ Evaluate the performance of hydraulic systems using direction control valves.

1.1Introduction

One of the most important considerations in any fluid power system is control. If control components are not properly selected, the entire system does not function as required. In fluid power, controlling elements are called valves.

There are three types of valves:

1. **Directional control valves (DCVs):** They determine the path through which a fluid transverses a given circuit. **Pressure control valves:** They protect the system against overpressure, which may occur due to a sudden surge as valves open or close or due to an increase in fluid demand.
2. Flow control valves: Shock absorbers are hydraulic devices designed to smooth out pressure surges and to dampen hydraulic shock.

In addition, the fluid flow rate must be controlled in various lines of a hydraulic circuit. For example, the control of actuator speeds can be accomplished through use of flow control valves. Non-compensated flow control valves are used where precise speed control is not required because the flow rate varies with pressure drop across a flow control valve. It is important to know the primary function and operation of various types of control components not only for good functioning of a system, but also for discovering innovative methods to improve the fluid power system for a given application.

1.2Directional Control Valves

A valve is a device that receives an external signal (mechanical, fluid pilot signal, electrical or electronics) to release, stop or redirect the fluid that flows through it. The function of a DCV is to control the direction of fluid flow in any hydraulic system. A DCV does this by changing the position of internal movable parts. To be more specific, a DCV is mainly required for the following purposes:

- To start, stop, accelerate, decelerate and change the direction of motion of a hydraulic actuator.
- To permit the free flow from the pump to the reservoir at low pressure when the pump's delivery is not needed into the system.
- To vent the relief valve by either electrical or mechanical control.
- To isolate certain branch of a circuit.

Any valve contains ports that are external openings through which a fluid can enter and exit via connecting pipelines. The number of ports on a DCV is identified using the term "way." Thus, a valve with four ports is a four-way valve A DCV consists of a valve body or valve housing and a valve mechanism usually mounted on a sub-plate. The ports of a sub-plate are threaded to hold the tube fittings which connect the valve to the fluid conductor lines. The valve mechanism directs the fluid to selected output ports or stops the fluid from passing through the valve. DCVs can be classified based on fluid path, design characteristics, control methods and construction.

1.2.1 Classification of DCVs based Fluid Path

Based on fluid path, DCVs can be classified as follows:

- Check valves.
- Shuttle valves.
- Two-way valves.
- Three-way valves.
- Four-way valves.

1.2.2 Classification of DCVs based on Design Characteristics

Based on design characteristics, DCVs can be classified as follows:

- An internal valve mechanism that directs the flow of fluid. Such a mechanism can either be a poppet, a ball, a sliding spool, a rotary plug or a rotary disk.
- Number of switching positions (usually 2 or 3).
- Number of connecting ports or ways.
- Method of valve actuation that causes the valve mechanism to move into an alternate position.

1.2.3 Classification of DCVs based on the Control Method

Based on the control method, DCVs can be classified as follows:

- **Direct controlled DCV:**A valve is actuated directly on the valve spool. This is suitable for small-sized valves.
- **Indirect controlled DCV:**A valve is actuated by a pilot line or using a solenoid or by the combination of electrohydraulic and electro-pneumatic means. The use of solenoid reduces the size of the valve. This is suitable for large-sized valves.

1.2.4 Classification of DCVs based on the Construction of Internal Moving Parts

Based on the construction of internal moving parts, DCVs can be classified as follows:

- **Rotary spool type:** In this type, the spool is rotated to change the direction of fluid. It has longitudinal grooves. The rotary spools are usually manually operated.
- **Sliding spool type:** This consists of a specially shaped spool and a means of positioning the spool. The spool is fitted with precision into the body bore through the longitudinal axis of the valve body. The lands of the spool divide this bore into a series of separate chambers. The ports of the valve body lead into these chambers and the position of the spool determines the nature of inter-connection between the ports.

Table 1.1

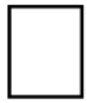
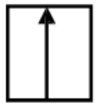
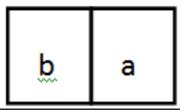
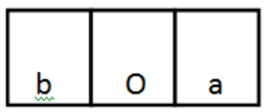
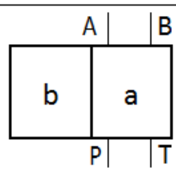
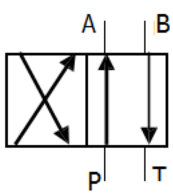
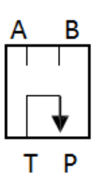
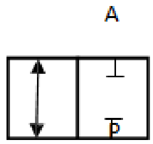
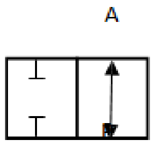
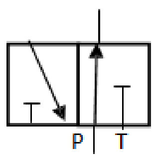
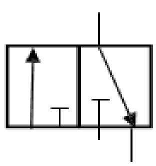
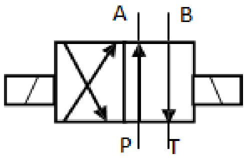
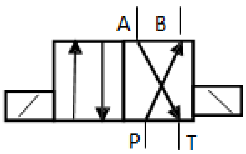
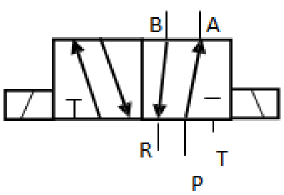
	Each individual switching portion is shown in a square
	Flow path is indicated by means of arrow within a square
	Closed position
	Two-position valve
	Three-position valve
	Ports added to the two-position valve
	Two flow paths
	Two ports are connected, two ports are closed

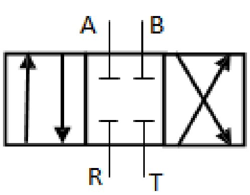
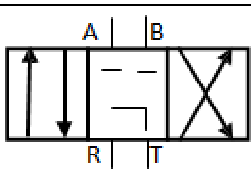
Table 1.2

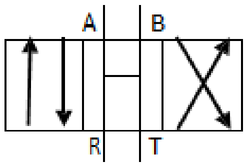
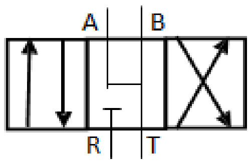
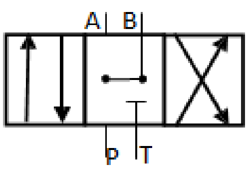
2/2-way valve: 2-ports and 2-position DCV	
	Normally closed position: P is not connected to A. When the valve is not actuated, the way is closed.
	Normally open position: P is connected to A. When the valve is not actuated, the way is open.

3/2 way valve : 3ports and 2 position DCV	
	Normally open position: P is connected to A. When the valve is not actuated, the way is open.
	Normally open position: P is connected to A. When the valve is actuated, the way is closed

4/2-way valve – 4-port and 2-position DCV	
	P is connected to A B is connected to T
	Position 2: P is connected to B A is connected to T

5/2-way valve – 5-port and 2-position DCV	
	Normal position: P is connected to B A is connected to R

4/3-way valve – 4-port and 3-position DCV	
	P, T, A, B
	Mid-position pump reticulating: P to T, A and B closed

	H-Mid-position closed: P to A, B to T
	Mid-position working lines depressurized: P, A to B to T
	P to A to B, T

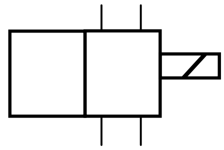
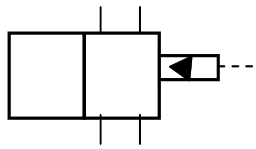
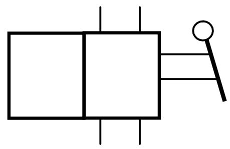
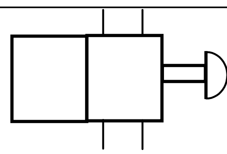
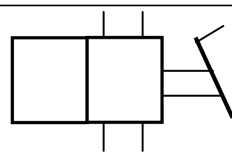
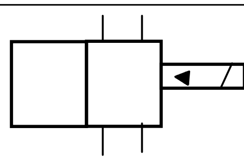
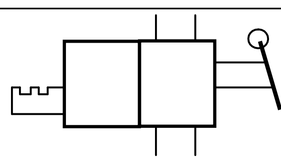
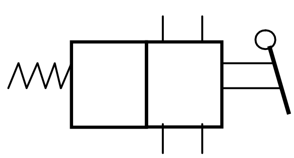
1. Each different switching position is shown by a square.
2. Flow directions are indicated by arrows.
3. Blocked ports are shown by horizontal lines.
4. Ports are shown in an appropriate flow direction with line arrows

The switching position, flow direction, and port for different configurations is represented in Table 1.1. Two-way, three-way, four-way and five-way representation is shown in Table 1.2.

1.3Actuating Devices

Direction control valves may be actuated by a variety of methods. Actuation is the method of moving the valve element from one position to another. There are four basic methods of actuation: Manual, mechanical, solenoid-operated and pilot-operated. Several combinations of actuation are possible using these four basic methods. Graphical symbols of such combinations are given in Table 1.3.

- **Manually operated:** In manually operated DCVs, the spool is shifted manually by moving a handle pushing a button or stepping on a foot pedal. When the handle is not operated, the spool returns to its original position by means of a spring.
- **Mechanically operated:** The spool is shifted by mechanical linkages such as cam and rollers.
- **Solenoid operated:** When an electric coil or a solenoid is energized, it creates a magnetic force that pulls the armature into the coil. This causes the armature to push the spool of the valve.
- **Pilot operated:** A DCV can also be shifted by applying a pilot signal (either hydraulic or pneumatic) against a piston at either end of the valve spool. When pilot pressure is introduced, it pushes the piston to shift the spool.

	Solenoid operated
	Pilot operated
	Manual operated
	Push button
	Foot operated
	Pilot-operated solenoid
	Two-position detent
	Spring return

1.4 Check Valve

The simplest DCV is a check valve. A check valve allows flow in one direction, but blocks the flow in the opposite direction. It is a two-way valve because it contains two ports. Figure 1.1 shows the graphical symbol of a check valve along with its no-flow and free-flow directions.

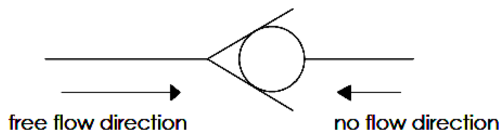


Figure 1.1 Graphical symbol of a check valve.

In Fig. 1.2, a light spring holds the ball against the valve seat. Flow coming into the inlet pushes the ball off the seat against the light force of the spring and continues to the outlet. A very low pressure is required to hold the valve open in this direction. If the flow tries to enter from the opposite direction, the pressure pushes the ball against the seat and the flow cannot pass through.

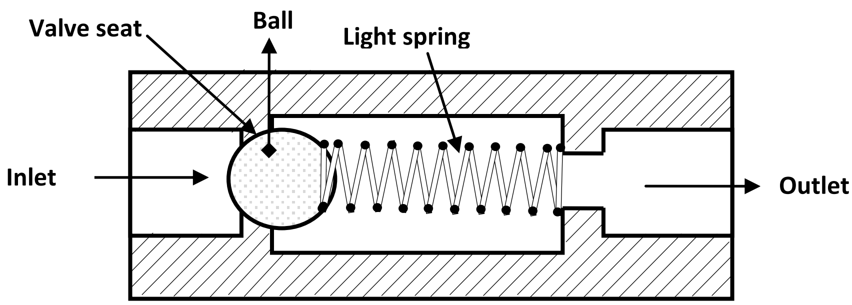


Figure 1.2 Ball-type check valve.

Figure 1.3 provides two schematic drawings showing the operation of a poppet check valve. A poppet is a specially shaped plug element held on a valve seat by a light spring. Fluid flows through the valve in the space between the seat and poppet. In the free flow direction, the fluid pressure overcomes the spring force. If the flow is attempted in the opposite direction, the fluid pressure pushes the poppet in the closed position. Therefore, no flow is permitted

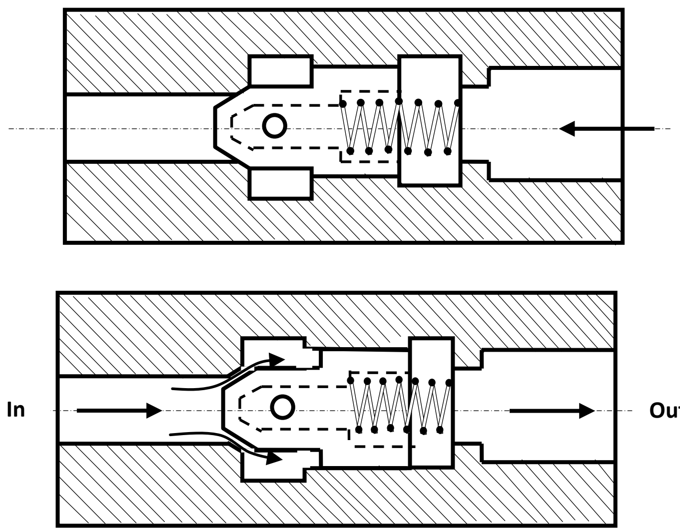


Figure1.3 Poppet check valve: (a) Open and (b) closed position

1.4.1 Advantages of a poppet valve

- Virtually zero leakage in closed position.
- Poppet elements do not stick even when left under pressure for long periods.
- Fast, consistent response time: typically 15 ms.

1.4.2 *Disadvantages of a Poppet Valve*

A poppet valve has the following disadvantages:

- Axial pressure balance is impossible and considerable force may be needed to open the poppet against the flow at a high pressure. This limits valves that have direct mechanical actuation to low flow duties.
- Generally individual poppets are required for each flow path that significantly increases the complexity of multi-port valves.
- Lapping and super finishing of valves add cost.

1.5 Pilot-Operated check Valve

A pilot-operated valve along with its symbol is shown in Fig. 1.4. This type of check valve always permits free flow in one direction but permits flow in the normally blocked opposite direction only if the pilot pressure is applied at the pilot pressure point of the valve. The check valve poppet has the pilot piston attached to the threaded poppet stem by a nut.

The light spring holds the poppet seated in a no-flow condition by pushing against the pilot piston. The purpose of the separate drain port is to prevent oil from creating a pressure build-up at the bottom of the piston. The dashed line in the graphical symbol represents the pilot pressure line connected to the pilot pressure port of the valve. Pilot check valves are used for locking hydraulic cylinders in position.

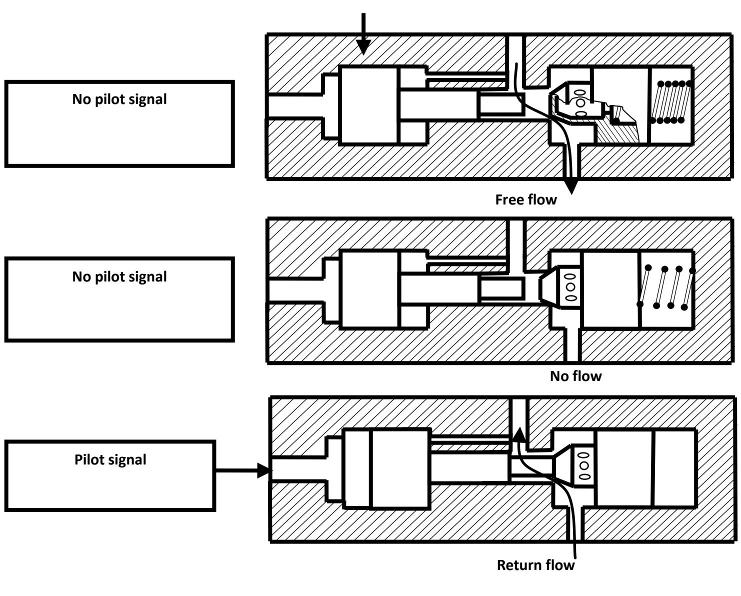


Figure 1.4Pilot-perated check valve

1.6 Shuttle Valve

A shuttle valve allows two alternate flow sources to be connected in a one-branch circuit. The valve has two inlets P_1 and P_2 and one outlet A. Outlet A receives flow from an inlet that is at a higher pressure. Figure 1.5 shows the operation of a shuttle valve. If the pressure at P_1 is greater than that at P_2, the ball slides to the right and allows P_1 to send flow to outlet A. If the pressure at P_2 is greater than that at P_1, the ball slides to the left and P_2 supplies flow to outlet A

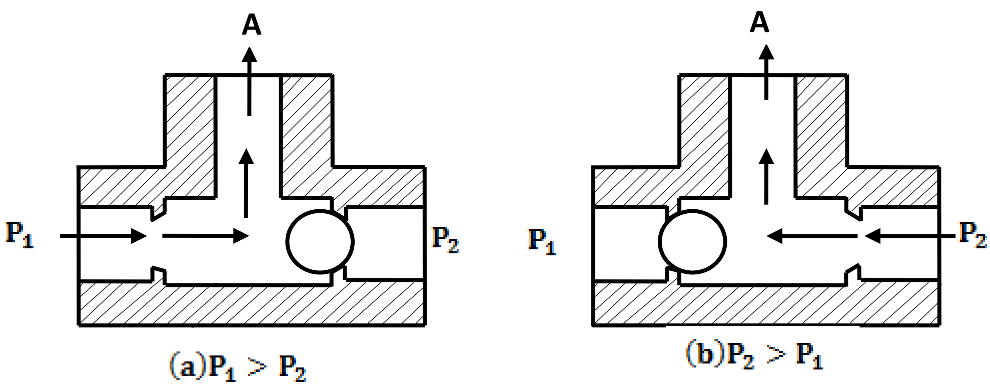


Figure 1.5 Shuttle valve: (a) Flow from left to outlet and (b) flow from right to outlet in Fig. 1.5.

One application for a shuttle valve is to have a primary pump inlet P1 and a secondary pump inlet P2 connected to the system outlet A The secondary pump acts as a backup, supplying flow to the system if the primary pump loses pressure. A shuttle valve is called an "OR" valve because receiving a pressure input signal from either P1 or P2 causes a pressure output signal to be sent to A. Graphical symbol of shuttle valve is shown in Fig. 1.6.

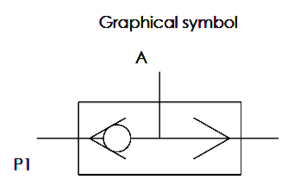
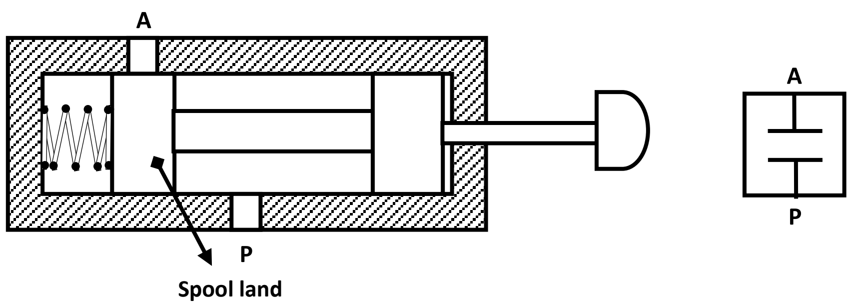


Fig 1.6 symbol of Shuttle valve

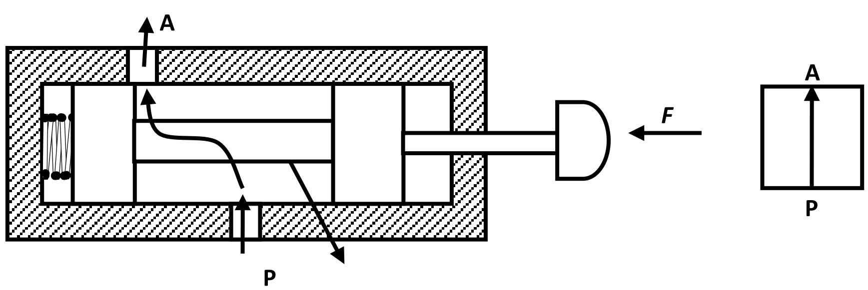
DIRECTIONAL CONTROL VALVE [CONTINUED]

1.7.1 2/2-Way DCV (Normally Closed)

Figure 1.7shows a two-way two-position (normally closed) of spool type. A spool valve consists of a cylindrical spool that slides back and forth inside the valve body to connect or block flow between the ports. The larger diameter portion of the spool, the spool land blocks flow by covering the port. This particular valve has two ports labeled P and A. P is connected to the pump line and A is connected to the outlet to the system. Figure 1.7(a) shows the valve in its normal state and its corresponding symbol. The valve is held in this position by the force of the spring. In this position, the flow from the inlet port P is blocked from going to the outlet port A. Figure 1.7(b) shows the valve in its actuated state and its corresponding symbol. The valve is shifted into this position by applying a force to overcome the resistance of the spring. In this position, the flow is allowed to go to the outlet port.



(a)



(b)

Figure1.7 Two-way–two-position normally closed DCV. (a) Ports A and P are not connected when force is not applied (valve unactuated). (b) Ports A and P are connected when force is applied (valve actuated).

1.7.2 *2/2-Way DCV (Normally Opened)*

Figure 1.8 shows a two-way, two-position normally open DCV. The spring holds the valve in a position in which ports P and Aare connected as shown in Fig1.8.(a). When the valve is actuated, the flow is blocked from going to A as shown in Fig1.8.(b). The complete graphic symbol for the given DCV is shown in Fig.1.8(c).

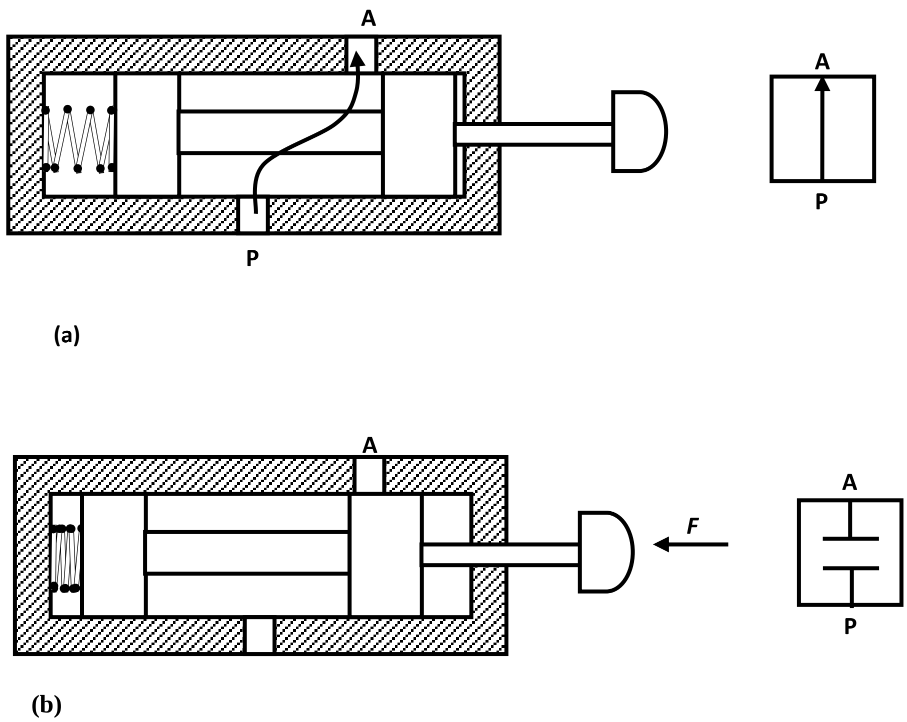
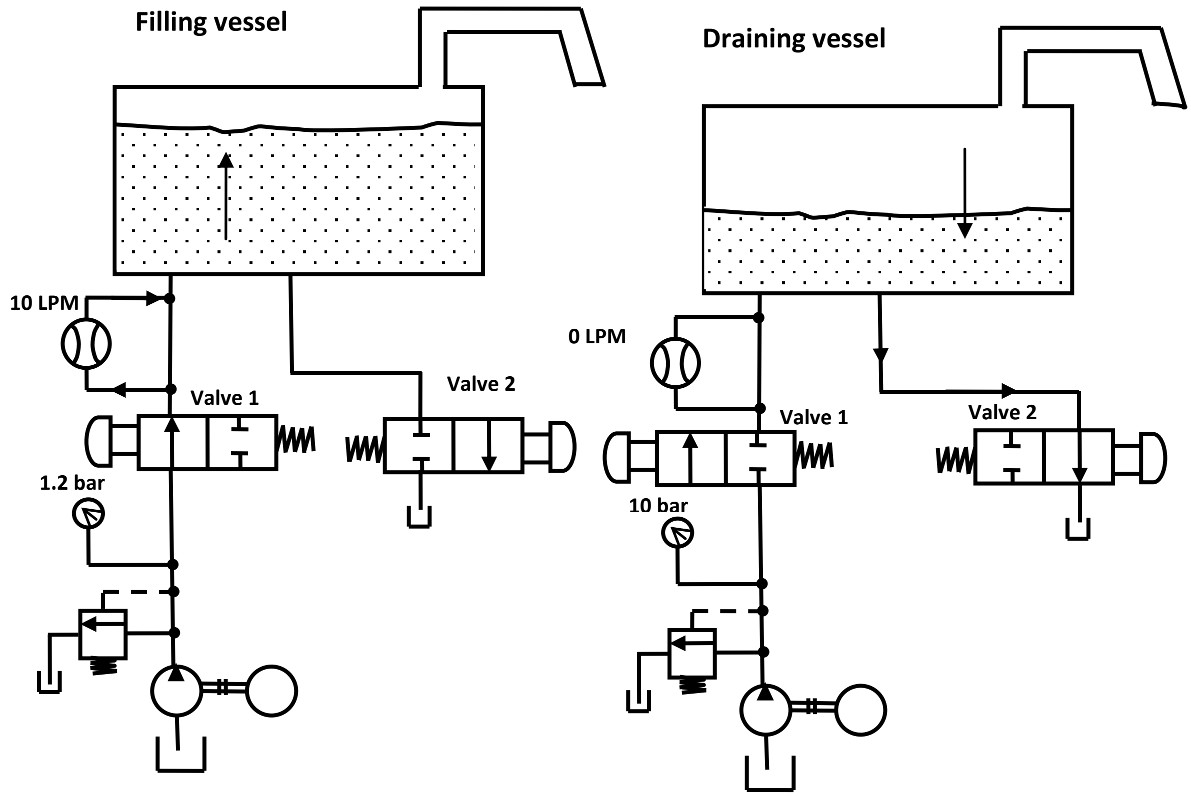


Figure 1.8 2/2 DCV normally opened. (a) Ports A and P are connected when force is not applied (valve unactuated). (b) Ports A and P are not connected when force is applied
(valve actuated)

1.7.3 *Application of 2/2 DCV*

A pair of two-way valves is used to fill and drain a vessel. In Fig.1.9(a), valve 1 is shifted to the open position, while valve 2 remains closed. This fills the vessel. In Fig.1.9(b), valve 2 is shifted to open position and valve 1 remains closed. This drains the vessel.



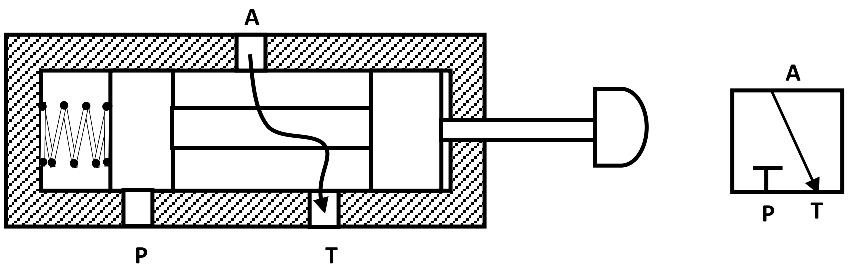
(a) (b)

Figure 1.9 Application of 2/2

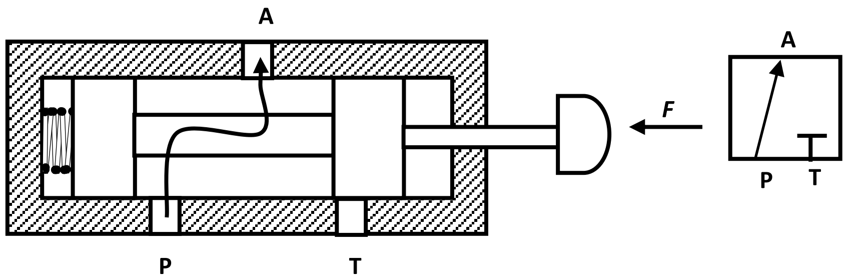
1.8Three-Way Direction Control

1.8.1 ***3/2-Way DCV (Normally Closed)***

Three-way valves either block or allow flow from an inlet to an outlet. They also allow the outlet to flow back to the tank when the pump is blocked, while a two-way valve does not. A three-way valve has three ports, namely, a pressure inlet (P),an outlet to the system(A)and a return to the tank(T). Figure 1.10shows the operation of a 3/2-way valve normally closed. In its normal position, the valve is held in position by a spring as shown in Fig. 1.10(a). In the normal position, the pressure port P is blocked and outlet A is connected to the tank. In the actuated position shown in Fig. 1.10(b), the pressure port is connected to the tank and the tank port is blocked.



(a)

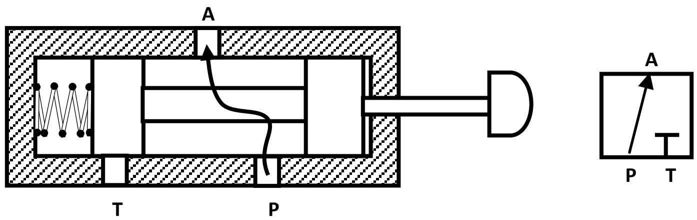


(b)

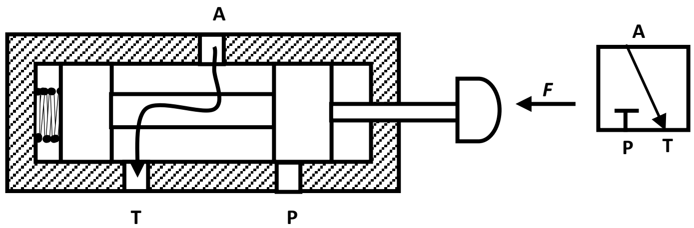
Figure 1.10 3/2-way DCV (normally closed). (a) Ports A and T are connected when force is not applied (valve unactuated). (b) Ports A and P are connected when force is applied (valve actuated).

1.8.2*3/2-Way DCV (Normally Opened)*

Figure 1.11 shows a three-way two-position DCV (normally open)with push button actuation and spring return. In the normal position, shown in Fig. 1.11(a), the valve sends pressure to the outlet and blocks the tank port in the normal position. In the actuated position, the pressure port is blocked and the outlet is vented to the tank.



(a)



(b)

(a) **Figure 1.11** 3/2-way DCV (normally opened). (a) Ports A and P are connected when force is not applied (valve unactuated). (b) Ports A and T are connected when force is applied (valve actuated).

1.8.3 *Applications of 3/2 DCV and 3/3 DCV*

3/2 DCV and 3/2 DCV find application in the following ways: **Application of 3/2 DCV for controlling a single-acting cylinder:** A 3/2 DCV is used to control a single-acting cylinder. Figure 1.12(a) shows the valve in its normal position in which the pressure port is blocked and the outlet is returned to the tank. This allows the force of the to act on the piston and retract the cylinder. The cylinder remains in the retracted position as long as the valve is in this position. In Fig.1.12(b), the valve position is shifted by the actuation of the push button. This connects the pressure port P with outlet A and the tank port is blocked. This applies pump flow and pressure to the piston and the cylinder extends against the light force of the spring.

1. **Application of 3/3 DCV in filling and draining the vessel:** A three-way, three-position DCV may be used to fill and drain a vessel. In this application, the closed neutral is required to hold the vessel at some constant fluid level(Fig. 1.13).
2. **Application of 3/3 DCV in controlling a gravity return single-acting cylinder:** A gravity return-type single-acting cylinder is controlled by a three-way DCV. A third position called neutral may be desired for its application. This position shown as the center position in the symbol blocks all these ports. This position holds the cylinder in a mid-stroke position. Many cylinder applications require this feature. Figure 1.14 introduces another type of actuation manual lever and detent. A detent is a mechanism that holds the valve in any position into which it is shifted. The detented valve has no normal position because it remains indefinitely in the last position indicated. When the valve is in the closed neutral position or the retract position, the pump flow goes over the pressure relief valve because the pressure port is blocked.
3. **Application of 3/2-way valve for controlling a double-acting cylinder:** Double-acting cylinders can be controlled with two 3/2-way valves so arranged that when one valve pressurizes one end of the cylinder, the other valve exhausts the other end and vice versa(Fig. 1.15).

Figure 1.12 Application of 3/2 valve–control of single-acting cylinder:(a) return; (b) extend.

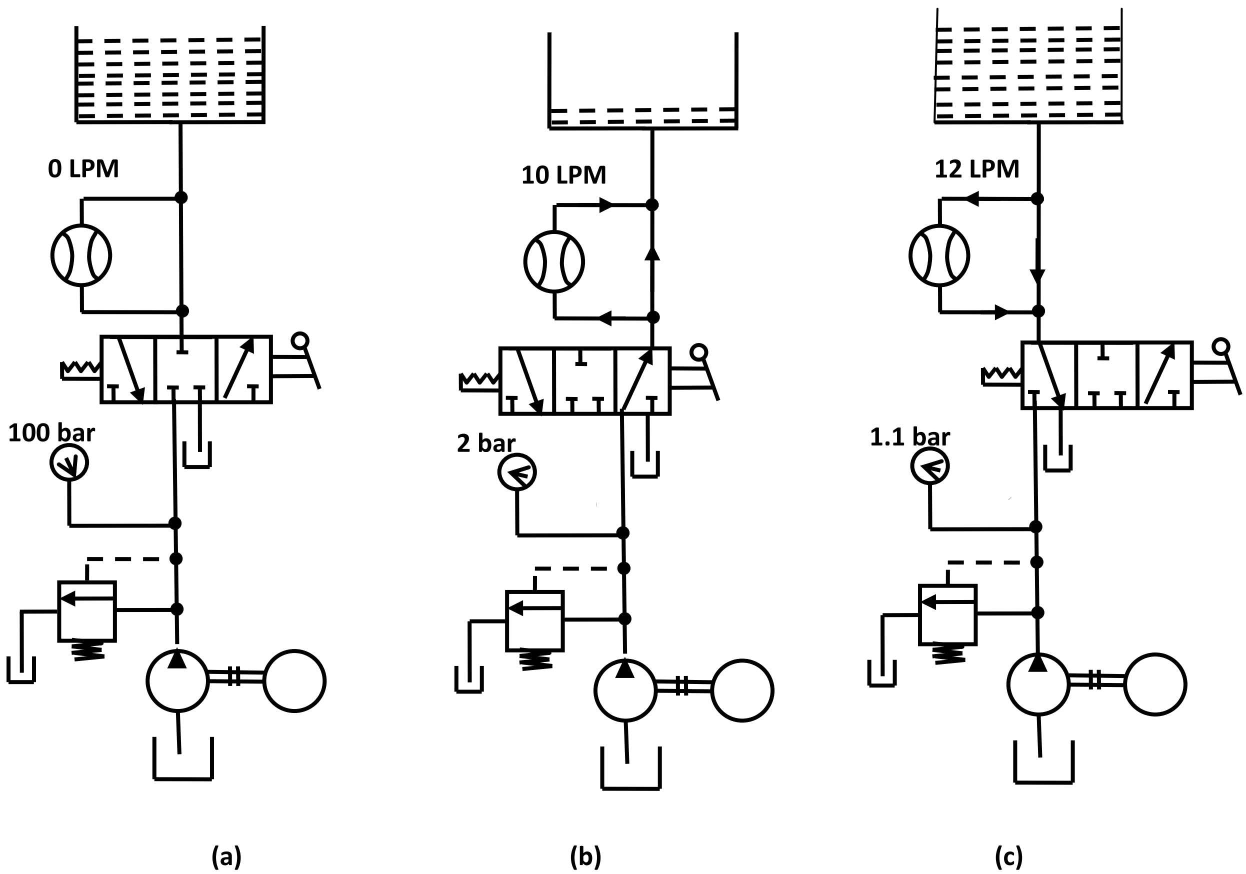


Figure 1.13 Application of 3/2 valve–filling and draining a vessel: (a) hold; (b) fill; (b) drain.

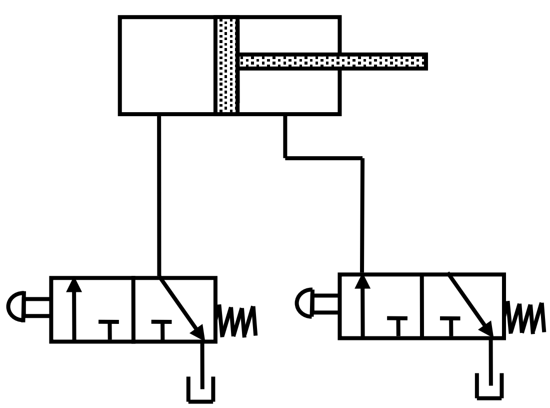


Figure 1.14 Application of 3/2 valve –controlling a double-acting cylinder.

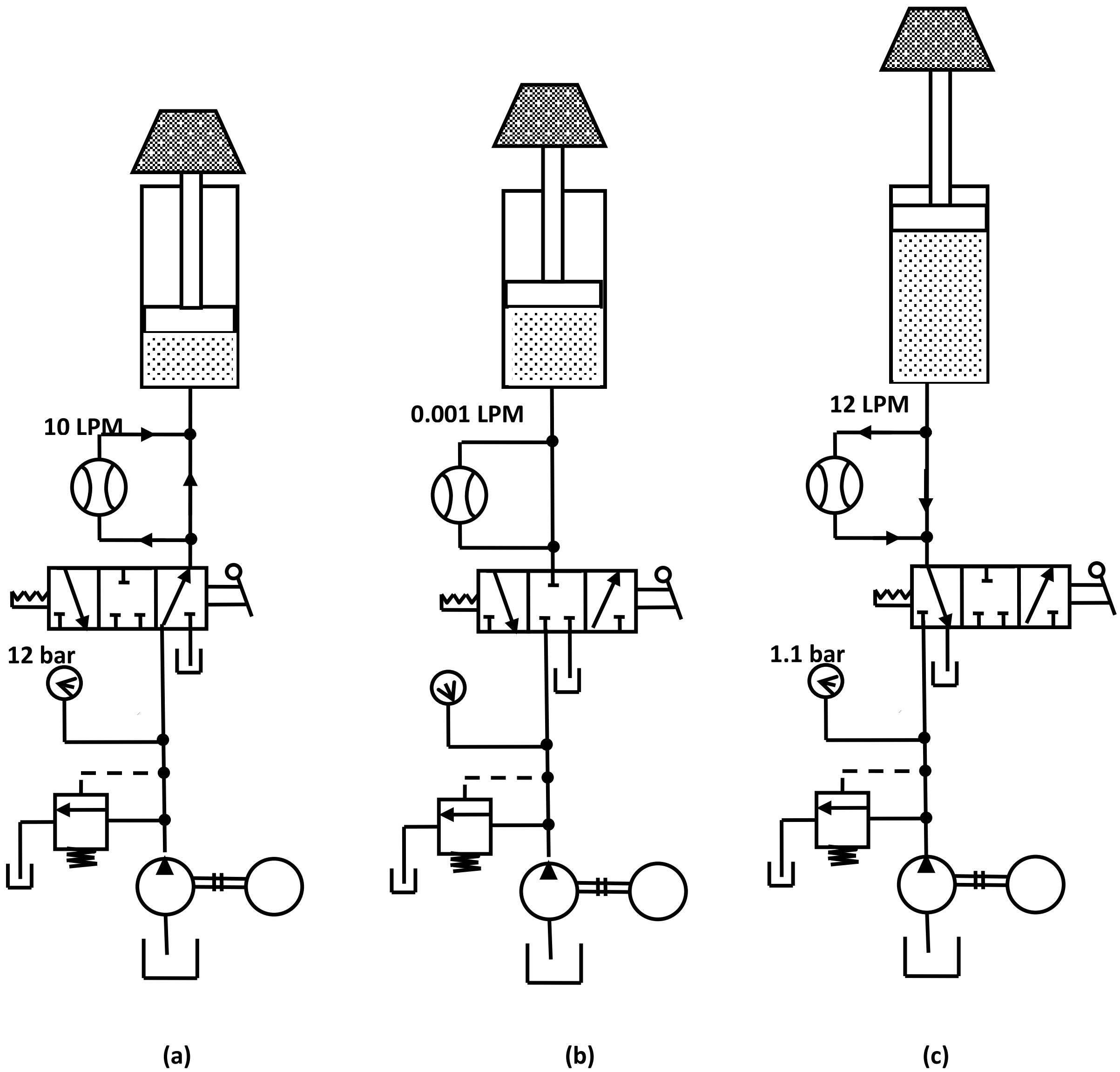


Figure 1.15 Application of 3/3 valve –controlling a single-acting cylinder.

1.9Four-Way Direction Control Valves

Four-way DCVs are capable of controlling double-acting cylinders and bidirectional motors. Figure 1.16 shows the operation of a typical 4/2 DCV. A four-way has four ports labeled P,T,A and B. Pis the pressure inlet and T is the return to the tank; A and B are outlets to the system. In the normal position, pump flow is sent to outlet B. Outlet A is connected to the tank. In the actuated position, the pump flow is sent to port A and port B connected to tank T. In four-way DCVs, two flows of the fluids are controlled at the same time, while two-way and three-way DCVs control only one flow at a time. Figure 1.16 (c) shows the complete graphic symbol for a four-way two-piston DCV.

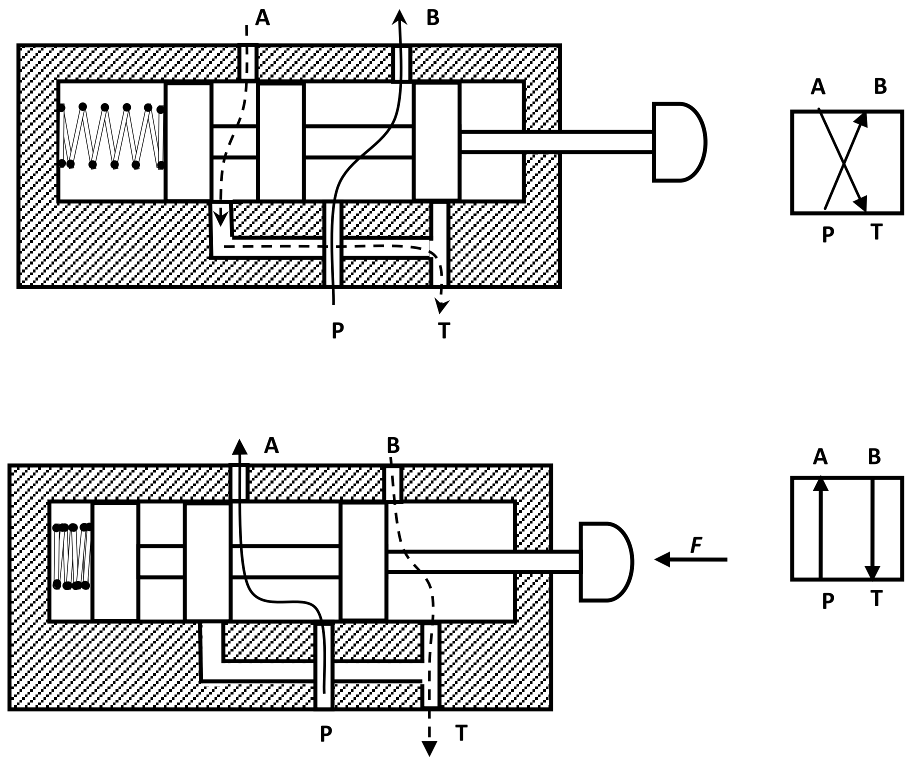


Figure 1.16 Four-way DCV.

1.9.1 ***Applications of 4/2 DCV and 4/3 DCV***

4/2 DCV and 4/3 DCV find applications in the following ways:

1. **Application of 4/2-way valve to control a double-acting cylinder:** A four-way DCV is used to control a double-acting cylinder. When the valve is in the normal position, the pump line is connected to the end of the cylinder and the blind end is connected to the tank as shown in Fig.1.17(a). The cylinder retracts when the cylinder is in this position. When the cylinder is fully retracted, the pump flow goes over to the pressure relief valve and back to the tank. In Fig.1.17(b), the pump line is connected to the blind end of the cylinder and the rod end is connected to the tank. This causes the cylinder to extend. When the cylinder is fully extended, the pump flow again goes over the pressure relief valve to the tank.

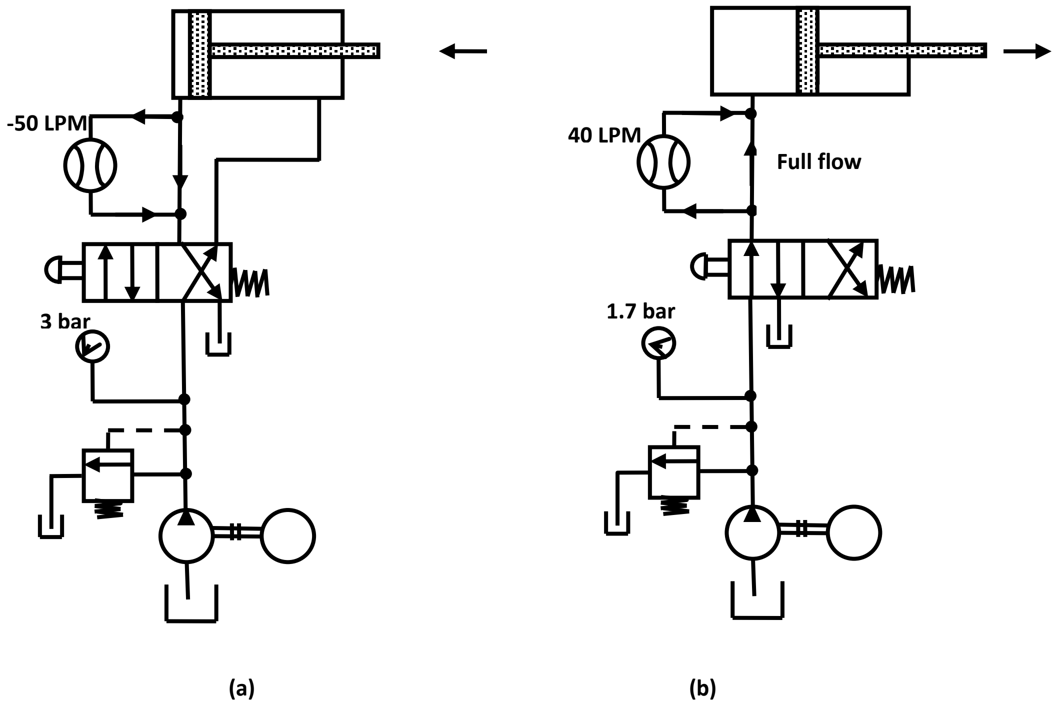
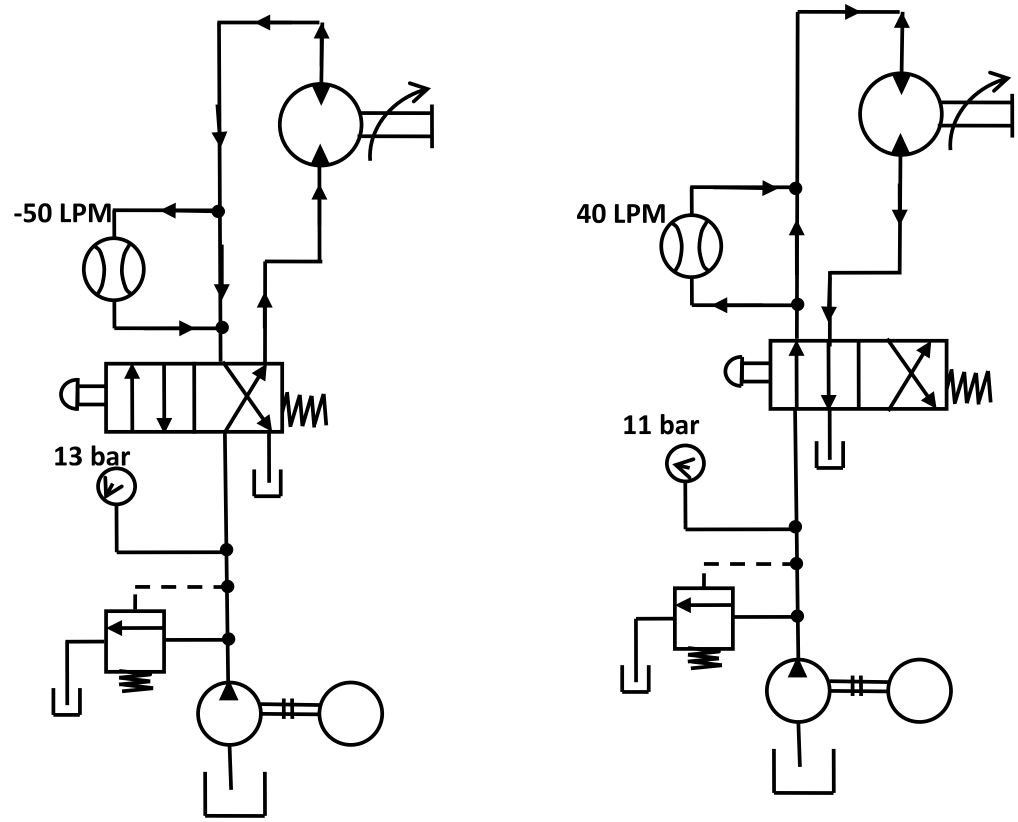


Figure 1.17 Application of a 4/2-way valve – control of single-acting cylinder: (a) return; (b) extend.

2. **Application of 4/2 DCV for controlling bi-directional motors:** A four-way DCV is also used to control bi-directional hydraulic motors. Figure 1.18shows the schematic for this application. Unlike the cylinder, the motor rotates continuously and does not force the fluid over the pressure relief valve.



(a) (b)

Figure 1.18 Application of 4/2-way valve – control of a bi-directional motor.

The four-way, two-position DCVs used in the previous two applications are sometimes impractical because they continually send pump flow and pressure to the actuator in one direction or the other. Many cylinder and motor applications require a third DCV position or neutral in which the actuator is subjected to pump pressure. Four-way three-position circuits are therefore used in many hydraulic circuits. Many types of neutrals are available; the most common of them are as follows:

- Closed neutral.
- Tandem neutral.
- Float neutral.
- Open neutral.
- Regenerative neutral.

3. **Application of 4/3 DCV (closed neutral) for controlling a double-acting cylinder:** Figure 1.19 shows it in a simple cylinder circuit. The valve shown here is spring centered, which means that it always returns to the neutral position automatically when not actuated. For closed neutral, the pump line is blocked so that the flow must pass over the pressure relief valve the pressure is at the system maximum. This is wasteful thing because it generates power in the form of pressure and flow, but does not use it. The wasted energy in the system goes as heat. This is undesirable because the hydraulic fluid becomes thinner (less viscous) as it heats up. When the fluid becomes

too thin, it does not lubricate effectively. This is the result of increased wear.The outlet lines to the cylinder are blocked, so the cylinder is held trimly in position. This is because the lines are full of hydraulic fluid that is incompressible. This type of neutral could also be used to control a motor. Just like cylinder, the motor is held tired in position when the valve is in the neutral.

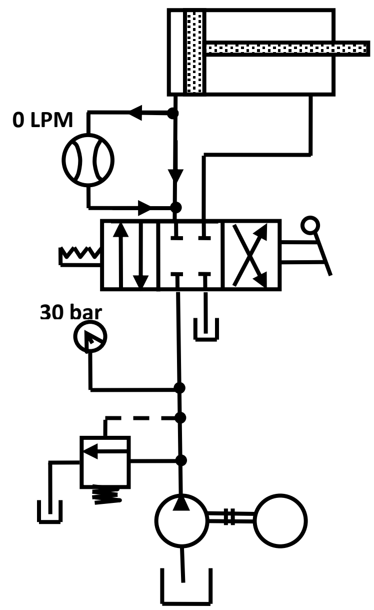


Figure 1.19 Application of 4/3-way valve – closed neutral.

4. **Application of 4/3 DCV (tandem neutral) for controlling a double-acting cylinder:** Figure 1.20 shows it in a simple cylinder circuit. The pump flow is allowed to flow back to the tank through the DCV when it is in the neutral. This is a very desirable situation because only pressure in the pump line is due to the flow resistance of the lines and DCV. This keeps the pressure low when the valve is in the neutral. In this situation, the system is said to be unloaded because the power consumption is reduced. This wastes much less energy than does a closed central neutral that forces the fluid over the pressure relief valve at a high pressure. The cylinder is held in position with a tandem neutral because the outlet port is blocked.

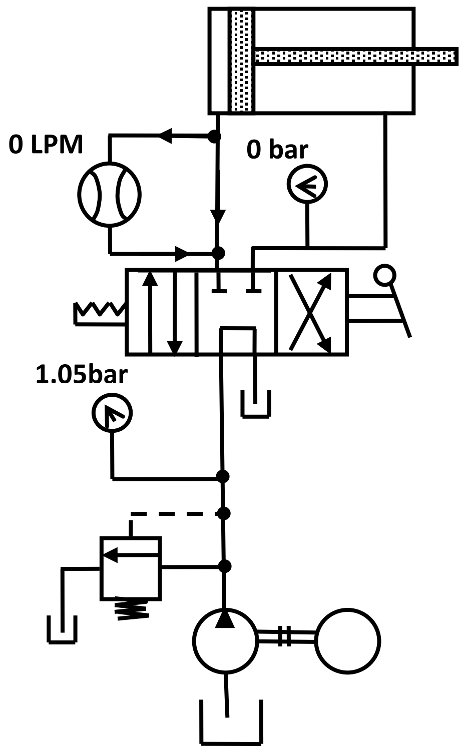


Figure 1.20 Application of 4/3-way valve – tandem neutral.

5. **Application of 4/3 DCV (float neutral) for controlling a bidirectional motor:** Figure 1.21 shows a four-way with a float neutral controlling a bidirectional motor. The pressure port is blocked so that the pump flow is forced over the pressure relief valve. Because both the outlets are connected to the tank, the motor floats or spins freely when the DCV is in the neutral. This type is used in motor circuits because it allows the motor to spin to a stop when the valve is shifted to the neutral. This is often preferable to shifting to a closed position because motors often build up a great deal of momentum. Shifting the valve closed in this situation causes a large pressure hike in the outlet line because the motor tends to keep spinning and tries to push the fluid into its outlet. This is known as shifting shock. Float neutrals are often desirable for cylinder circuits in some applications.

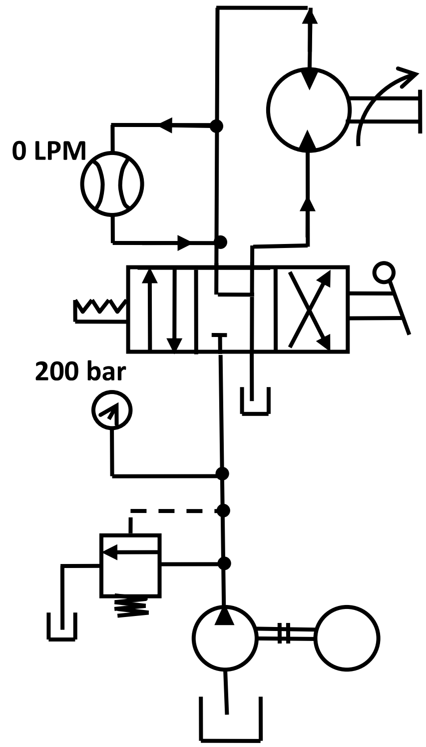


Figure 1.21 Application of 4/3-way valve – floating neutral.

6. **Application of 4/3 DCV (open neutral) for controlling a double-acting cylinder:** Figure 1.22 shows the four-way with an open neutral controlling a cylinder. Flow always follows the path of least resistance, so the pump flow goes back to the tank. Because the outlets are also connected to the tank, the cylinder floats when this valve is in neutral. This is desirable in a circuit in which some external force must position the cylinder when in the neutral.

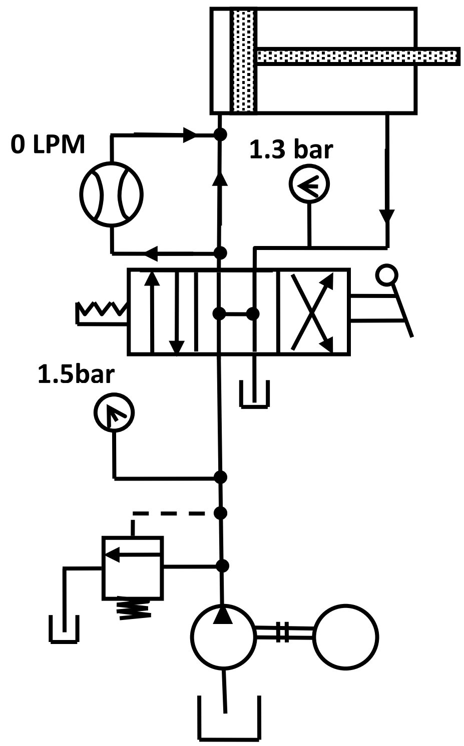


Figure 1.22 Application of 4/3-way valve – open neutral.

Application of 4/3 DCV (regenerative neutral) for controlling a double-acting cylinder: A regenerative neutral is considerably different in its function than other types. A regenerative term is used to describe a system in which the waste is fedback into the system to supplement the input power. In this neutral, the pressure port is connected to both outlets and the tank port is blocked. Figure 1.23 shows a four-way with a negative neutral controlling a cylinder. When this valve is shifted to the neutral, the pump pressure is applied to both sides of the piston. Because the piston area in the rod side of the cylinder is smaller than that on the blind side, there is a net force applied to extend the piston rod. As the piston extends, it forces the outlet flow from the rod side back into the valve, where it combines with the pump flow and goes to the blind end of the cylinder. This causes the considerable increase in cylinder speed. This is the purpose of the regenerative neutral that instead of sending the return flow back to the tank, it sends it into the inlet side of the cylinder, thereby increasing its speed.

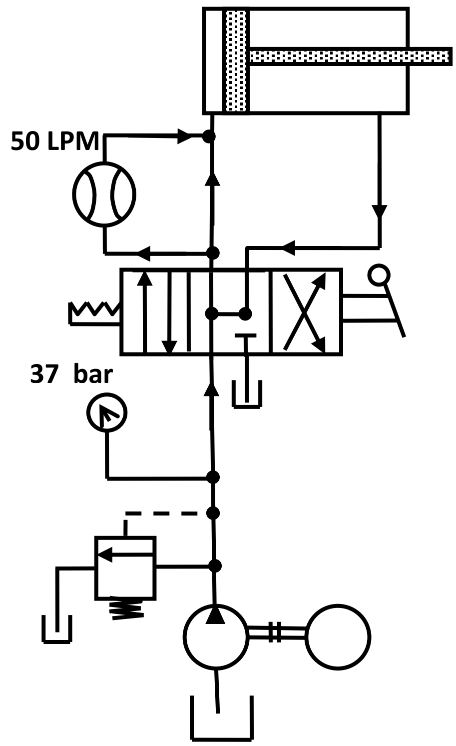


Figure 1.23 Application of 4/3-way valve – regenerative neutral.

1.10 Solenoid-Actuated Valve

A spool-type DCV can be actuated using a solenoid as shown in Figure. 1.24. When the electric coil (solenoid) is energized, it creates a magnetic force that pulls the armature into the coil. This causes the armature to push on the push pin to move the spool of the valve.

Like mechanical or pilot actuators, solenoids work against a push pin, which in turn actuates a spool. There are two types of solenoid designs used to dissipate the heat developed in electric current flowing in the coil. The first type dissipates the heat into surrounding air and is referred to as an "air gap solenoid." In the second type "wet pin solenoid," the push pin contains an internal passage way that allows the tank port oil to communicate between the housing of the valve and the housing of the solenoid. Wet pin solenoids do a better job in dissipating heat because the cool oil represents a good heat sink to absorb heat from the solenoid. As the oil circulates, the heat is carried into the hydraulic system where it can be easily dealt with.

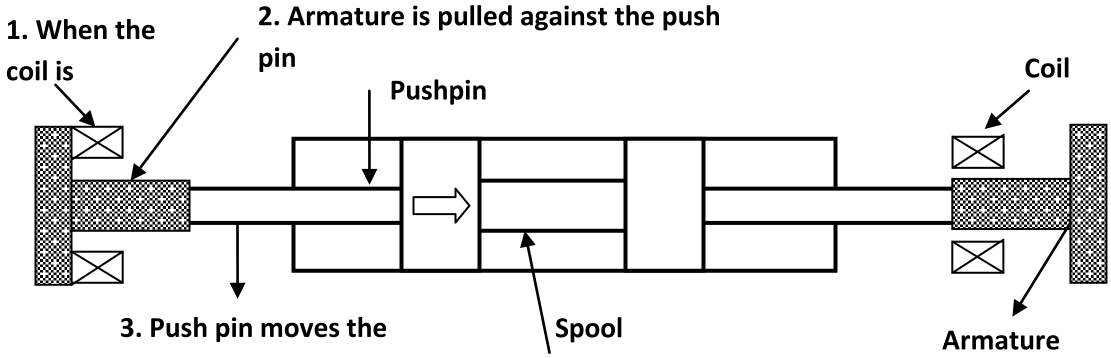


Figure 1.24 Solenoid valve.

In the case of direct current (DC) solenoids, the current develops a magnetic field of fixed polarity. The DC solenoids are practically safe from burning out if the correct voltage is applied. The solenoid force depends not only on the solenoid design and current but also on the core position. The available commercial solenoids produce a force of 60–70 N. For a greater force, the number of turns of coil or current should be increased.

Alternating current (AC) solenoids function in the same manner as DC solenoids but their magnetic fields are influenced by the alternating current. The magnetic force is high when AC current is at its positive or negative peak. As the current changes from positive to negative, it must pass through neutral points where there is no current or no force. Due to this, load can push the core slightly out of equilibrium. This is commonly referred to as buzz. To eliminate buzz, shading coils are used. A shading coil creates its own magnetic field but the current produced lags behind the coil current and thus helps to prevent buzz. A comparison between AC and DC solenoids is given in Table 1.3.

Table 1.3 Comparison between AC and DC solenoids

Parameter	**DC Solenoid**	**AC Solenoid**
Switching time	50–60 ms	20 ms
Service life expectations	20–50 million cycles	10–20 million cycles
Max. switching frequency	Up to 4 cycles/s	Up to 2 cycles/s
Continuous operation	Unlimited	15–20 min for dry solenoids. 60–80 min for wet solenoids
Relative cost	1	1.2
Occurrence rate	10	2

1.11Pilot-Operated Direction Control Valves

Pilot-operated DCVs are used in a hydraulic system operating at a high pressure. Due to the high pressure of the system, the force required to actuate the DCV is high. In such systems, operation at a high pressure uses a small DCV that is actuated by either a solenoid or manually. This pilot DCV in turn uses the pressure of the system to actuate the main DCV as shown in Figure. 1.25.

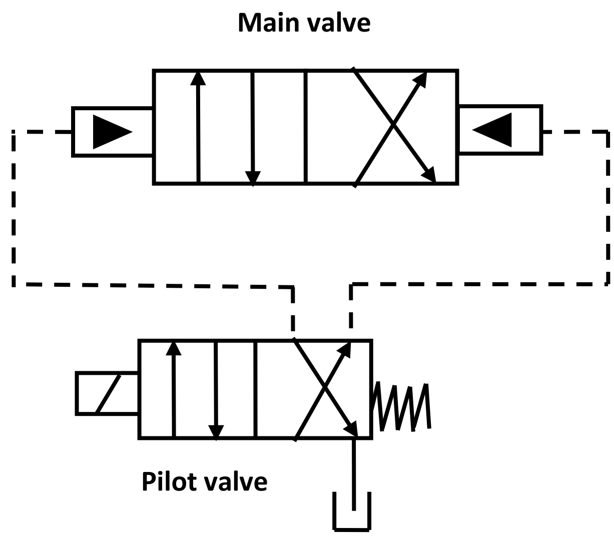


Figure 1.25 Pilot-operated DCVs.

Lecture 17

DIRECTIONAL CONTROL VALVE [continued]

1.11.1 Applications of Pilot-Operated Valve to Control the Table of a Surface Grinder

Figure 1.26shows the application of a pilot-operated DCV where the actuation of a double-acting cylinder is used to reciprocate the table of a surface grinder. The table is fitted with adjustable stops as shown in the figure. The pilot valve is a DCV that is actuated by a push button. During the operation when stop S_1 hits push button B_1, the pilot valve sends a pilot signal to the main valve to shift the configuration shown in the right envelope of the main valve. This actuates the double-acting cylinder to extend. At the end of the extension, stroke S_2 hits push button B_2, which causes the pilot signal directions to be reversed. Due to this change in the pilot signal direction, the main valve moves to the configuration shown in the left envelope of the main valve. This in turn actuates the double-acting cylinder to retract.
Thus, a pilot valve controls a main valve and the main valve used to control the double-acting cylinder.

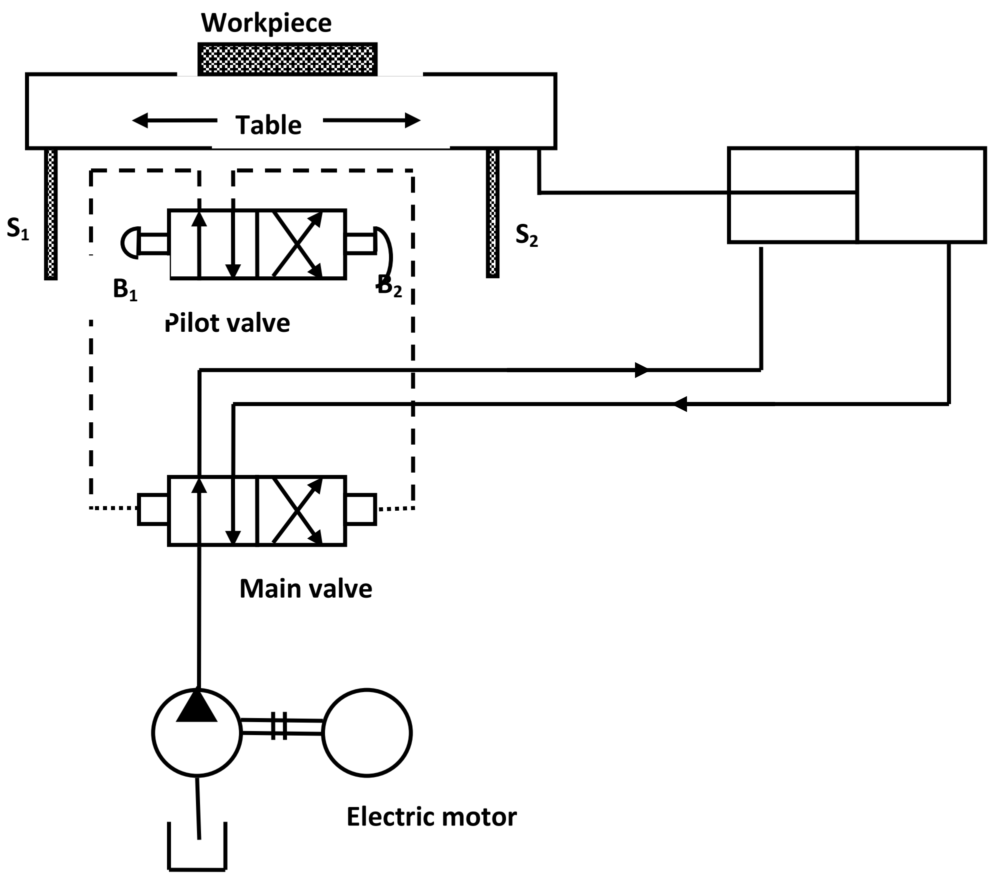


Figure 1.26 Application of pilot-operated DCVs.

1.12Piston Overlap

The switching characteristics of a valve are decided by the piston overlap. A distinction is made between the positive, negative and zero overlap. Overlap is defined as the longitudinal difference between the

length of land and that of the port. The magnitude of overlap changes during unoperated and operated conditions.
The piston overlap determines the oil leakage rate. Overlapping is significant for all types of valve. The most favorable overlap is selected in accordance with the application.

1. **Positive switching overlap:** During the reversing procedure, all parts are briefly closed against one another. Hence, switching imparts “pressure peaks” and make hard advance.
2. **Negative switching overlap:** During the reversing procedure, all ports are briefly interconnected. Pressure collapses briefly (load drops down).
3. **Zero overlap:** Edges meet. Important for fast switching, short switching paths.
4. **Pressure advanced opening:** The pump is first of all connected to the power componentand then the power component is discharged into the reservoir.

5. Outlet advanced opening: The outlet of the power component is first discharged to the reservoir before the inlet is connected to the pump.

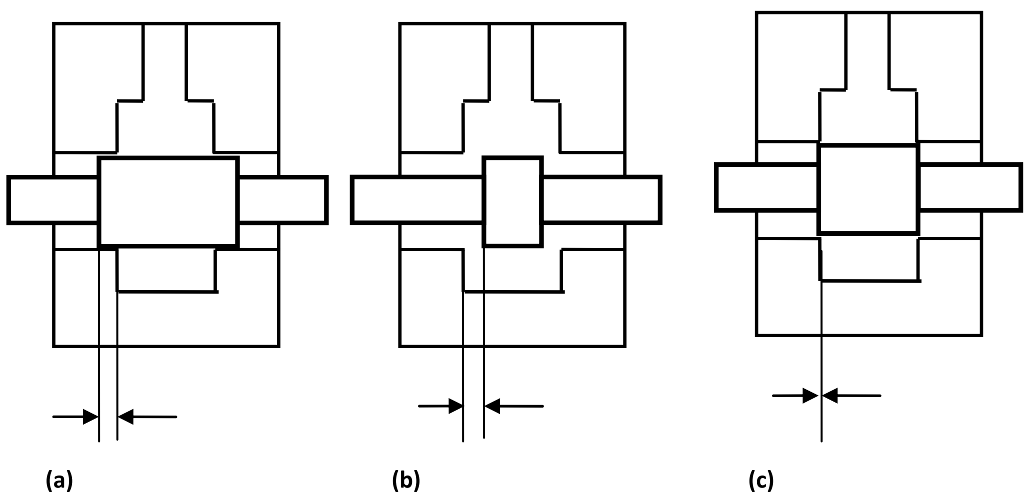


Figure 1.27 Valve overlap: (a) Positive overlap; (b) negative overlap; (c) zero overlap.

1.12Miscellaneous Industrial Circuits

This section examines some simple circuits that are commonly used in industry. This will help the reader to develop the ability to read hydraulic schematics and to understand the operation of basic circuits.
Figure1.28shows a circuit in which a cylinder is used to raise and lower a large weight from above. The cylinder is controlled by a four-way DCV with a tandem neutral. In Fig. 1.28(a), the DCV is in the neutral.Therefore, the pump flow is unloaded to the tank at a low pressure. The cylinder should hold position because the outlet ports from the DCV that connect to the cylinder are blocked. It does not hold position, however, if the cylinder is in the orientation shown because the weight pulls the cylinder down, causing pressure in the rod end line. The pressure causes a small amount of leakage within the DCV, and the cylinder begins to creep downward. This can be remedied by placing a pilot-to-open check valve in the rod end line, as shown. The pilot-to-open check valve does not allow flow out of the rod end of the cylinder unless pressure is applied to the pilot line, thereby preventing cylinder creep. The check acts to counterbalance the weight. When the DCV is shifted to extend the cylinder (lowering the weight), the pump pressure from the blind end line holds open the check and allows flow to return to the tank from the blind end. When the cylinder is retracted (the weight is raised), flow from the pump goes through the check to the rod end. The check has no effect in this direction.

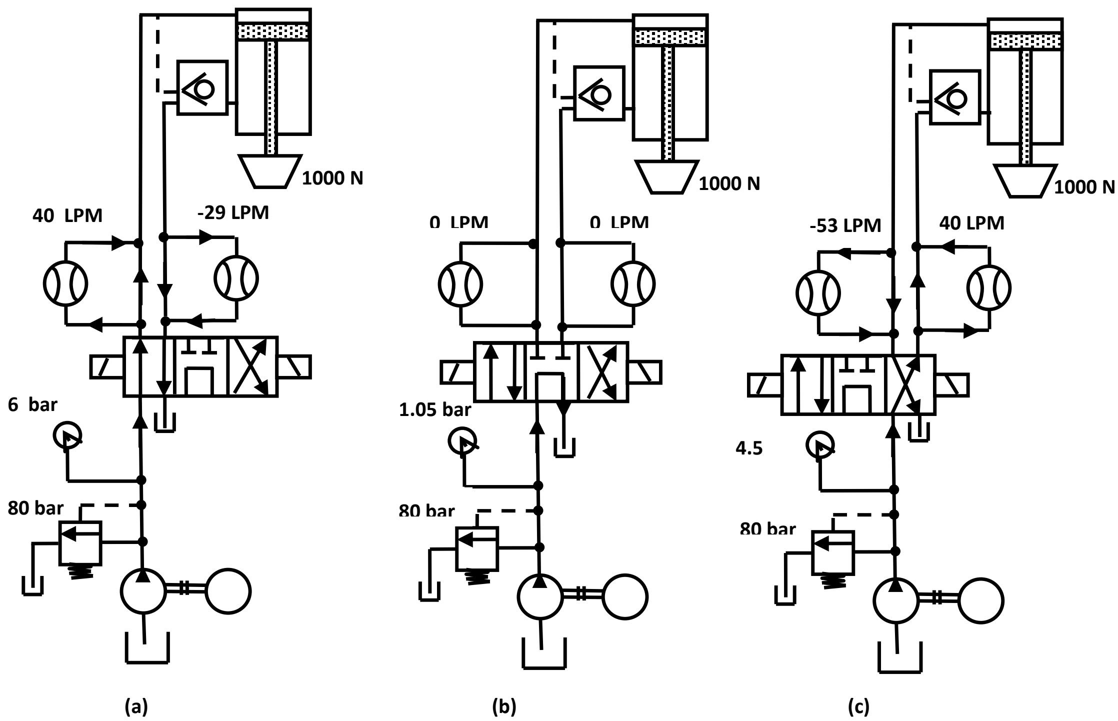


Figure 1.28 Raising and lowering large weights: (a Extend cylinder; (b) hold cylinder; (c) return cylinder.

Figure1.29 shows a circuit that utilizes a shuttle valve. This circuit allows either of the two three-way buttons to operate a single-acting cylinder. The figure shows both three-ways in their normal positions. The cylinder is vented to the tank and remains retracted under the force of the spring. In Fig. 1.30, three-way number 1 is shifted and pump flow is sent to the cylinder through the path shown. In Fig. 1.31, the cylinder is extended with valve number 2. This circuit could be used on a long machine with buttons on either end for convenience. A shuttle valve is used in many other applications in which one of two flow paths may supply a single branch of circuit.

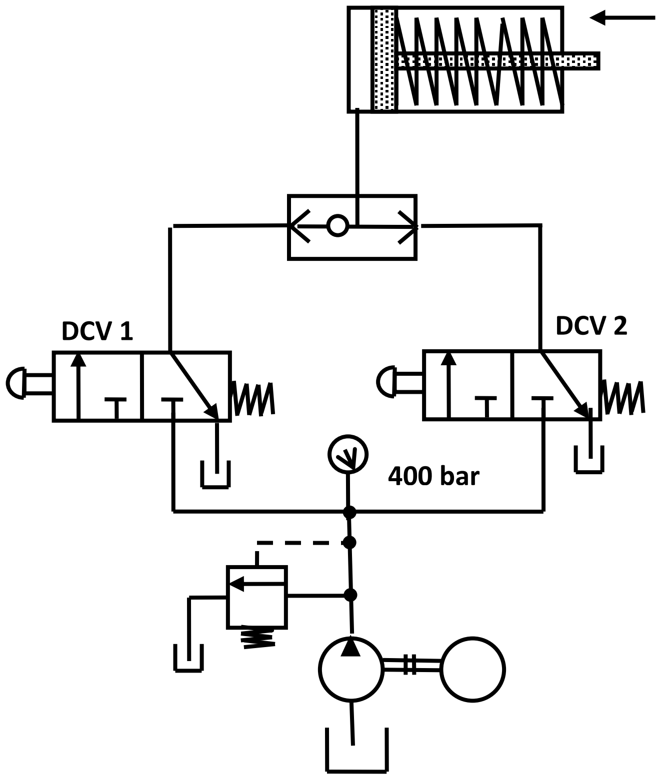


Figure 1.29 Use of shuttle valves to control single-acting cylinders(return).

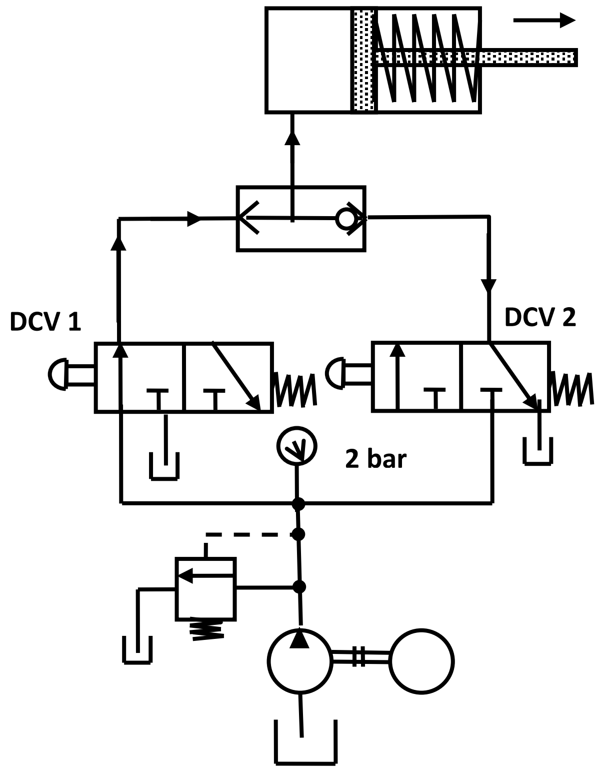


Figure 1.30 Use of shuttle valves to control single-acting cylinders (forward).

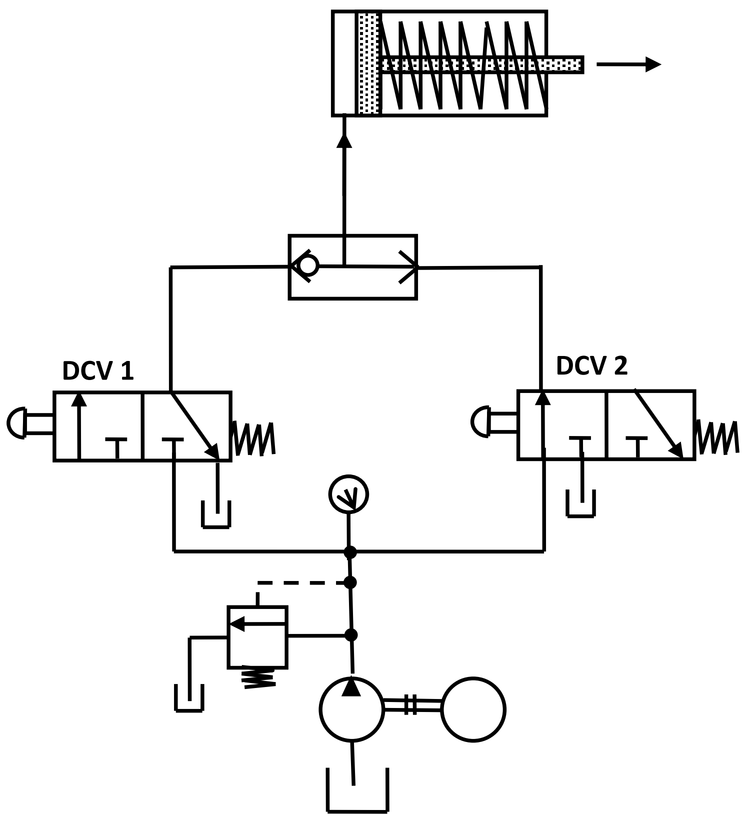


Figure 1.31 Use of shuttle valves to control single-acting cylinders (forward).

Figure1.32 shows a regenerative circuit that automatically switches off regeneration when full force is necessary. This circuit could be used in a hydraulic press where the cylinder must extend quickly under no load, then bottoms out and must apply full force to the work piece. Instead of using a four-way with a regenerative neutral, this circuit uses a four-way DCV in conjunction with a three-way, pilot-operated DCV. The three-way is shifted when sufficient pressure is applied to its pilot line, which is connected to the blind end of the cylinder. In the figure, the four-way is shifted to the left position and flow is sent to the blind end of the cylinder. Because the cylinder is not loaded, the pressure in the blind end is very low and is not sufficient to shift the three-way. The flow from the rod end combines with the pump flow, causing the cylinder to extend rapidly. When the cylinder bottoms out, pressure immediately builds up in the blind end line to the relief valve setting because there is no other path for pump flow. The three-way valve is then shifted into the left position and pressure is relieved from the rod side because it is connected to the tank port. The pressure is then applied to the blind sidethat causes full force to be applied to the work piece. When the four-way DCV is shifted into the right position, the cylinder retracts at a normal speed. In this circuit, the reduction in force capability caused by regeneration is not an issue because during the regeneration portion of the cycle, the cylinder is not loaded. The primary advantage of using regeneration is that a smaller pump can be purchased that is less expensive to buy and operate.

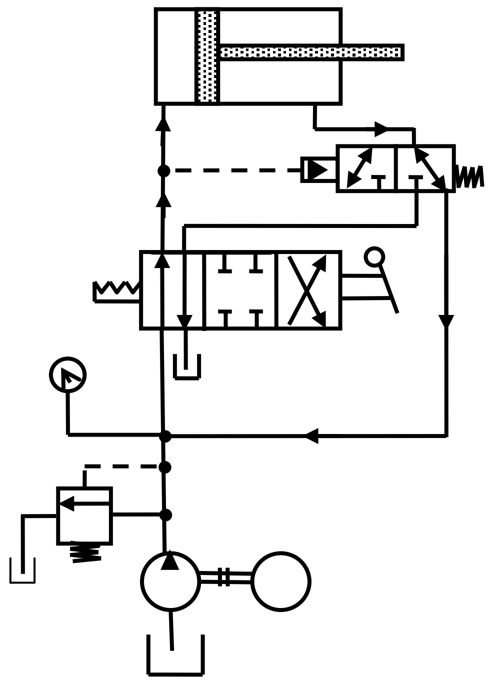


Figure 1.32 Regenerative circuit (position 1).

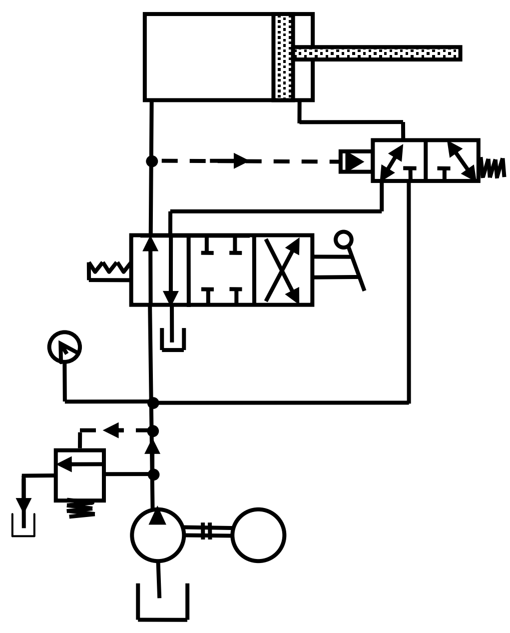


Figure 1.33 Regenerative circuit (position 2).

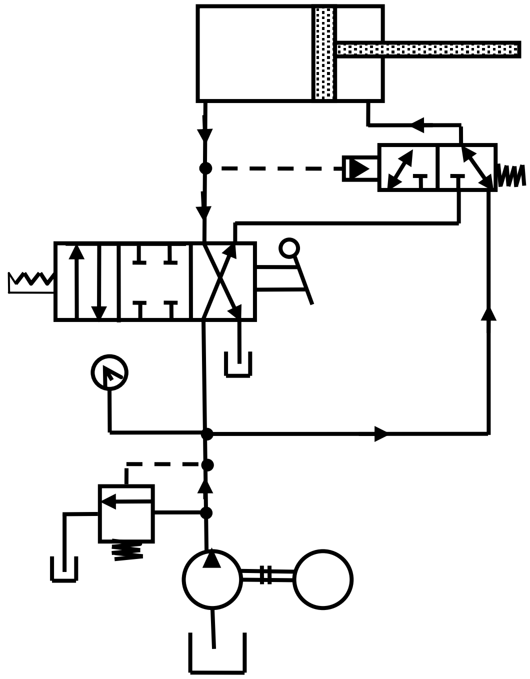


Figure 1.34 Regenerative circuit (position 3).

1.13Direction Control Valve Mounting

DCVs can be mounted in two ways:

1. **Inline:** There are threaded connections in the valve itself. Fittings are screwed directly into the valve. This method has several major disadvantages:

- Each time the valve is disconnected; there is a probability of damaging the valve by stripping the threads.
- The threads wear each time the unit is disconnected, causing contamination and increased probability of leakage.

2. **Sub-plate:** The bottom of the valve has unthreaded connections. The valve is then attached to a sub-plate that has matching connections. The sub-plate has threaded connections to which the fittings are attached. Sealing at the valve interface is by using O-rings, which fit into small recesses around the DCV port. The advantages are as follows:

- The sub-plate has less leakage, less contamination and a smaller probability of doing damage during assembly and disassembly.
- Valve replacement is simpler and easier.
- Multiple valves can be connected on a manifold.

The valves and sub-plates are available with several standard patterns for the valve ports.

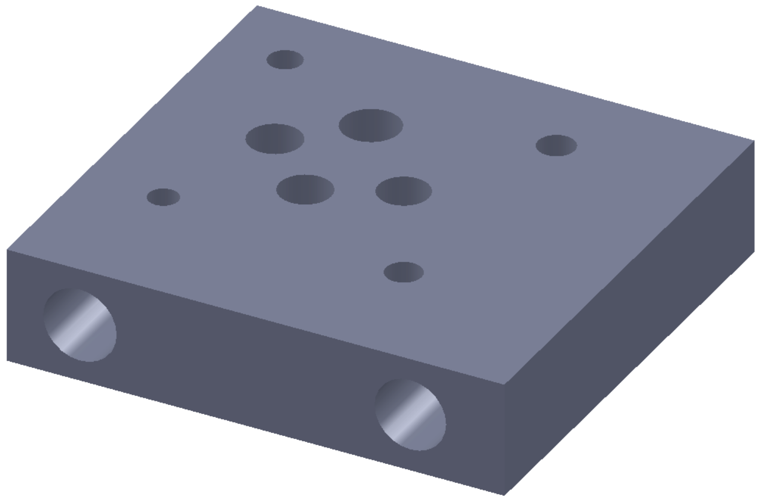


Figure 1.35 DCV mountings.

A manifold is a sub-plate that has connection for two or more valves. This method can be used to create an integrated hydraulic circuit in which many connections are inside the manifold itself, eliminating the need for fittings and plumbing between the valves. The advantages are a more compact design, less leakage, less contamination, easy replacement of valves, etc. The only drawback is that it requires more design and testing time thereby increasing the expense.

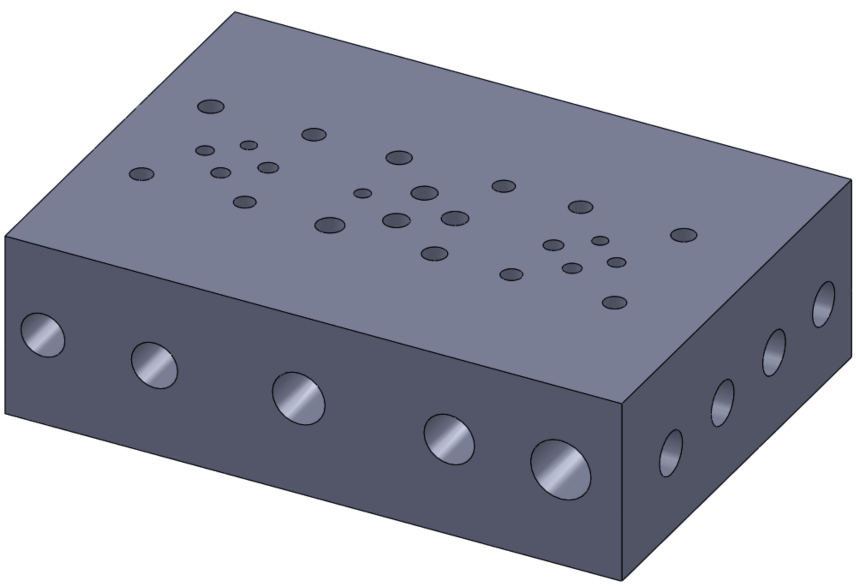


Figure 1.36 DCV mounting manifold.

Cartridge valves are also used in conjunction with manifolds. They are very compact and alternative to a spool-type design. They screw directly into a cavity in a manifold and therefore do not require a separate valve port and mounting holes. The advantages of manifold circuits are magnified when cartridge valves are used. ******

1.8DCVSpecifications

The most critical specification when selecting a DCV is its maximum pressure and flow ratings. It is common for a high pressure rating to be given for the pressure and outlet ports and a lower value to tank port. Ratings of 3000–5000 psi are typical for the former, while the latter has ratings of 500–1000 psi. The different ratings are because the spool seal is often exposed to the tank port, but no other ports. Valves that can handle a high pressure to all of their ports are also available. Operating above the maximum pressure rating leads to increased leakage and also permanent damage to the valve.
The flow rate is largely determined by the size of the valve itself. Larger valves can handle larger flow rates but are heavy and expensive. Standard valves have ratings 10–250 GPM. Operating a DCV above its maximum flow rating most likely results in a large pressure drop across the valve. This lost energy is converted into heat and is not wasteful, but leads to increased component wear as the oil becomes thinner and doesnot lubricate. Operating above the maximum flow rating leads to permanent damage to the valve itself.
When selecting a DCV for an application, we may also know what the pressure drop will be across the valve at a particular flow rate. Manufacturers typically provide graphs that relate pressure drop to flow rate through valve for each model. Separate curves are given for different port-to-port connections. These curves represent data for a particular fluid and viscosity, most commonly standard hydraulic oil at 100 SSU. Manufacturers often give a correction factor for fluids at other viscosities. A fluid with a higher viscosity has a higher pressure drop at a given flow rate because a thicker fluid is more difficult to move through the valve.

1.9Material for DCVs

Following are the materials for DCVs:

1. **Valve body:** It is made of carbon steel, ductile cast iron and stainless steel. Aluminum alloys are also preferred for low-pressure applications. High-strength aluminum alloys are used for aircraft applications. Stainless steel is used for corrosive environment. Sometimes plastics are also used for low-temperature applications**.**
2. **Valve spool:** It is made of hardened steel, ground and polished 15 Ni2Cr1Mo15 of hardness 60-62 HRC, machined to 2–3 µm tolerance. Valve spool bore clearance is usually in the order of 5–10 µm.

Example 1 A cylinder with a bore diameter of 7 cm and a rod diameter of 3.125 cm is to be used in a system with a 45 LPM pump. Use the graph in Fig. 1.36to determine the pressure drops across the DCV when the cylinder is retracting (P->B,A->T).

Solution

The flow from P to B is the pump flow into the rod end, so this can be read from the graph

$$\Delta p = 3.2 \text{ bar (approx.)}$$

The flow from A->T is the return flow out of the blind end. This flow rate is greater than the pump flow and must be determined by the following method:

(a) Calculate the piston area:

$$A_{\mathrm{P}} = \frac{\pi}{4}(D_{\mathrm{P}}^2) = \frac{\pi}{4}(7^2) = 38.5 \text{ cm}^2$$

(b) Calculate the rod area:

$$A_{\mathrm{R}} = \frac{\pi}{4}(D_{\mathrm{R}}^2) = \frac{\pi}{4}(3.125^2) = 7.7 \text{ cm}^2$$

(c) Calculate the return flow:

$$Q_{\text{return, R}} = \frac{Q_{\text{pump}}}{A_p - A_R} \times A_p$$

$$= \frac{Q_{\text{pump}}}{38.5 - 7.7} \times 38.5$$

$$= 1.25 \times 45$$

$$= 56.25\,\text{LPM}$$

The flow from A to T can now be read from the graph

Δp =6.2 (approx.)

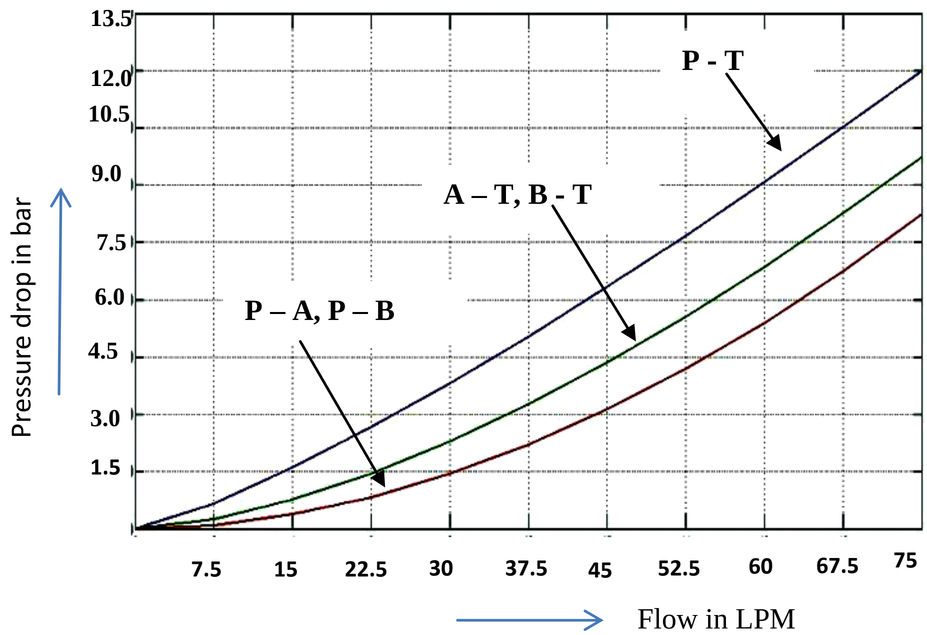


Figure 1.36

Example 2 Derive an expression to estimate leakage through the spool and housing bore for concentric leakage path. Refer Fig.1.37and the description of symbols.

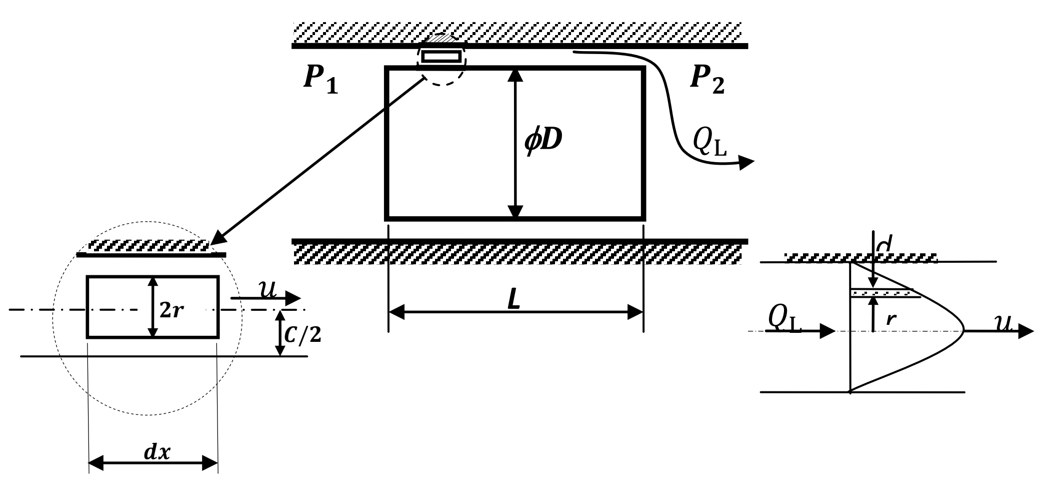


Figure 1.37

Solution

The typical pressure in the hydraulic system is in the order of 700–800 bar. The internal leakage is one of the major problems and it results from wear of the components. Figure 1.37shows the internal leakage through a radial clearance between two concentric cylindrical bodies, a spool and sleeve, for example.

Let

$a =$ Constant (m/s)

$c =$ Radial clearance (m)

$D =$ Spool diameter (m)

$F_p =$ Pressure force acting on the fluid element (N)

$F_r =$ Shear force acting on the fluid element (N)

$Q_L =$ Leakage flow rate (m^3/s)

$L =$ Length of the leakage path (m)

$c =$ Radial clearance from the midpoint of gap (m)

$R_L =$ Resistance to leakage (Ns/m^5)

$y =$ Distance between the element side surface and solid boundary (m)

$\Delta p =$ Pressure drop across the radial clearance (Pa)

Assuming the steady-state flow and forces at equilibrium, we can write the following:

The pressure force is $F_p = 2r\pi Ddp$

The frictional force is $F_r = 2\pi Ddx\tau$

Also,

$$r = 0.5c - y$$

$$\frac{du}{dy} = -\frac{du}{dr}$$

From Newton's law of viscosity,

$$\tau = \mu\frac{du}{dy} = -\mu\frac{du}{dr}$$

$$F_p = F_r$$

$$\mu\frac{du}{dr} = \frac{r\,dP}{dx} \text{ or } du = -\frac{r}{\mu}\frac{dP}{dx}dr$$

The pressure gradient $\dfrac{dP}{dx} = \text{constant}$

$$\frac{dP}{dx} = \frac{\Delta P}{L}, \text{ where } \Delta P = P_1 - P_2$$

The velocity distribution in the radial clearance is found by integrating

$$du = -\frac{r}{\mu}\frac{dP}{dx}dr$$

$$u = \int -\frac{r}{\mu}\frac{dP}{dx}dr + a = a - \frac{r^2}{2\mu}\frac{dP}{dx}$$

Applying boundary conditions,

$$u = \frac{1}{2\mu}\frac{\Delta P}{L}\left(\frac{c^2}{4} - r^2\right)$$

The leakage flow rate

$$Q_{\mathrm{L}} = \int_{\frac{c}{2}}^{\frac{c}{2}} \mu\pi D dr = \frac{\pi D c^3 \Delta P}{12\mu L}$$

$$\Delta P = \frac{12\mu L Q_{\mathrm{L}}}{\pi D c^3} = R_{\mathrm{L}} \times Q_{\mathrm{L}}$$

It is important to note that leakage is inversely proportional to the viscosity and directly proportional to the cube of the radial clearance. If the radial clearance is doubled due to wear, the internal leakage increases eight times. The power loss due to leakage is given by

$$\begin{aligned}\Delta N &= Q_{\mathrm{L}} \times \Delta P \\ &= \frac{\pi D c^3 \Delta P^2}{12\mu L} \\ &= \frac{\Delta P^2}{R_{\mathrm{L}}} = R_{\mathrm{L}} \times Q_{\mathrm{L}}{}^2\end{aligned}$$

The internal leakage reduces the effective flow rates and increases the power losses. The dissipated power is converted to heat and leads to serious oil overheating problems. Therefore, it is important to keep the oil viscosity within the predetermined limits over the whole operating temperature range. This is done by using hydraulic oils of convenient viscosity index and implementation of oil coolers.

Example 3 Write an expression to estimate leakage through spool and housing bore for eccentric leakage path. Refer Fig.1.38and the description of symbols. Compare with concentric leakage path and comment.

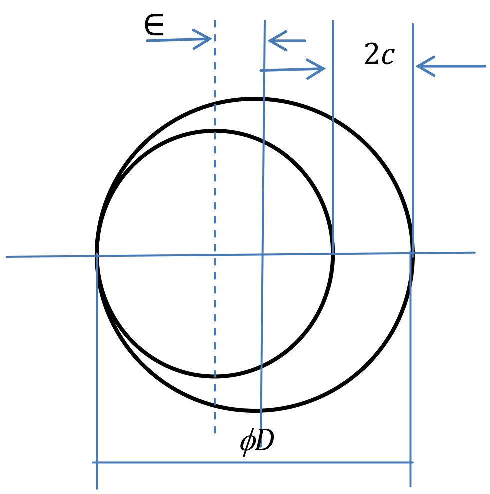


Figure 1.38

Solution

In the case of eccentric mounting, the radial clearance is not constant and the flow rate is given by

$$Q_L == \frac{\pi D c^3 \Delta P}{12\mu L}\left[1+\frac{3}{2}\left(\frac{\in}{c}\right)^3\right]$$

This is highly significant finding because if the inner cylinder just touches the outer cylinder, the flow rate is increased by 2.5 times the valve with the concentric cylinders assuming the same pressure drop.

Example 4 The land in a spool valve separates two fluid passages. The land has a 25mm length and 18.7325± 0.005mm diameter and operates in a 18.75± 0.01 mm bore. We assume that the spool is concentric. The pressure difference across the land is 20.68 MPa. Calculate the leakage flow rate past this land for minimum, nominal and maximum leakage conditions assuming a fluid with minimum, nominal and maximum viscosities of 5.84, 32.1 and 800 mm^2/s.

Solution

The minimum height of the passage is achieved for the maximum diameter of the spool and the minimum diameter of the bore

$$c_{min} = \frac{(18.75-0.01)-(18.7325+0.005)}{2} = 0.00125\ \text{mm}$$

Thenominal height of the passage is achieved when both the spool and bore have their nominal dimensions

$$c_{nom} = \frac{(18.75)-(18.7325)}{2} = 0.00875\,\text{mm}$$

Themaximum height of the passage is achieved for the minimum diameter of the spool and the maximum diameter of the bore

$$c_{max} = \frac{(18.75+0.01)-(18.7325-0.005)}{2} = 0.01625\ \text{mm}$$

Conversion of viscosity

cSt (mm^2/s)	cP (SG = 0.9)	Ns/m^2 Pascal seconds
5.84	5.256	0.005256
32.1	28.89	0.02889
880	792	0.792

Case 1: Minimum height of the passage

$$Q_{min} = \frac{\pi D c^3 \Delta P}{12\mu L}$$
$$= \frac{\pi(0.01875)\ (0.00125\times10^{-3})^3 \times 20.68\times10^6}{12\times0.005256\times0.0025}$$
$$=1.5088\times10^{-8}\ \ m^3/s$$

Case 2: Nominal height of the passage

$$Q_{nom} = \frac{\pi D c^3 \Delta P}{12\mu L}$$
$$= \frac{\pi(0.01875)\ (0.00875\times10^{-3})^3 \times 20.68\times10^6}{12\times0.02889\times0.0025}$$
$$=9.4146\times10^{-7}\ \ m^3/s$$

Case 3: Maximum height of the passage

$$Q_{\max} = \frac{\pi D c^3 \Delta P}{12 \mu L}$$

$$= \frac{\pi(0.01875)\,(0.01625\times10^{-3})^3\times 20.68\times10^6}{12\times0.792\times0.0025}$$

$$= 2.199\times10^{-7}\ \text{m}^3/\text{s}$$

Review Questions

1. Explain briefly the function of DCVs.
2. Draw a schematic of 4/3 DCV that is direct operated electrically and briefly explain its function.
3. Draw a schematic of 3/2 DCV that is manually operated and briefly explain its function.
4. State the different ways of control of DCVs.
5. How are DCVs classified?
6. Explain the construction and operation of electric solenoids and compare the DC and AC solenoids.
7. Cite the classification of check valves and explain the function of pilot-operated check valve, giving the necessary drawing.
8. What is the difference between an open-center and closed-center type of DCV?
9. What is a shuttle valve? Name one application.

Objective-Type Questions
Fill in the Blanks

1. A valve is a device that receives an external signal to release, ------ the fluid that flows through it.
2. DCVs determine-----through which a fluid transverses a given circuit.
3. A check valve allows flow in --------, but blocks the flow in the opposite direction.
4. In 4/3 DCV,for-------- the pump line is blocked so that the flow must pass over the pressure relief valve the pressure is at the system maximum.
5. In 4/3 DCV, for float neutral,the ------- is blocked and the outlet is connected to the tank.
6. In 4/3 DCV, for open neutral, the pressure port and the outlets are both connected to the ----.

State True or False

1. Pressure control valves protectthe system against overpressure, which may occur due to a sudden surge.
2. A pilot-operated check valve always permits flow in one direction only.
3. A shuttle valve allows two alternate flow sources to be connected in a one-branch circuit.
4. In 4/3 DCV, tandem neutral, the pump flow is allowed to flow to the system.
5. The purpose of the regenerative neutral is that instead of sending the return flow back to the tank, it sends it into the inlet side of the cylinder, thereby decreasing its speed.

Answers
Fill in the Blanks

1.Stop or redirect
2.Path
3.Only one direction
4.Closed neutral
5.Pressure port
6. Tank

State True or False

1.True
2.False
3.True
4.False
5.False

Lecture 18

PRESSURE-CONTROL VALVES

Learning Objectives

Upon completion of this chapter, the student should be able to:

- Explain various functions of pressure-control valves.
- Explain various classifications of pressure-control valves.
- Describe the working construction of various pressure-control valves.
- Differentiate between a pressure relief valve, a pressure-reducing valve, a sequence valve and an unloading valve.
- Identify the graphic symbols for various types of pressure-control valves.
- Explain different applications of pressure-control valves.
- Explain the working principle of solenoid-actuated valves.
- Calculate the pressure loss in pressure relief and unloading valves.
- Evaluate the performance of hydraulic systems using pressure-control valves.

1.1 Introduction

Hydraulic energy is produced as long as the prime mover (usually an electric motor) drives the pump, and hydraulic pressure develops by resistance to pump flow.Hence, the hydraulic system suffers damage if the pump flow is not stopped or off loaded (recirculate) back to the tank during non-action periods of the circuit.Non-action periods arise from stalling an actuator, or by reaching the end of the stroke or the circuit sequence, or during the time-delay periods of the circuit sequence.

In order to avoid hydraulic system damage, power wastage and overheating of the hydraulic fluid, circuit designers use a variety of cleverly designed systems to control maximum system pressure and pump flow during non-action periods.
Pressure-control valves are used in hydraulic systems to control actuator force (force = pressure × area) and to determine and select pressure levels at which certain machine operations must occur.Pressure controls are mainly used to perform the following system functions:

- Limiting maximum system pressure at a safe level.
- Regulating/reducing pressure in certain portions of the circuit.
- Unloading system pressure.
- Assisting sequential operation of actuators in a circuit with pressure control.
- Any other pressure-related function by virtue of pressure control.
- Reducing or stepping down pressure levels from the main circuit to a lower pressure in a sub-circuit.

Pressure-control valves are often difficult to identify mainly because of the many descriptive names given to them. The function of the valve in the circuit usually becomes the basis for its name. The valves used for accomplishing the above-mentioned system functions are therefore given the following names:

- Pressure-relief valve.
- Pressure-reducing valve.
- Unloading valve
- Counterbalance valve.

- Pressure-sequence valve.
- Brake valve.

1.2 Pressure-Relief Valves

Pressure-relief valves limit the maximum pressure in a hydraulic circuit by providing an alternate path for fluid flow when the pressure reaches a preset level. All fixed-volume pump circuits require a relief valve to protect the system from excess pressure. Fixed-volume pumps must move fluid when they turn. When a pump unloads through an open-center circuit or actuators are in motion, fluid movement is not a problem. A relief valve is essential when the actuators stall with the directional valve still in shifted position.

A relief valve is similar to a fuse in an electrical system. When circuit amperage stays below the fuse amperage, all is well. When circuit amperage tries to exceed fuse amperage, the fuse blows and disables the circuit. Both devices protect the system from excess pressure/current by keeping it below a preset level. The difference is that when an electrical fuse blows, it must be reset or replaced by maintenance personnel before the machine cycles again. This requirement alerts electrician'sabouta possible problem before restarting the machine. Without the protection of a fuse, the electrical circuit would finally overheat and start a fire.

Similarly, in a hydraulic circuit, a relief valve opens and bypasses fluid when pressure exceeds its setting. The valve then closes again when pressure falls. This means that a relief valve can bypass fluid anytime, or all the time, without intervention by maintenance. Many fixed-volume pump circuits depend on this bypassing capability during the cycle, and some even bypass fluid during idle time. A well-designed circuit never bypasses fluid unless there is a malfunction, such as a limit switch not closing or an operator over-riding the controls. This eliminates most overheating problems and saves energy.
There are two different designs of relief valves in use: direct-acting and pilot-operated. Both types have advantages and work better in certain applications.

1.2.1 Simple Pressure-Relief Valve

The most widely used type of pressure control valve is the pressure-relief valve because it is found in practically every hydraulic system. Schematic diagram of simple relief valve is shown in Fig. 1.1 and three-dimensional view is shown in Fig. 1.2. It is normally a closed valve whose function is to limit the pressure to a specified maximum value by diverting pump flow back to the tank. A poppet is held seated inside the valve by a heavy spring. When the system pressure reaches a high enough value, the poppet is forced off its seat. This permits flow through the outlet to the tank as long as this high pressure level is maintained. Note the external adjusting screw, which varies spring force and, thus, the pressure at which the valve begins to open (cracking pressure)(Fig. 1.3).
It should be noted that the poppet must open sufficiently to allow full pump flow. The pressure that exists at full pump flow can be substantially greater than cracking pressure. The pressure at full pump flow is the pressure level that is specified when referring to the pressure setting of the valve. It is the maximum pressure level permitted by the relief valve.

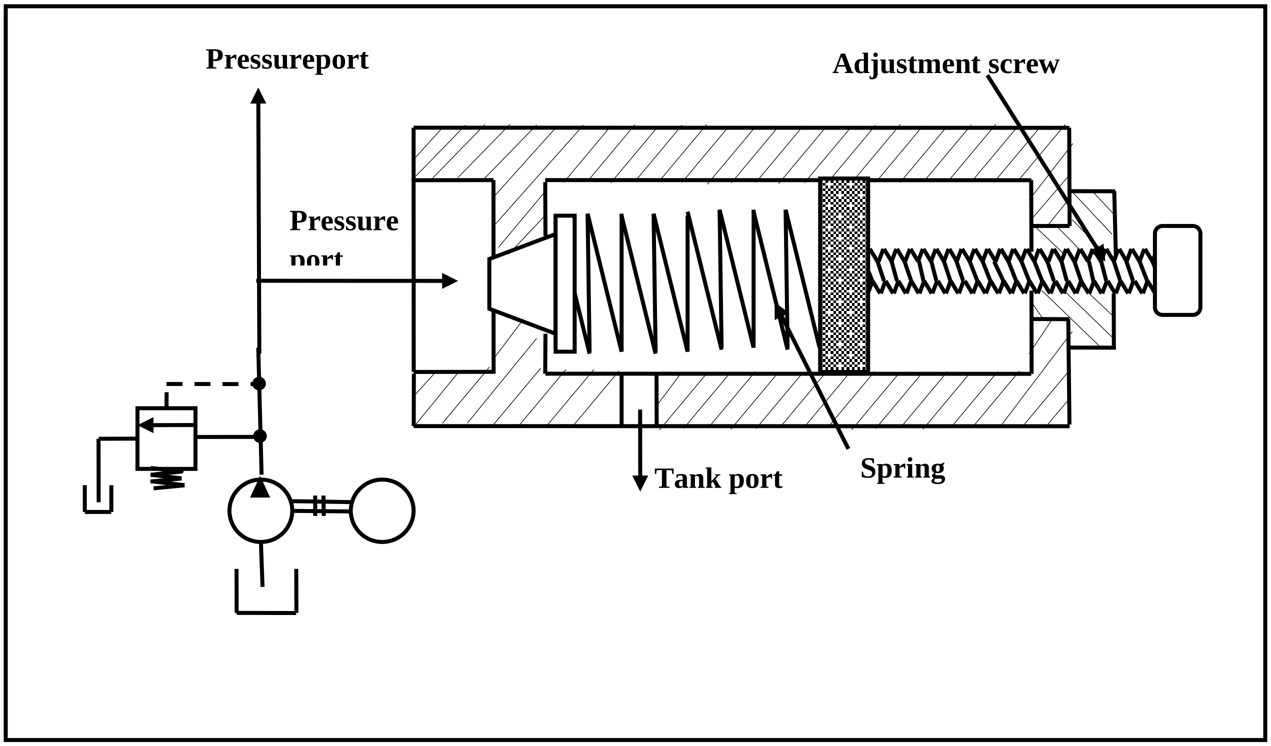


Figure1.1 Simple pressure-relief valve.

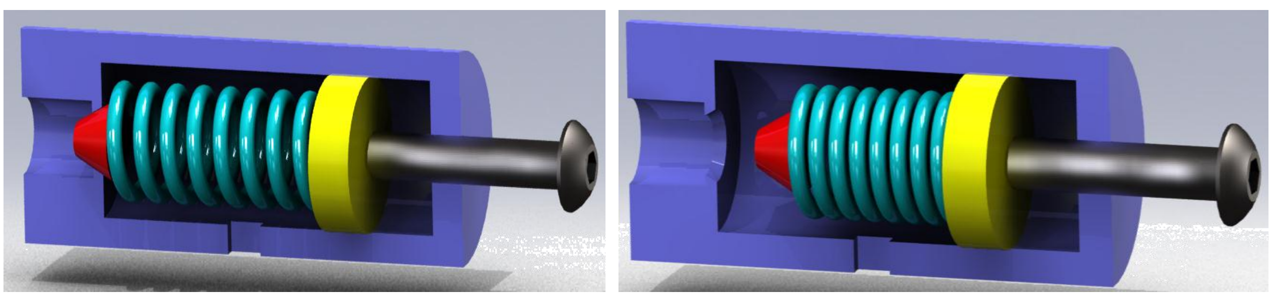


Figure 1.2 Three-dimensional view of simple pressure-relief valve.

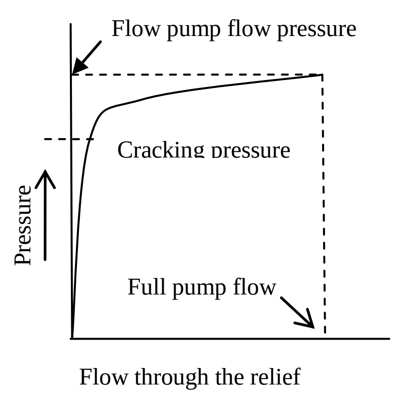


Figure 1.3 Characteristics of a relief valve.

If the hydraulic system does not accept any flow, then all the pump flow must return to the tank via the relief valve. The pressure-relief valve provides protection against any overloads experienced by the actuators in the hydraulic system. Of course, a relief valve is not needed if a pressure-compensated vane pump is used. Obviously one important function of a pressure-relief valve is to limit the force or torque produced by hydraulic cylinders or motors.
The main advantage of direct-acting relief valves over pilot-operated relief valves is that they respond very rapidly to pressure buildup. Because there is only one moving part in a direct-acting relief valve, it can open rapidly, thus minimizing pressure spikes.

1.2.2 Compound Pressure Relief Valve(Pilot-Operated Pressure Relief Valve)
A pilot-operated pressure-relief valve consists of a small pilot relief valve and main relief valve as shown in Fig. 1.4. It operates in a two-stage process:

1. The pilot relief valve opens when a preset maximum pressure is reached.
2. When the pilot relief valve opens, it makes the main relief valve open.

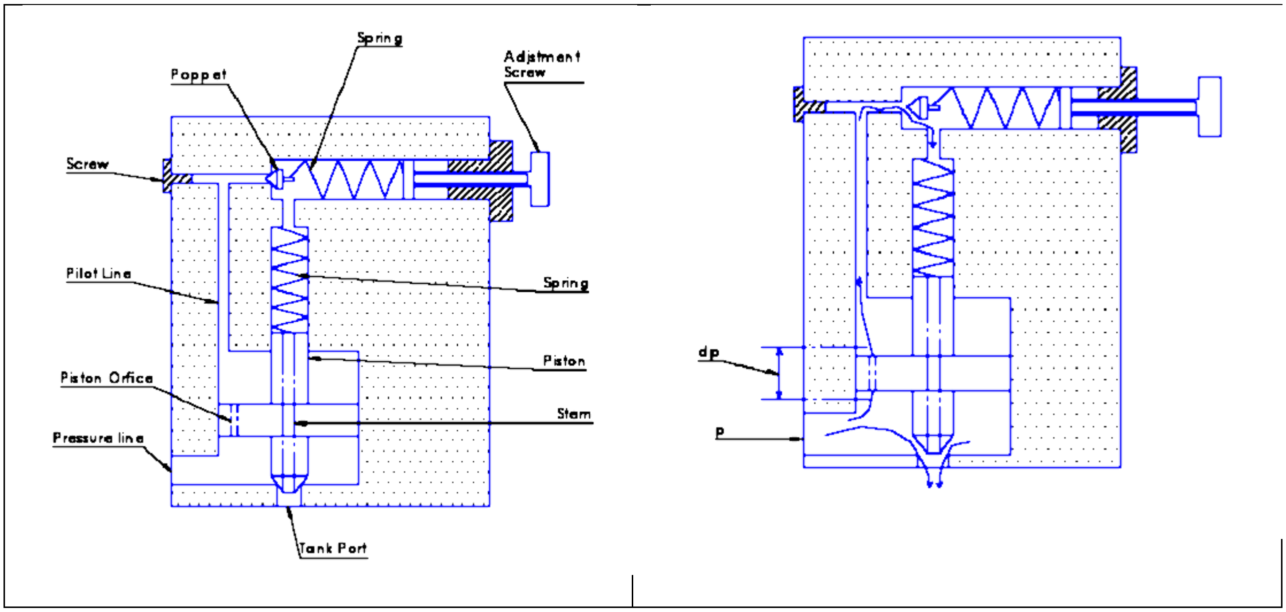


Figure 1.4 Compound relief valve.
The pilot-operated pressure-relief valve has a pressure port that is connected to the pump line and the tank port is connected to the tank. The pilot relief valve is a poppet type. The main relief valve consists of a piston and a stem. The main relief piston has an orifice drilled through it. The piston has equal areas exposed to pressure on top and bottom and is in a balanced condition due to equal force acting on both the sides. It remains stationary in the closed position. The piston has a light bias spring to ensure that it stays closed. When the pressure is less than that of relief valve setting, the pump flow goes to the system. If the pressure in the system becomes high enough, it moves the pilot poppet off its seat. A small amount of flow begins to go through the pilot line back to the tank. Once flow begins through the piston orifice and pilot line, a pressure drop is induced across the piston due to the restriction of the piston orifice. This pressure drop then causes the piston and stem to lift off their seats and the flow goes directly from the pressure port to the tank.
The advantages of pilot-operated pressure-relief valves over direct-acting pressure-relief valves are as follows:

1. Pilot-operated pressure-relief valves are usually smaller than direct-acting pressure-relief valves for the same flow and pressure settings.
2. They have a wider range for the maximum pressure settings than direct-acting pressure-relief valves.
3. They can be operated using a remote while direct-acting pressure-relief valves cannot.

Graphic symbol of a pressure-relief valve is shown in Fig. 1.5. The symbol shows that the valve is normally closed (the arrow is offline). On one side of the valve, pressure is fed in (the dashed line) to try to open the valve, while on the other side, the spring tries to keep it adjustable, allowing the adjustment of pressure level at which the relief valve opens. The arrow through the spring signifies that it is adjustable, allowing the adjustment of pressure level at which the relief valve opens.

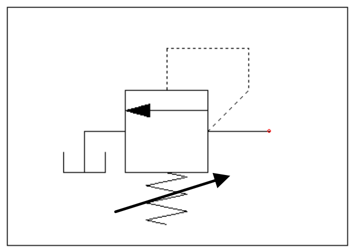


Figure 1.5 Symbolic representation of a simple pressure-relief valve.

Example 1.1

A pressure-relief valve has a pressure setting of 140 bar. Compute the kW loss across this valve if it returns all the flow back to the tank from a 0.0016 m^3/s pump.

Solution: We have

$$\begin{aligned}\text{kW power} &= pQ \\ &= (140\times10^5)\times(0.0016\times10^{-3}) \\ &= 22.4\ \text{kW}\end{aligned}$$

Example 1.2

A pressure-relief valve contains a poppet with an area of 4.2 cm^2 on which the system pressure acts. During assembly, a spring with a spring constant of 3300 N/cm is installed in the valve to hold the poppet against its seat. The adjustment mechanism is then set so that the spring is initially compressed to 0.5 cm from its free-length condition. In order to pass full pump flow through the valve at the pressure-relief valve pressure setting, the poppet must move 0.30 cm from its fully closed position.

(a) Determine the cracking pressure.

(b) Determine the full pump flow pressure (pressure-relief valve pressure setting).

(c) What should be the initial compression of the spring in pressure-relief valve if the full pump flow is to be 40% greater than the cracking pressure?

Solution:
(a) Cracking pressure:

Force required to fully close is the product of initial displacement and spring constant

$$F_{\text{valve closed}} = K\,S_{\text{initial}} = 3200\ \text{N/cm} \times 0.50\ \text{cm} = 1600\ \text{N}$$

Now we can calculate the cracking pressure knowing the cracking force

$$p_{\text{cracking}} A_{\text{poppet}} = 1600\ \text{N}$$
$$\Rightarrow p_{\text{cracking}} (4.20 \times 10^{-4}\ \text{m}^2) = 1600\ \text{N}$$
$$\Rightarrow p_{\text{cracking}} = 381 \times 10^4\ \text{N/m}^2 = 3.81\ \text{MPa}$$

(b) Full pump flow pressure (pressure-relief valve pressure setting):

Force required to fully open is the product of final displacement and spring constant

$$F_{\text{fully open}} = K\,S_{\text{fully open}} = 3200\ \text{N/cm} \times 0.8\ \text{cm} = 2560\ \text{N}$$

Now this force must be equal to product of full pump pressure and area of poppet.

$$p_{\text{full pump flow}} A_{\text{poppet}} = 2650\ \text{N}$$
$$\Rightarrow p_{\text{full pump flow}} (4.20 \times 10^{-4}\ \text{m}^2) = 2650\ \text{N}$$
$$\Rightarrow p_{\text{full pump flow}} = 610 \times 10^4\ \text{N/m}^2 = 6.10\ \text{MPa}$$

(c) Initial compression of spring:

$$F_{\text{valve closed}} = K\,l = 3200l = p_{\text{cracking}} A_{\text{poppet}}$$

Now cracking pressure can be calculated as follows

$$p_{\text{cracking}} = \frac{\text{Force}}{\text{Area}} = \frac{3200\,l}{4.20 \times 10^{-4}} = 762 \times 10^4\, l$$

Also we know that force required to fully open is given by product of full pump flow and area of poppet.

$$\begin{aligned} F_{\text{fully open}} &= K\ (l + 0.3) \\ &= 3200(l + 0.3) \\ &= 3200l + 960 \\ &= p_{\text{full pump flow}} A_{\text{poppet}} \end{aligned}$$

Now

$$p_{\text{full pump flow}} = \frac{3200l + 960}{4.20 \times 10^{-4}} = (762l + 229)10^4$$

We can now calculate the ratio of pump full flow pressure to cracking pressure as

$$\frac{p_{\text{full pump flow}}}{p_{\text{cracking}}} = \frac{(762l + 229)10^4}{(762l)10^4} = 1.40$$

Solving we get l = 0.75 cm.

Example 1.3

A pressure-relief valve has a pressure setting of 200 bar. Determine the power loss across the valve if all the pump flow of 120 L/min flows back to the reservoir through this valve.

Solution:

Pump flow Q = 120 L/min = 2 L/s = 0.002 m^3/s

Pressure setting of the valve = 200 bar = 200 × 10^5 N/m^2

Therefore, the power loss across the pressure-relief valve is

$$\frac{200\times10^5\times 0.002}{1000} = 40\text{ kW}$$

Example 1.4

A pressure-relief valve contains a poppet with a 3.87 cm^2 area on which the system pressure acts. The poppet must move 0.381 cm from its fully closed position in order to pass pump flow at the pressure-relief valve setting (full pump flow pressure). The pressure required to overcome the external load is 68.95 bar. Assume that the pressure-relief valve setting is 50% higher than the pressure required to overcome the external load. If the valve-cracking pressure is 10% higher than the pressure required to overcome the external load, find the following:

(a) The required spring constant of the compression in the valve.

(b) The required initial compression of the spring from its free length condition as set by the spring adjustment mechanism of the pressure-relief valve.

Solution:

a) At full pump flow pressure, spring force equals hydraulic force on the poppet: Total spring compression (S) = Initial compression (L) + Full poppet stroke

$$\Rightarrow k(L+0.00381)=4002.5\text{ N}$$

$$\Rightarrow kL+0.00381\,k=4002.5\text{ N}$$

Also at cracking pressure, spring force equals hydraulic force on the poppet. Thus, we have

Spring force= Cracking force

$$kL = 0.00381\,k$$

$$=1.1\times68.95\times10^5\text{ N/m}^2\times3.87\times10^{-4}\text{ m}^2$$

$$=2935.8\text{ N}$$

Substituting values of $k$in $kL+0.00381\,k=4002.5\text{ N}$, we get

$$2935.8+0.00381\,k=4002.5\text{ N}$$

$$\Rightarrow k=279986.22\text{ N/m}$$

(b) From part (a), we have

$$k=279986.22\text{ N/m}$$

$$kL=2935.8\text{ N}$$

This implies

$$279986.22\times L=2935.8\text{ N}$$

$$\Rightarrow \quad L=\frac{2935.8}{279986.22}\text{ m}=0.0104\text{ m}=1.04\text{ cm}=10.4\text{ mm}$$

1.3 Pressure-Reducing Valve

The second type of valve is a pressure-reducing valve. This type of valve (which is normally open) is used to maintain reduced pressures in specified locations of hydraulic systems. It is actuated by downstream pressure and tends to close as this pressure reaches the valve setting. Schematic diagram of pressure reducing valve is shown in Fig. 1.6, symbolic representation is shown in Fig. 1.7 and three-dimensional view is shown in Fig. 1.8.

A pressure-reducing valve uses a spring-loaded spool to control the downstream pressure. If the downstream pressure is below the valve setting, the fluid flows freely from the inlet to the outlet. Note that there is an internal passageway from the outlet which transmits outlet pressure to the spool end opposite the spring. When the outlet (downstream) pressure increases to the valve setting, the spool moves to the right to partially block the outlet port. Just enough flow is passed to the outlet to maintain its preset pressure level. If the valve closes completely, leakage past the spool causes downstream pressure to build up above the valve setting. This is prevented from occurring because a continuous bleed to the tank is permitted via a separate drain line to the tank.

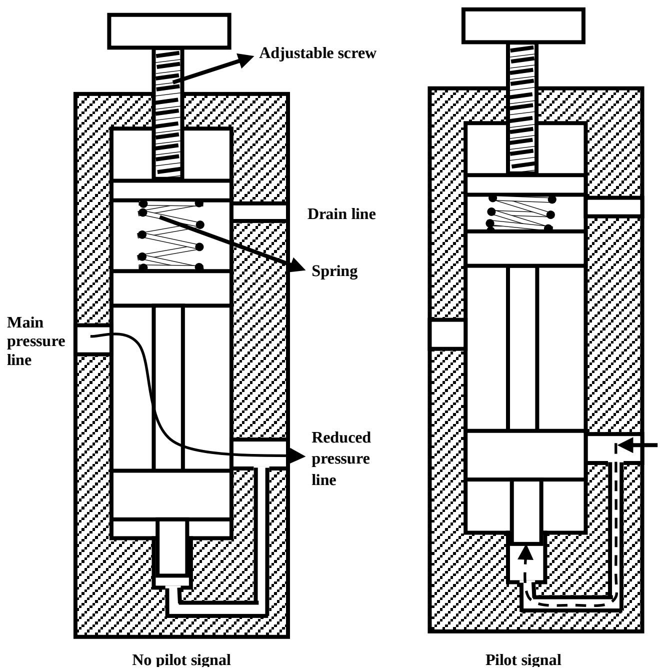


Figure 1.6 Pressure-reducing valve.

Reverse free flow through the valve is only possible if the pressure exceeds the valve setting.The valve then closes, thus making reverse flow impossible. Therefore, pressure-reducing valves are often equipped with a check valve for reverse free flow.

External forces acting onto a linear actuator increase the pressure between the pressure-reducing valve and the actuator. In some systems, it is therefore desirable to relieve excess fluid from the secondary system to the tank in order to maintain a constant downstream pressure, regardless of such external forces.

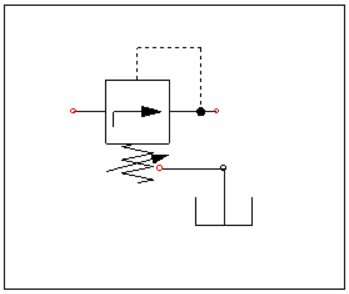


Figure1.7 Symbolic representation of a pressure-reducing valve.

A reducing valve is normally open. It reads the downstream pressure. It has an externaldrain.This is represented by a line connected from the valve drain port to the tank. The symbol shows that the spring cavity has a drain to the tank.

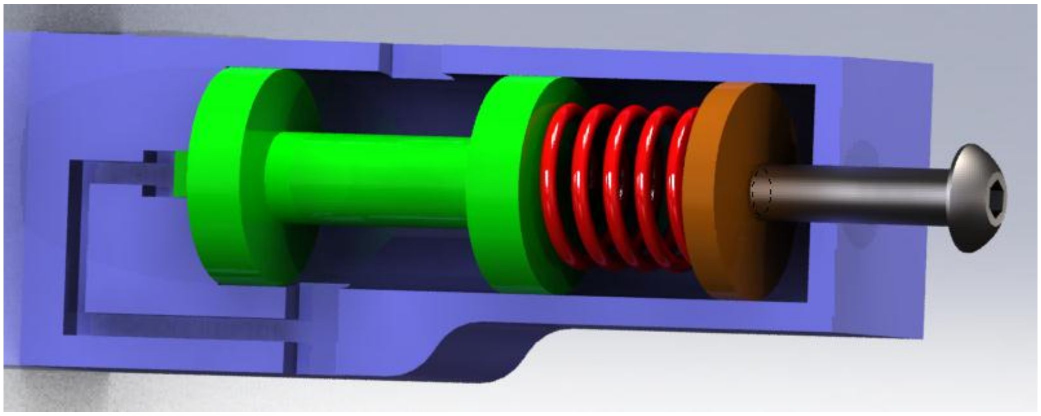


Figure 1.8 Three-dimensional view of a pressure-reducing valve.

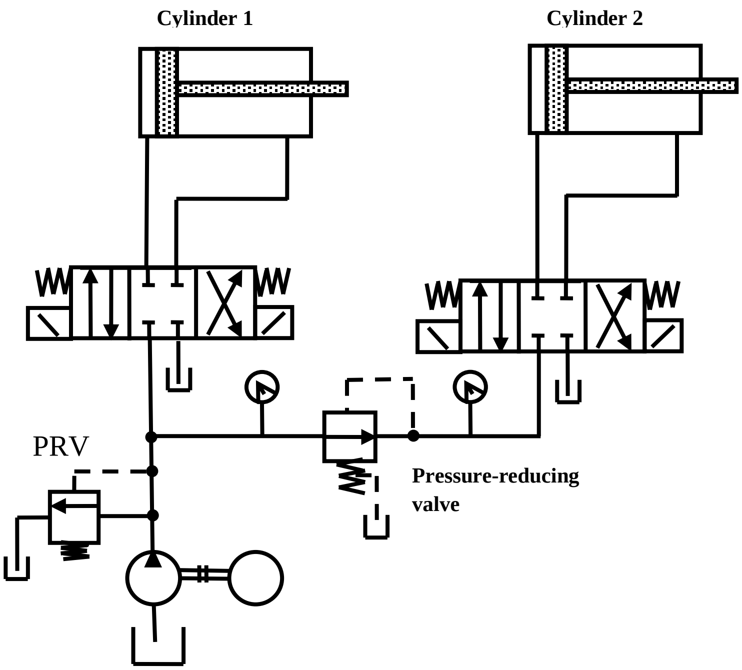


Figure 1.9 Application of a pressure-reducing valve.

Figure1.9shows an application for a pressure-reducing valve. Here two cylinders are connected in parallel. The circuit is designed to operate at a maximum pressure p_1, which is determined by the relief valve setting. This is the maximum pressure at which cylinder 1 operates. By the function of the machine, cylinder2 is limited to pressurep_2 ($p_2<p_1$).This can be accomplished by placing a pressure-reducing valve in the circuit in the location shown in Fig. 1.9.If the pressure in the cylinder2 circuit rises above p_2, the pressure-reducing valve closes partially to create a pressure drop across the valve. The valve then maintains the pressure drop so that the outlet pressure is not allowed to rise above p_2 setting.
The disadvantage of this method is that the pressure drop across the reducing valve represents the lost energy that is being converted into heat. If the pressure setting of the reducing valve is set very low relative to the pressure in the rest of the system, the pressure drop is very high, resulting in excessive heating of the fluid. When the hydraulic oil becomes too hot, its viscosity reduces causing increased component wear.

Example 1.5

The primary part of a circuit is operating at 180 bar. A secondary circuit supplied from the primary circuit via a pressure-reducing valve requires a constant flow of 30 L/min at 100 bar. Find the power loss over pressure-relief valve. Comment on the type of cooling.

Solution: The power loss over the pressure-reducing valve is

$$\frac{(180-100)\times 30}{600} = 4 \text{ kW}$$

4 kW heat cannot be dissipated by natural cooling and heat exchanger may be required. In practice, the cost of fitting a heat exchanger and operating costs should be weighed against alternative circuitry such as a two-pump system.

1.4Unloading Valves

Unloading valves are pressure-control devices that are used to dump excess fluid to the tank at little or no pressure. A common application is in high-low pump circuits where two pumps move an actuator at a high speed and low pressure.The circuit then shifts to a single pump providing a high pressure to perform work.
Another application is sending excess flow from the cap end of an oversize-rod cylinder to the tank as the cylinder retracts. This makes it possible to use a smaller, less-expensive directional control valve while keeping pressure drop low.

1.4.1 Direct-Acting Unloading Valve

A direct-acting unloading valve consists of a spool held in the closed position by a spring. The spool blocks flow from the inlet to the tank port under normal conditions. When a high-pressure fluid from the pump enters at the external-pilot port, it exerts force against the pilot piston. (The small-diameter pilot piston allows the use of a long, low-force spring.) When the system pressure increases to the spring setting, the fluid bypasses to the tank (as a relief valve would function). When the pressure goes above the spring setting, the spool opens fully to dump the excess fluid to the tank at little or no pressure.

1.4.2 Pilot-Operated Unloading Valve

A pilot-operated unloading valve has less pressure override than its direct-acting counterpart.So it does not dump part of the flow prematurely.

A pilot-operated unloading relief valve is the same as a pilot-operated relief valve with the addition of an unloading spool. Without the unloading spool, this valve would function just like any pilot-operated relief valve. Pressure buildup in the pilot section would open some flow to the tank and unbalance the poppet, allowing it to open and relieve excess pump flow.

Schematic diagram of unloading valve is shown in Fig. 1.10.In a pilot-operated unloading valve; the unloading spool receives a signal through the remote-pilot port when pressure in the working circuit goes above its setting. At the same time, pressure on the spring-loaded ball in the pilot section starts to open it. Pressure drop on the front side of the unloading spool lowers back force and pilot pressure from the high-pressure circuit forces the spring-loaded ball completely off its seat. Now there is more flow going to the tank than what the control orifice can keep up with. The main poppet opens at approximately 20 psi. Now, all high-volume pump flow can go to the tank at little or no pressure drop and all horsepower can go to the low-volume pump to do the work. When pressure falls approximately 15% below the pressure set in the pilot section, the spring-loaded ball closes and pushes the unloading spool back for the next cycle.

An unloading valve requires no electric signals. This eliminates the need for extra persons when troubleshooting. These valves are very reliable and seldom require maintenance, adjustment or replacement.An unloading valve unloads the pump when the desired pressure is reached.It allows rapid discharge of pressurized oil near atmospheric pressure.As soon as the system pressure reaches the setting pressure that is available at the pilot port, it lifts the spool against the spring force.When the spool is held by the pilot pressure, the delivery from the pump goes to the tank.An unloading valve is used to perform operations such as stamping, coining, punching, piercing, etc.

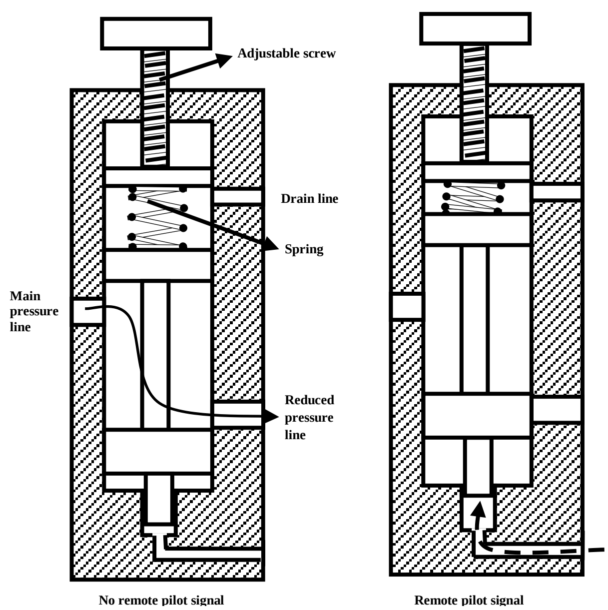


Figure 1.10 Unloading valve.

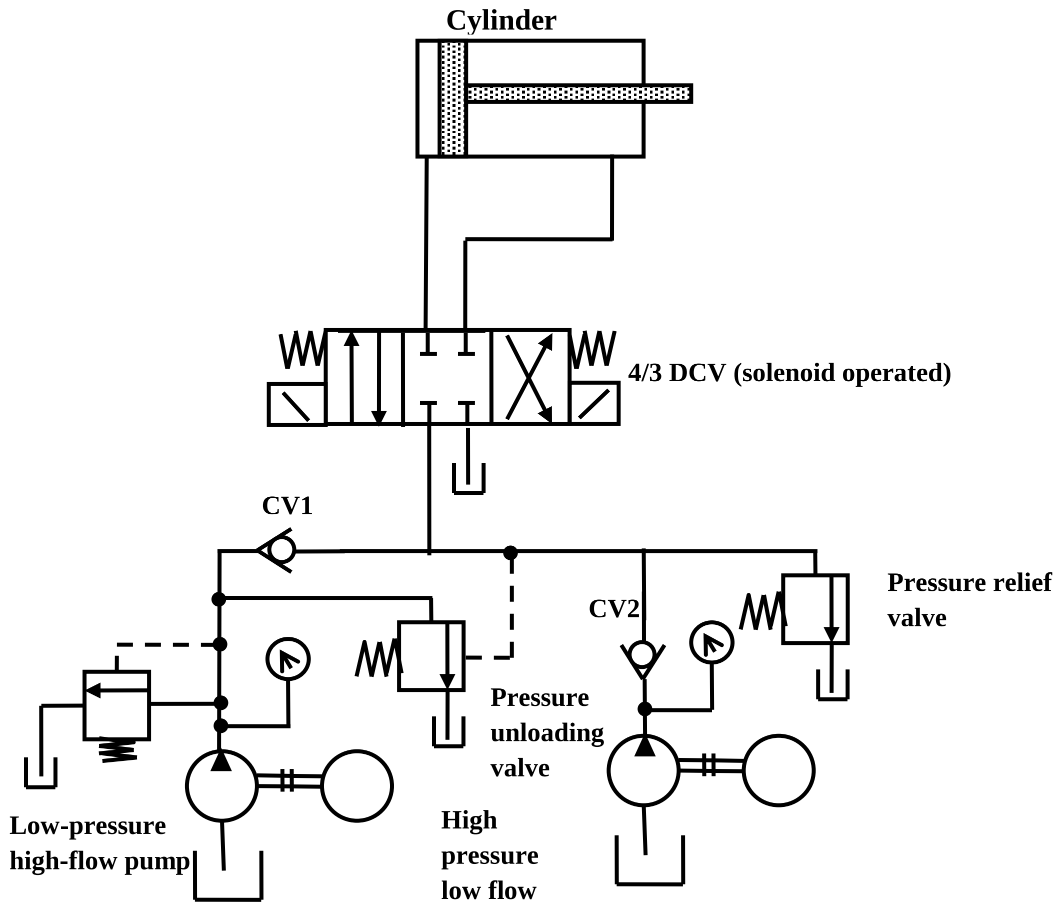


Figure1.11 Application of unloading valve in a punching press (high–low circuit).

Figure1.11shows the application of unloading valve in a punching press. It is a circuit that uses a high-pressure, low-flow pump in conjunction with a low-pressure, high-flow pump. In a punching press, the hydraulic cylinder must extend rapidly over a great distance with low-pressure, high-flow requirements. This rapid extension of cylinder occurs under no external load (when the punching tool approaches the sheet metal).But during punching operation for short motion, the pressure requirements are high due to punching load. During this cylinder travel, high-pressure, low-flow requirements are needed.

When punching operation begins, the increased pressure opens the unloading valve to unload the low-pressure pump. The purpose of relief is to protect the high-pressure pump from over pressure at the end of the cylinder stroke and when direction control valve (DCV) is in its spring centered mode. The check valve protects the low-pressure pump from high pressure, which occurs during punching operation that occurs at the end of cylinder extension and when the DCV is in its spring centered mode.
The above circuit given in Fig. 1.11 eliminates the necessity of having a very expensive high-pressure, high-flow pump.

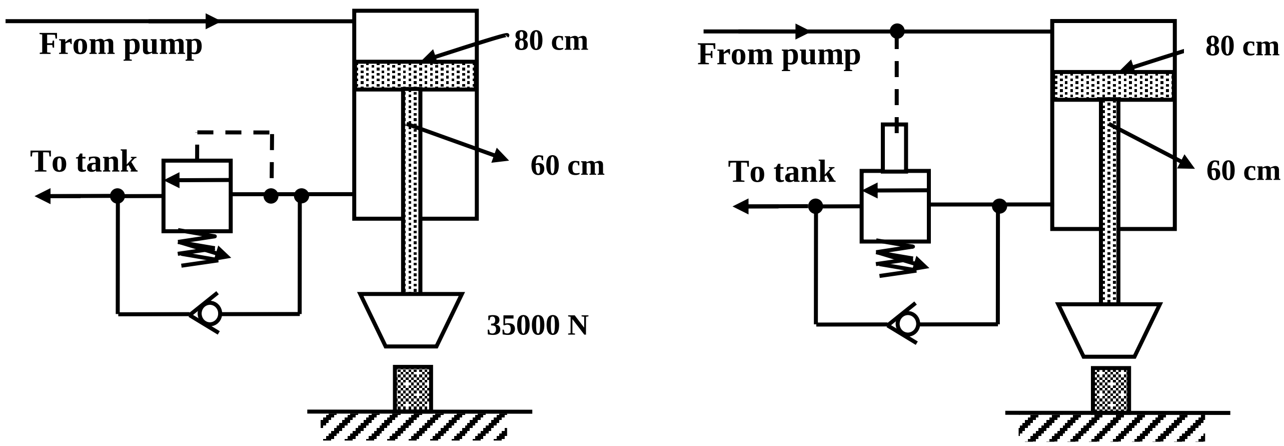
Example 1.6

Consider a 100 kN press as shown in Fig. 1.22. Weight of the tool = 5 kN. Cylinder bore =80 mm. Cylinder rod = 60 mm. (a) Find the pressure at annulus side to balance tools.

(b) Find the pressure to achieve 100 kN pressing force.

(c) If an over-center valve is used instead of a counterbalance valve, what is the pressure to achieve 100 kN pressing force. Comment on the results obtained.

In part (c), use an over-center valve with a 2:1 pilot input ratio set at 23 bar to balance the tools instead of the counterbalance valve.



With counterbalance valve **With over-center valve**

Figure 1.22

Solution:

Full bore area A_p = 0.082 × π/4 = 0.005 m^2

Annulus area A_A = (0.0082−0.0062) × π/4 = 0.0028 m^2

Case 1: Using counterbalance valve:

Pressure at annulus side to balance tools = $\dfrac{5\times10^3}{0.0028}\times10^{-5}$ = 17.8 bar

Suggested counterbalance valve setting = 17.8 × 1.3 = 23 bar

Pressure at the full bore side of cylinder to overcome counterbalance = 23 × 0.0028/0.005 = 13 bar

Pressure to achieve 100 kN pressing force = $\dfrac{10010^3\times10^{-5}}{0.005}$ +13 = 213 bar

Case 2: Using over-center valve (brake valve): Let us use an over-center valve with a 2:1 pilot input ratio set at 23 bar to balance the tools instead of the counterbalance valve.

Pressure on the pilot required to open the valve = 23/2 = 11.5 bar

Pressure at the full bore side to drive down the tooling = 11.5 bar

Pressure required to achieve 100 kN pressing force is

$$\frac{(100-5)\times10^3\times10^{-5}}{0.005} = 190 \text{ bar}$$

This is greater than 11.5 bar pressure needed to pilot the over-center valve open. Therefore, there will be no back-pressure set up on the annulus side of the piston during the pressing operation.

Example 1.7

(a) An unloading valve is used to unload the 0.0016 m^3/s pump. If the pump discharge pressure during unloading equals 2 bar, how much hydraulic kW power is being wasted?

(b) In the counterbalance circuit shown in Fig. 1.13, load to be lifted is 10 kN and a cylinder bore area of 0.002 m^2 (equivalent to 50 mm diameter) is used. Find the counterbalance valve setting.

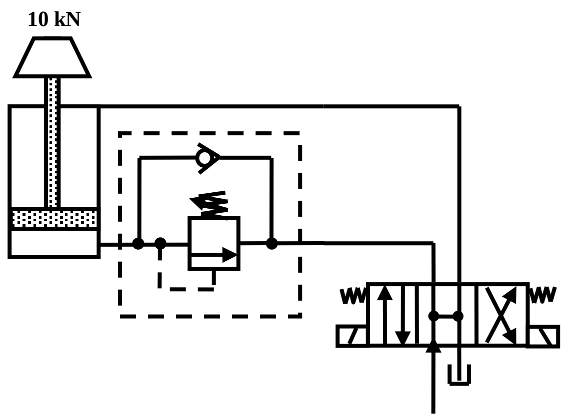


Figure 1.13

Solution:

(a) We have

$$\begin{aligned} \text{kW power} &= pQ \\ &= (2\times10^5)\times(0.0016\times10^{-3}) \\ &= 0.32\ \text{kW} \end{aligned}$$

(b) We have

$$\text{Load-induced pressure} = \frac{10\times10^3}{0.002\times10^5} = 50\ \text{bar}$$

The counterbalance valve setting should be 50 × 1.3 = 65 bar

Lecture 19

PRESSURE-CONTROL VALVES [CONTINUED]

1.5 Counterbalance Valve

Schematic diagram of counterbalance valve is shown in Fig. 1.14. These normally closed valves are primarily used to maintain a back pressure on a vertical cylinder to prevent it from falling due to gravity. They are used to prevent a load from accelerating uncontrollably. This situation can occur in vertical cylinders in which the load is a weight. This can damage the load or even the cylinder itself when the load is stopped quickly at the end of the travel.

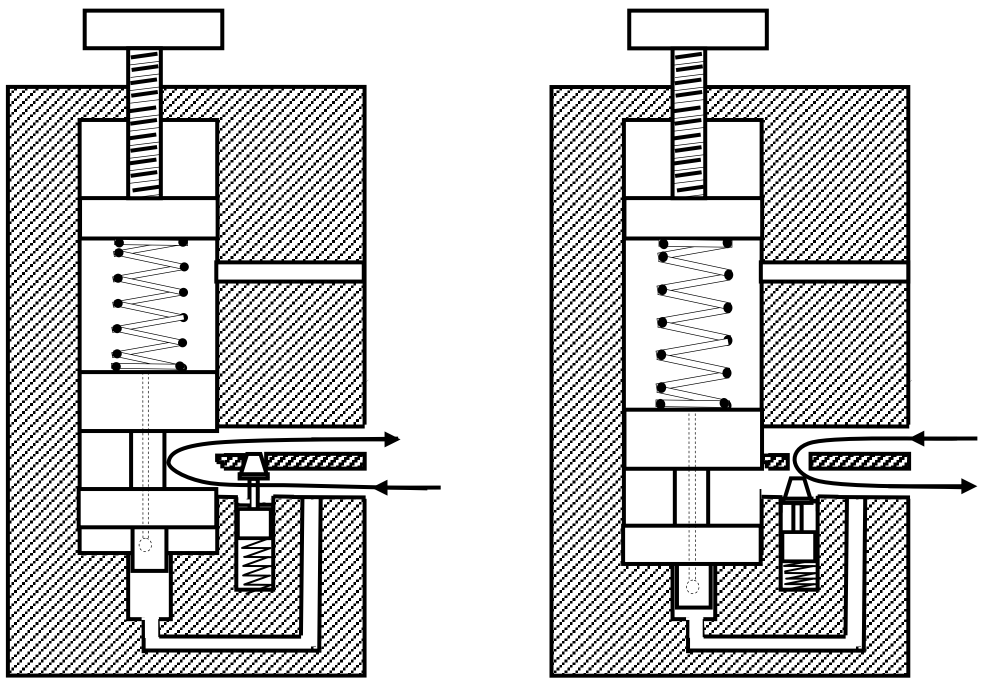


Figure 1.14 Counterbalance valve.

valve's primary port is connected to the cylinder's rod end and the secondary port to the directional control valve. The pressure setting is slightly higher than that required to keep the load from free-falling. When the pressurized fluid flows to the cylinder's cap end, the cylinder extends, increasing pressure in the rod end and shifting the main spool in the counterbalance valve. This creates a path that permits the fluid to flow through the secondary port via the directional control valve and to the reservoir. As the load is raised, the integral check valve opens to allow the cylinder to retract freely.

If it is necessary to relieve back pressure at the cylinder and increase the force at the bottom of the stroke, the counterbalance valve can be operated remotely. Counterbalance valves are usually drained internally. When the cylinder extends, the valve must open and its secondary port should be connected to the reservoir. When the cylinder retracts, it matters little that load pressure is felt in the drain passage because the check valve bypasses the valve's spool. Graphic symbol of a pressure-reducing valve is shown in Fig. 1.15.

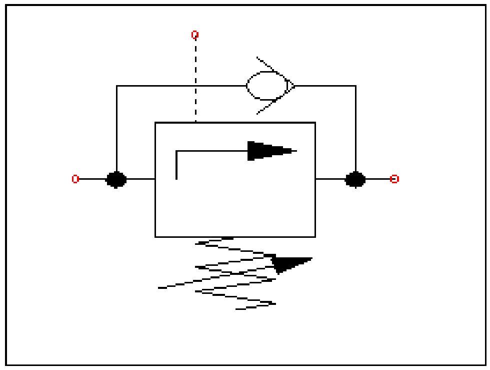


Figure1.15 Symbolic representation of a counterbalance valve.

1.5.1Application of a Counterbalance Valve

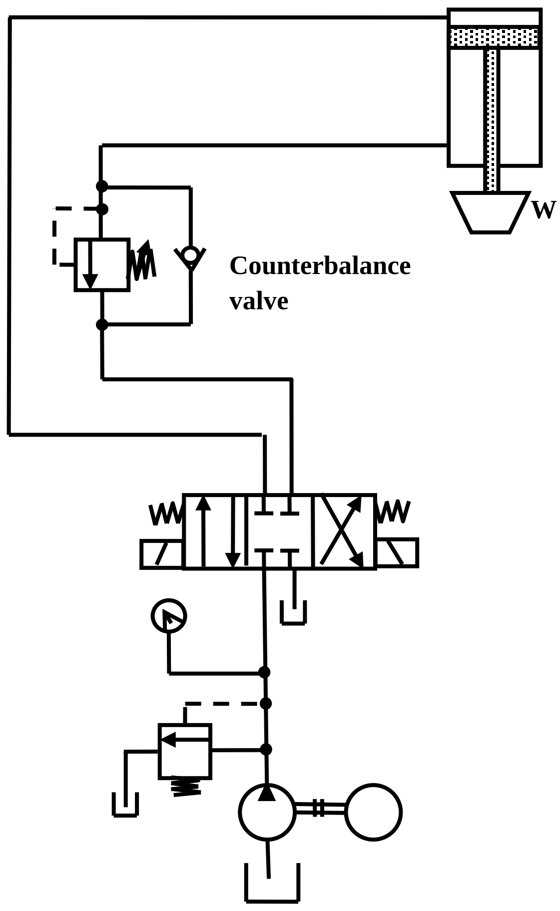


Figure 1.16 Application of a counterbalance valve.

Counterbalance valves are commonly used to counterbalance a weight or external force or counteract a weight such as a platen or a press and keep it from freefalling.Figure1.16 illustrates the use of a counterbalance or back-pressure valve to keep a vertically mounted cylinder in the upward position while the pump idles, that is, when the DCV is in its center position. During the downward movement of the

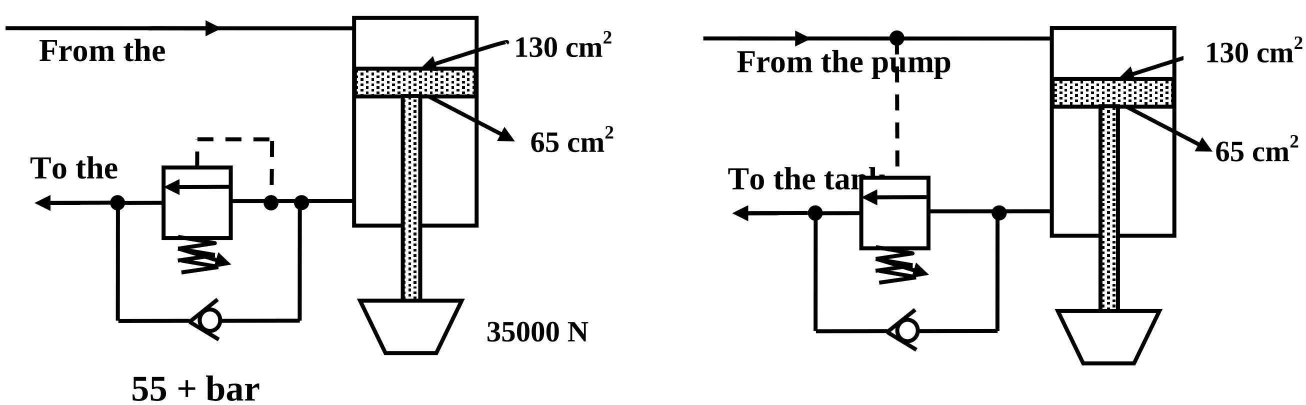
cylinder, the counterbalance valve is set to open at slightly above the pressure required to hold the piston up (a check valve does not permit flow in this direction). The control signal for the counterbalance valve can be obtained from the blank end or rod end of the cylinder.
If derived from the rod end, the pressure setting of the counterbalance valve equals the ratio of the load to the annulus area of the piston. If derived from the blank end, the pressure setting equals the ratio of load to the area of piston. This pressure is less and hence usually it has to be derived from the blank end. This permits the cylinder to be forced downward when pressure is applied on the top. The check valve is used to lift the cylinder up as the counterbalance valve is closed in this direction. The directional control valve unloads the pump.

1.6 Source of Pilot Pressure in Counterbalance Valves

When a counterbalance valve is used in large vertical presses, it may be important to analyze the source of pilot operating pressure. Figure 1.17 shows a comparison between direct pilot and remote pilot operation.
Through the application of Pascal's law, we have

$$p = \frac{\text{Force}}{\text{Area}} = \frac{35000}{65 \times 10^{-4}} \cong 55\,\text{bar}$$

If the pilot pressure is taken directly, then the counterbalance valve operates at about 55 bar or slightly higher because of inertia and friction. In the other case, where the remote pilot pressure is taken from the pressure line at the top of the cylinder, a choice of operating pressure can be made for the valve. A counterbalance valve is normally a closed valve and remains closed until acted upon by the remote pilot pressure source. Therefore, a much lower spring force can be selected to allow the valve to operate at a lesser pilot pressure. It should also be noted that the press load cannot move downward unless flow from the pump is directed into the top of the cylinder, which isa normal function of the machine.



(a) (b)

Figure 1.17(a) Direct pilot and (b) remote pilot.

1.7 Pressure Sequence Valve

A sequence valve is a pressure-control valve that is used to force two actuators to operate in sequence. They are similar to pressure-relief valves. Schematic diagram of sequence valve is shown in Fig. 1.18. Instead of sending flow back to the tank, a sequence valve allows flow to a branch circuit, when a preset pressure is reached. The check valve allows the sequence valve to be bypassed in the reverse direction. The component enclosure line indicates that the check valve is an integral part of the component. The sequence valve has an external drain line; therefore, a line must be connected from the sequence valve's drain port to the tank. The symbol for a sequence valve is shown in Fig. 1.19.

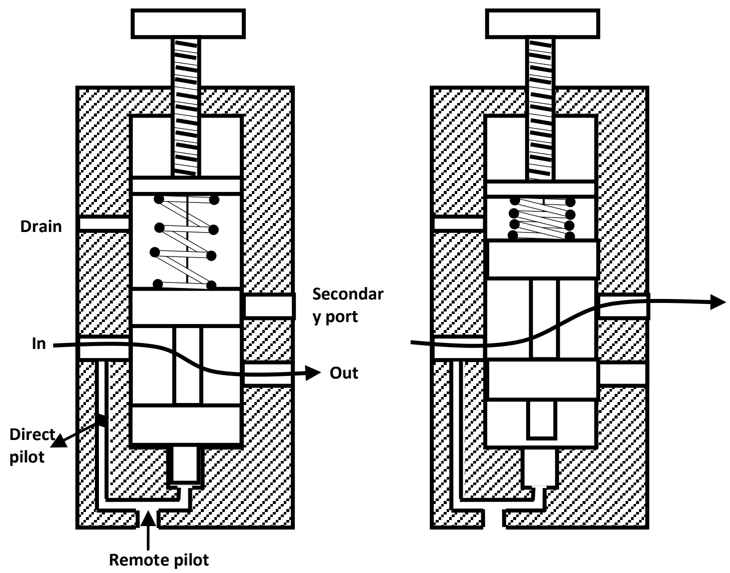


Figure 1.18 Sequence valve.

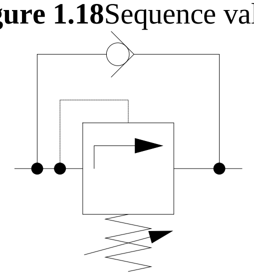


Figure 1.19 Sequence valve with a check valve.

1.7.1 Application of a Sequence Valve

The hydraulic circuit shown in Fig. 1.20 is an example of an application of a sequence valve in which a clamp cylinder extends first to hold a workpiece and then a second cylinder extends to bend the workpiece in the desired shape.

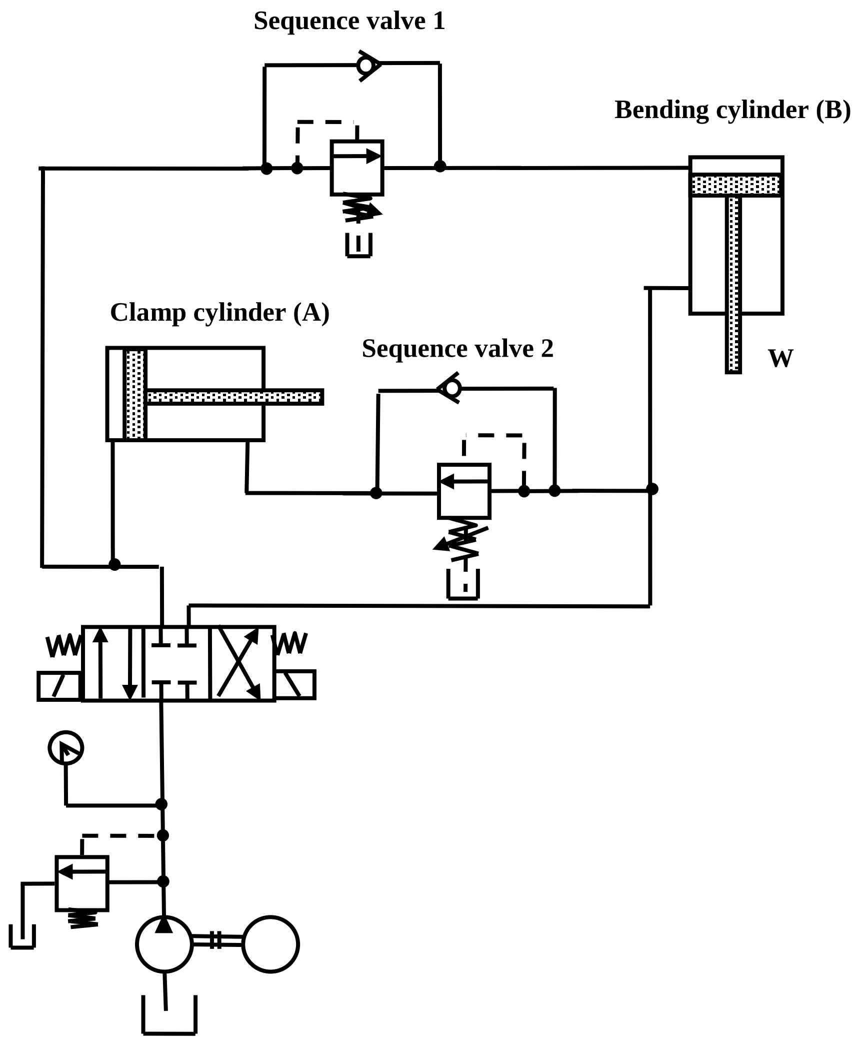


Figure 1.20 Application of a sequence valve.

In this circuit, two cylinders are connected in parallel. Without the sequence valve, these cylinders would extend together as they are both unloaded. In order for this circuit to function properly, the clamp cylinder must extend completely before the bending cylinder begins to extend. The sequence valve accomplishes this by not allowing flow into the bending cylinder branch of circuit until the clamp cylinder has reached the end of its stroke. When the clamp cylinder extends completely, the pressure rises and opens the sequence valve,thus allowing the bending cylinder to extend. The sequence valve must be set high enough so that it opens only after the complete extension of the clamp cylinder.

During the retraction of cylinders, the check valve allows the sequence valve to be bypassed. The sequence valve has no effect on the circuit in this situation. Both cylinders retract together because both are unloaded and split the pump flow.

1.8 Cartridge Valves

Cartridge valves consist of a valve shell that can be mounted in a standard recess in a valve block or manifold. The machine manufacturer does not have to worry about tolerances of moving spools and poppets because these are taken care by the hydraulic valve manufacturer. This is very advantageous for batch production and modularized packages or integrated circuits. Cartridge valves eliminate expensive and potentially leaking pipework and connectors.Cartridge valves can be used as follows:

1. Leak-proof direction control valve.
2. Check valve to obtain unidirectional flow.
3. Throttle valve to control and limit the rate of flow.

The valve shell or body has two main ports (A and B) that are connected or separated by a poppet or a spool. The poppet-type cartridge valve is basically a check valve that can be pilot operated in a number of ways, whereas the spool-type cartridge valve is used as a variable restrictor that is either normally open or closed by the action of the control or vice versa. The actions of the two types of cartridge valvesare completely different.

1.8.1 Poppet-Type Cartridge Valves

In some designs, a poppet fits into cavity and is held in position by a cover or a top plate that contains all pilot connections. Others are designed to fit the standard cavities used by some conventional cartridge valves. Logic elements that have a balanced poppet or spools can be modulated and are largely used as pressure controls. Those with unbalanced poppets are primarily used for switching functions such as directional controls or where the poppet movement can be limited as flow controls.

The principal advantages of poppet-type valves are as follows:

1. A very high flow rate for a relatively small physical size.
2. A positive seal can be obtained.
3. May be extremely rapid acting but can also be easily adopted for soft switching.
4. The shape of the poppet or spool together with its seat can be varied to give different operating characteristics to the valve assembly.

The major disadvantage is that unbalanced poppets, being responsive to pressure changes on all ports, may malfunction owing to pressure surges. Particular care has to be taken in the circuit design to ensure the safe operation. The opening and closing movements of the poppet in a cartridge valve are pressure-dependent and a function of the forces in these areas:

$A_{\mathrm{A}} = \text{Effective area of the poppet at port A}$

$A_{\mathrm{B}} = \text{Effective area of the poppet at port B}$

$A_{\mathrm{X}} = \text{Effective area of the poppet at port X}$

Then

$$A_{\mathrm{X}} = A_{\mathrm{A}} + A_{\mathrm{B}}$$

In a balanced poppet-type shown diagrammatically and symbolically in Fig.1.21, $A_{\mathrm{B}} = 0$ and the areas A_{X} and A_{A} are equal.Pilot X controls the function of the valve. If X is connected to port B, the valve operates as a check valve allowing flow from A to B by opening the poppet but preventing flow from B to A by closing the poppet. If port X is connected to the external pressure, the valve can be used to control closing or opening pressure.

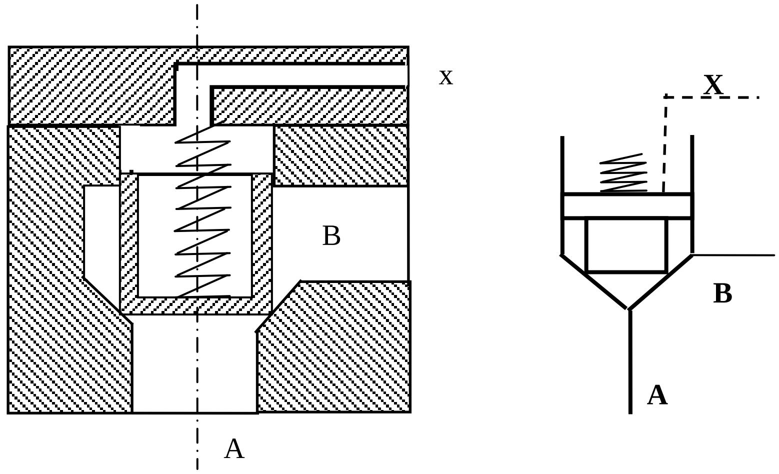


Figure 1.21 Balanced poppet cartridge valve area ratio, $A_{\mathrm{A}} = A_{\mathrm{X}}$.

In an unbalanced poppet-type valve shown diagrammatically and symbolically in Fig.1.22, it is possible to obtain a different area ratio, typically

$$A_{\mathrm{X}} : A_{\mathrm{A}} = 1:1.1, \text{ where } A_{\mathrm{B}} = 0.1 A_{\mathrm{A}}$$
$$A_{\mathrm{X}} : A_{\mathrm{A}} = 1:1.05, \text{ where } A_{\mathrm{B}} = 0.05 A_{\mathrm{A}}$$
$$A_{\mathrm{X}} : A_{\mathrm{A}} = 1:2, \text{ where } A_{\mathrm{B}} = A_{\mathrm{A}}$$

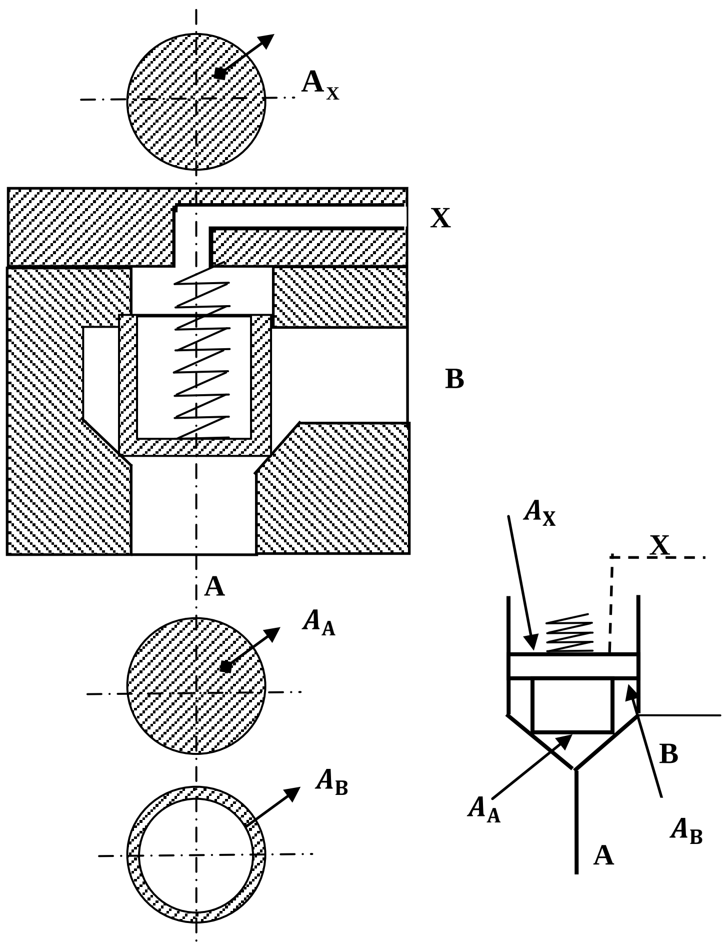


Figure 1.22 Unbalanced poppet cartridge valve area ratio, $A_X = A_A + A_B$.

With the pilot X vented, pressure at port A or B has to overcome the bias spring force only for flow in either direction. However, it can be held closed by a pressure on the pilot port that is dependent on the poppet area ratios.

Figure 1.23 shows how a large double-acting cylinder can be controlled using cartridge valves. A small double-solenoid-operated directioncontrol valve feeds pilot pressure signals to four cartridge valves that are coupled in pairs to each end of the double-acting cylinder. One cartridge valve from each pair is permanently connected to the tank drain line and the other to the pump pressure line. In the position drawn, all the four valves are held closed by the pilot pressure signals and the cylinder position is locked. When solenoid A is energized, pilot pressure is maintained on valves 1 and 3 which remain closed. But valves 2 and 4 are released and open under the influence of fluid pressure in the main system. The fluid under pressure, therefore, flows from the pump through the cartridge valve to the piston side of the cylinder and the cylinder extends. The fluid from the rod end of the cylinder flows through valve 4 back to the tank. When solenoid B is energized, cartridge valves 2 and 4 are closed under pilot pressure and valves 1 and 2 are released, causing the cylinder to retract.

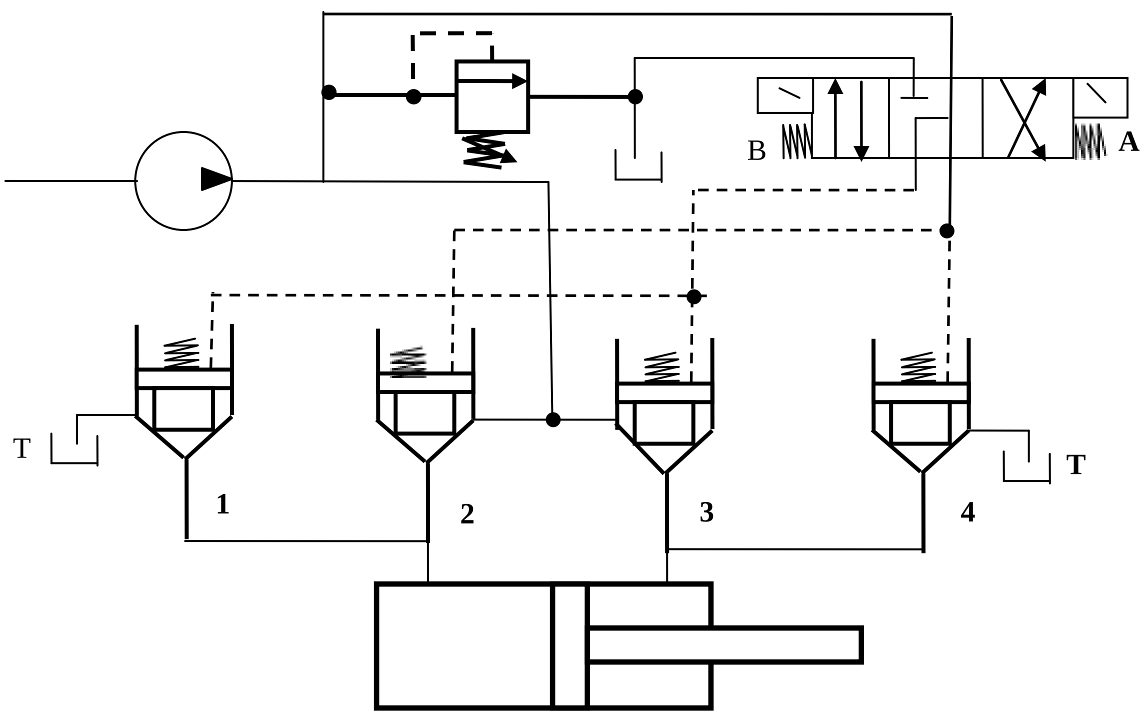


Figure 1.23 Control of a double-acting cylinder using cartridge logic valves.

Objective-Type Questions
Fill in the Blanks

1. A relief valve is similar to a ______ in an electrical system.
2. A pilot-operated pressure relief valve consists of a small ______ and a main relief valve.
3. ______ is used to maintain reduced pressures in specified locations of hydraulic systems.
4. ______ is used to maintain a back pressure on a vertical cylinder to prevent it from falling due to gravity.
5. A pilot-operated unloading relief valve is the same as a pilot-operated relief valve with the addition of an ______ spool.

State True or False

1. Pressure-relief valves limit the maximum pressure in a hydraulic circuit by providing an alternate path for fluid flow when the pressure reaches a preset level.
2. A pilot-operated pressure-relief valve cannot be operated using a remote.
3. A common application of an unloading valve is in high–low pump circuits and punching press.
4. Sequence valves are similar to pressure-relief valves.
5. An unloading valve requires electric signals.

Review Questions

1. Explain the function of pressure-control valves in hydraulic power systems.
2. Discuss in detail the static characteristics of a direct-operated relief valve, and explain how to reduce the over-ride pressure.
3. Draw schematically a pilot-operated relief valve and explain its function.
4. Discuss the application of a pilot-operated relief valve.
5. Discuss the principle of pressure reduction in fluid power systems.
6. Discuss briefly the operation of a pilot-operated pressure reducer.
7. Explain the function of a direct-operated sequence valve.
8. What are the differences between relief and sequence valves?
9. Discuss the application of a sequence valve used in hydraulic systems.
10. Name two applications of a counterbalance valve.
11. What is the function of a sequence valve?
12. What is the function of an unloading valve?
13. How is the unloading valve different from a pressure relief valve?
14. What is the function of a pressure relief valve in fluid power systems?
15. What is the advantage of using an unloading circuit when feed and speed of a machine need to be varied?

Answers
Fill in the Blanks

1.Fuse
2.Pilot relief valve
3.Pressure-reducing valve
4.Counterbalance
5. Unloading

State True or False

1.True
2.False
3.True
4.True
5.False

Lecture 20

FLOW-CONTROL VALVES

Learning Objectives
Upon completion of this chapter, the student should be able to:

- Explain various functions of flow-control valves.
- Explain various classifications of pressure-control valves.
- Describe the working and construction of various non-compensated flow-control valves.
- Differentiate between compensated and non-compensated flow-control valves.
- Identify the graphic symbols for various types of flow-control valves.
- Explain different applications of flow-control valves.
- Explain the working principle of bleed-off circuits.
- Evaluate the performance of hydraulic systems using flow-control valves.

1.1 Introduction

Flow-control valves, as the name suggests, control the rate of flow of a fluid through a hydraulic circuit. Flow-control valves accurately limit the fluid volume rate from fixed displacement pump to or from branch circuits. Their function is to provide velocity control of linear actuators, or speed control of rotary actuators. Typical application include regulating cutting tool speeds, spindle speeds, surface grinder speeds, and the travel rate of vertically supported loads moved upward and downward by forklifts, and dump lifts. Flow-control valves also allow one fixed displacement pump to supply two or more branch circuits fluid at different flow rates on a priority basis. Typically, fixed displacement pumps are sized to supply maximum system volume flow rate demands. For industrial applications feeding two or more branch circuits from one pressurized manifold source, an oversupply of fluid in any circuit operated by itself is virtually assured. Mobile applications that supply branch circuits, such as the power steering and front end loader from one pump pose a similar situation. If left unrestricted, branch circuits receiving an oversupply of fluid would operate at greater than specified velocity, increasing the likelihood of damage to work, hydraulic system and operator.

1.1.1 Functions of Flow-Control Valves

Flow-controlvalves have several functions, some of which are listed below:

1. **Regulate the speed of linear and rotary actuators:**They control the speed of piston that is dependent on the flow rate and area of the piston:

$$\text{Velocity of piston } (V_\text{p}) \text{ (m/s)} = \frac{\text{Flow rate in the actuator (m}^3\text{ / s)}}{\text{Piston area (m}^2\text{)}} = \frac{Q}{A_\text{p}}$$

2. **Regulate the power available to the sub-circuits by controlling the flow to them:**

$$\text{Power (W)} = \text{Flow rate (m}^3\text{/s)} \times \text{Pressure (N/m}^2\text{)}$$

$$\Rightarrow P = Q \times p$$

3. **Proportionally divide or regulate the pump flow to various branches of the circuit:**It transfers the power developed by the main pump to different sectors of the circuit to manage multiple tasks, if necessary.

A partially closed orifice or flow-control valve in a hydraulic pressure line causes resistance to pump flow. This resistance raises the pressure upstream of the orifice to the level of the relief-valve setting and any excess pump flow passes via the relief valve to the tank (Fig. 1.1).

In order to understand the function and operation of flow-control devices, one must comprehend the various factors that determine the flow rate(Q) across an orifice or a restrictor. These are given as follows:

1. Cross-sectional area of orifice.
2. Shape of the orifice (round, square or triangular).
3. Length of the restriction.
4. Pressure difference across the orifice (Δp).
5. Viscosity of the fluid.

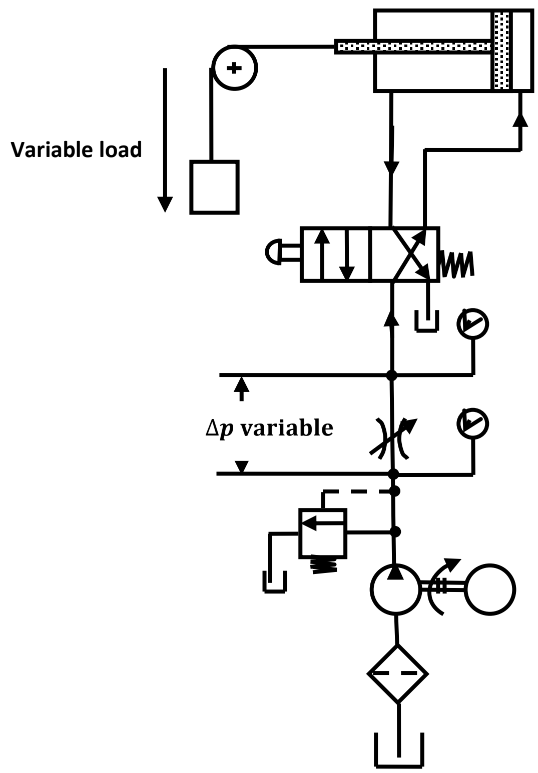


Figure 1.1 simple restrictor-type flow-control valves.

Thus, the law that governs the flow rate across a given orifice can be approximately defined as

$$Q^2 \propto \Delta p$$

This implies that any variation in the pressure upstream or downstream of the orifice changes the pressure differential Δp and thus the flow rate through the orifice (Fig. 1.2).

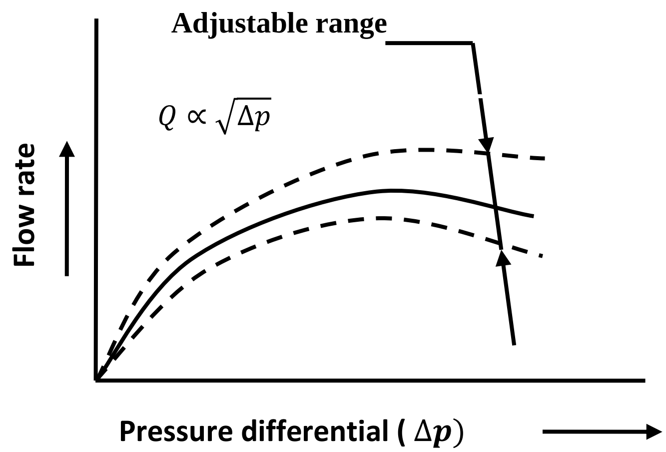


Figure 1.2 Variation of flow rate with pressure drop.

1.1.2Classification of Flow-Control Valves

Flow-control valves can be classified as follows:

1. Non-pressure compensated.
2. Pressure compensated.

1.1.2.1 Non-Pressure-Compensated Valves

Non-pressure-compensated flow-control valves are used when the system pressure is relatively constant and motoring speeds are not too critical. The operating principle behind these valves is that the flow through an orifice remains constant if the pressure drop across it remains the same. In other words, the rate of flow through an orifice depends on the pressure drop across it.

The disadvantage of these valves is discussed below. The inlet pressure is the pressure from the pump that remains constant. Therefore, the variation in pressure occurs at the outlet that is defined by the work load. This implies that the flow rate depends on the work load. Hence, the speed of the piston cannot be defined accurately using non-pressure-compensated flow-control valves when the working load varies. This is an extremely important problem to be addressed in hydraulic circuits where the load and pressure vary constantly.

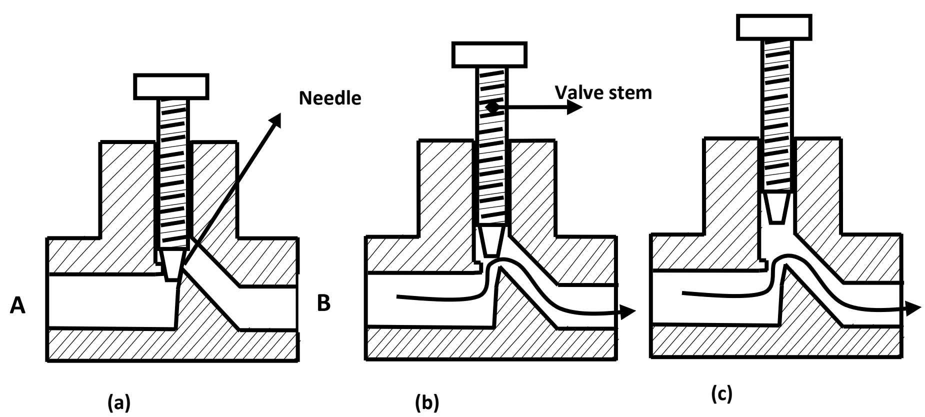


Figure 1.3 Non-pressure-compensated needle-type flow-control valve. (a) Fully closed; (b) partially opened; (c) fully opened.

Schematic diagram of non-pressure-compensated needle-type flow-control valve is shown in Fig. 1.3. It is the simplest type of flow-control valve. It consists of a screw (and needle) inside a tube-like structure. It has an adjustable orifice that can be used to reduce the flow in a circuit. The size of the orifice is adjusted by turning the adjustment screw that raises or lowers the needle. For a given opening position, a needle valve behaves as an orifice. Usually, charts are available that allow quick determination of the controlled flow rate for given valve settings and pressure drops.

Sometimes needle valves come with an integrated check valve for controlling the flow in one direction only. The check valve permits easy flow in the opposite direction without any restrictions. As shown in Fig. 1.4, only the flow from A to B is controlled using the needle. In the other direction (B to A), the check valve permits unrestricted fluid flow.

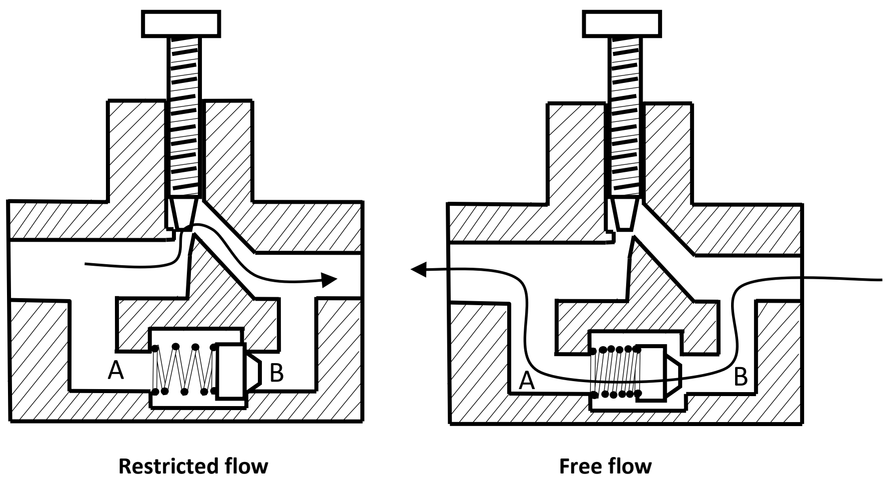


Figure 1.4 Flow-controlvalve with an integrated check valve.

1.1.2.2Pressure-Compensated Valves

Pressure-compensated flow-control valvesovercome the difficulty causedby non-pressure-compensated valves by changing the size of the orifice in relation to the changes in the system pressure. This is accomplished through a spring-loaded compensator spool that reduces the size of the orifice when pressure drop increases. Once the valve is set, the pressure compensator acts to keep the pressure drop nearly constant. It works on a kind of feedback mechanism from the outlet pressure. This keeps the flow through the orifice nearly constant.

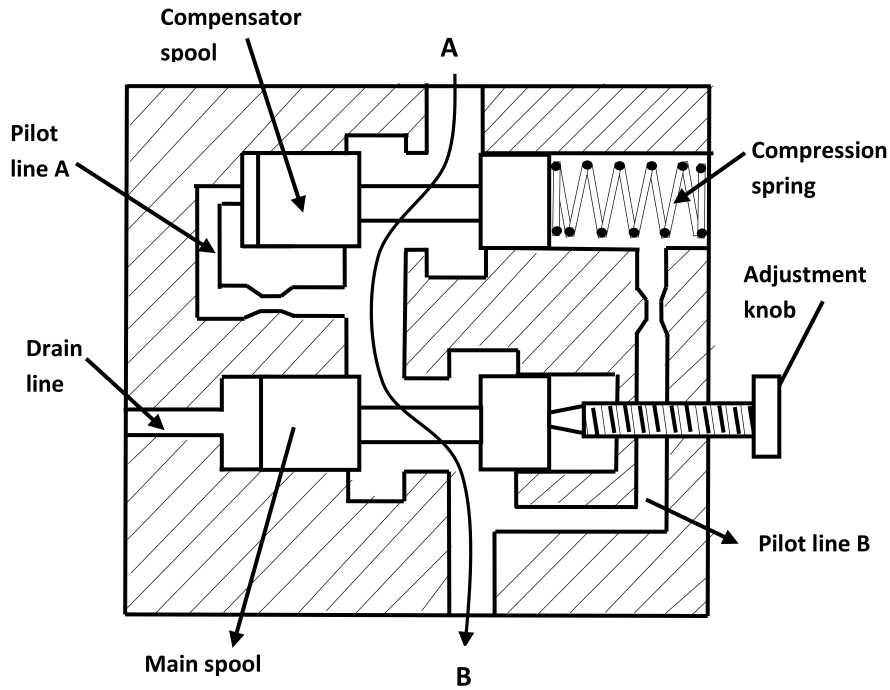


Figure 1.5 Sectional view of a pressure-compensated flow-control valve.

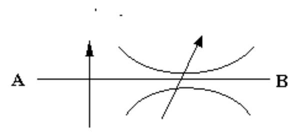


Figure 1.6 Graphic symbol of a pressure-compensated flow-control valve.

Schematic diagram of a pressure compensated flow-control valve is shown in Fig. 1.5 and its graphical symbol in Fig. 1.6. A pressure-compensated flow-control valve consists of a main spool and a compensator spool. The adjustment knob controls the main spool's position, which controls the orifice size at the outlet. The upstream pressure is delivered to the valve by the pilot line A. Similarly, the downstream pressure is ported to the right side of the compensator spool through the pilot line B. The compensator spring biases the spool so that it tends toward the fully open position. If the pressure drop across the valve increases, that is, the upstream pressure increases relative to the downstream pressure, the compensator spool moves to the right against the force of the spring. This reduces the flow that in turn reduces the pressure drop and tries to attain an equilibrium position as far as the flow is concerned.

In the static condition, the hydraulic forces hold the compensator spool in balance, but the bias spring forces it to the far right, thus holding the compensator orifice fully open. In the flow condition, any pressure drop less than the bias spring force does not affect the fully open compensator orifice,but any pressure drop greater than the bias spring force reduces the compensator orifice. Any change in pressure on either side of the control orifice, without a corresponding pressure change on the opposite side of the control orifice, moves the compensator spool. Thus, a fixed differential across the control orifice is maintained at all times. It blocks all flow in excess of the throttle setting. As a result, flow exceeding the preset amount can be used by other parts of the circuit or return to the tank via a pressure-relief valve.

Performance of flow-control valve is also affected by temperature changes which changes the viscosity of the fluid. Therefore, often flow-control valves have temperature compensation. Graphical symbol for pressure and temperature compensated flow-control valve is shown in Fig. 1.7.

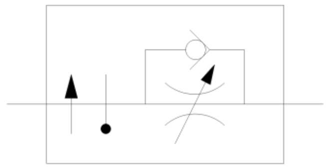


Figure 1.7 Pressure- and temperature-compensated flow-control valve.

1.2Speed-Controlling Circuits

In hydraulic operations, it is necessary to control the speed of the actuator so as to control the force, power, timing and other factors of the operation. Actuator speed control is achieved by controlling the rate of flow into or out of the cylinder.

Speed control by controlling the rate of flow into the cylinder is called meter-in control.Speed control by controlling the rate of flow out of the cylinder is called meter-out control.

1.2.1 Meter-In Circuit

Figure 1.8 shows a meter-in circuit with control of extend stroke. The inlet flow into the cylinder is controlled using a flow-control valve. In the return stroke, however, the fluid can bypass the needle valve and flow through the check valve and hence the return speed is not controlled. This implies that the extending speed of the cylinder is controlled whereas the retracing speed is not.

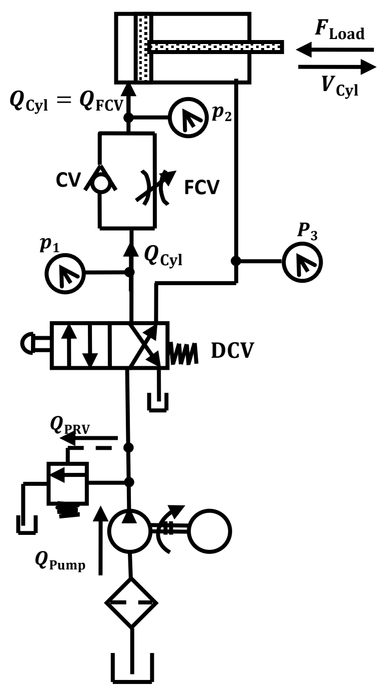


Figure 1.8 Meter-in circuit.

1.2.2 Meter-Out Circuit

Figure 1.9 shows a meter-out circuit for flow control during the extend stroke. When the cylinder extends, the flow coming from the pump into the cylinder is not controlled directly. However, the flow out of the cylinder is controlled using the flow-control valve (metering orifice). On the other hand, when the cylinder retracts, the flow passes through the check valve unopposed, bypassing the needle valve. Thus, only the speed during the extend stroke is controlled.

Both the meter-in and meter-out circuits mentioned above perform the same operation (control the speed of the extending stroke of the piston), even though the processes are exactly opposite to one another.

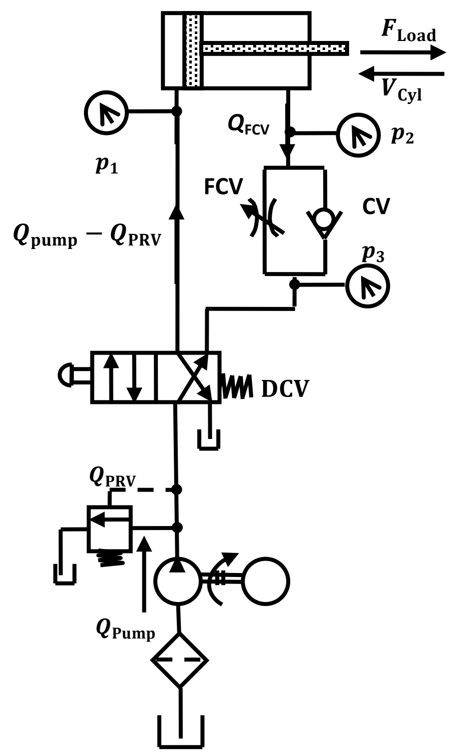


Figure 1.9 Meter-Out circuit.

1.2.3 Bleed-Off Circuit

Compared to meter-in and meter-out circuits, a bleed-off circuit is less commonly used. Figure 1.10 shows a bleed-off circuit with extend stroke control. In this type of flow control, an additional line is run through a flow-control valve back to the tank. To slow down the actuator, some of the flow is bledoff through the flow-control valve into the tank before it reaches the actuator. This reduces the flow into the actuator, thereby reducing the speed of the extend stroke.

The main difference between a bleed-off circuit and a meter-in/meter-out circuit is that in a bleed-off circuit, opening the flow-control valve decreases the speed of the actuator, whereas in the case of a meter-in/meter-out circuit, it is the other way around.

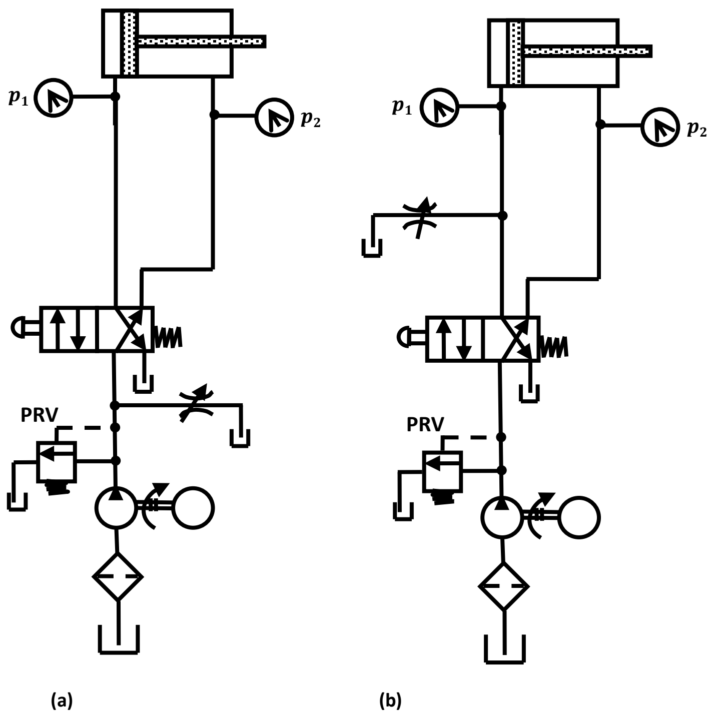


Figure 1.10 Bleed-off circuits:(a) Bleed-off for both directions and (b) bleed-off for inlet to the cylinder or motor.

Example 1.1

A 55-mm diameter sharp-edged orifice is placed in a pipeline to measure the flow rate. If the measured pressure drop is 300 kPa and the fluid specific gravity is 0.90, find the flow rate in units of $\mathrm{m^3/s}$.

Solution: For a sharp-edged orifice, we can write

$$Q = 0.0851 A C_\mathrm{V} \sqrt{\frac{\Delta p}{\mathrm{SG}}}$$

where Q is the volume flow rate in LPM, C_V is the capacity coefficient = 0.80 for the sharp-edge orifice, c = 0.6 for a square-edged orifice, A is the area of orifice opening in mm^2, Δp is the pressure drop across the orifice (kPa) and SG is the specific gravity of the flowing fluid = 0.9. Now,

$$A_\mathrm{orifice} = \frac{\pi}{4}(D_\mathrm{orifice}{}^2) = \frac{\pi}{4}(55^2) = 2376\ \mathrm{mm}^2$$

Using the orifice equation we can find the flow rate as

$$Q\ (\mathrm{LPM}) = 0.0851 \times 2376 \times 0.80 \sqrt{\frac{300}{0.9}}$$

$$= 2953.3\ \mathrm{LPM} = 0.0492\ \ \mathrm{m^3/s}$$

Example 1.2

For a given orifice and fluid, a graph can be generated showing a Δp versus Q relationship. For the orifice and fluid in Example 1.1, plot the curves and check the answers obtained mathematically. What advantage does the graph have over the equation? What is the disadvantage of the graph?

Solution: From Example 1.1, we have

$$Q\text{ (LPM)} = 0.0851 \times 2376 \times 0.80\sqrt{\frac{300}{0.9}}$$

We can write the general expression as

$$Q\text{ (LPM)} = 0.0851 \times 2376 \times 0.80\sqrt{\frac{\Delta p}{0.9}} = 161.76\sqrt{\frac{\Delta p}{0.9}} = 170.5\sqrt{\Delta p}$$

Using Excel, the graph shown in Fig. 1.11 is obtained.

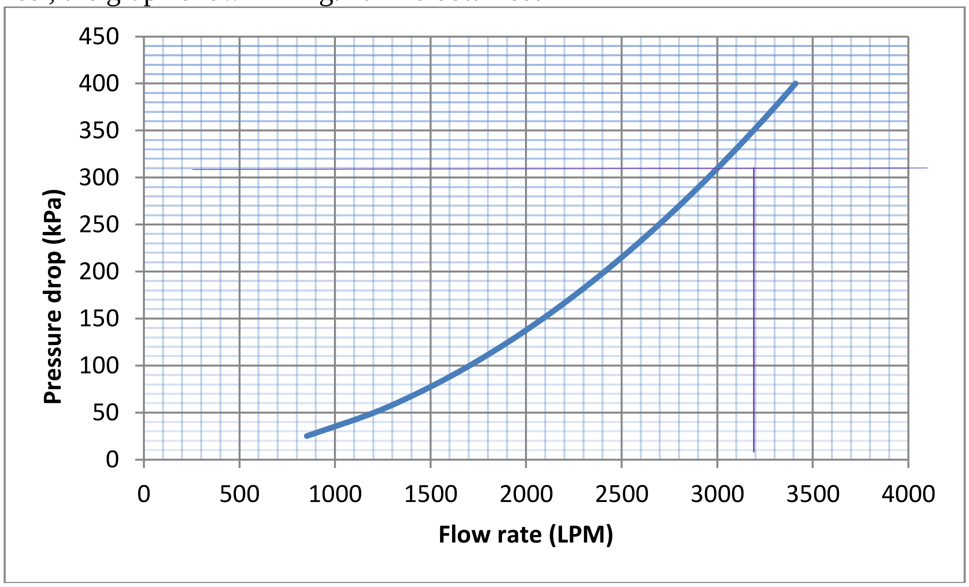


Figure 1.11 Pressure drop versus flow rate.

From the graph, corresponding to $\Delta p = 300$ kPa, we get Q = 2950 LPM which is close to 2953.3 LPM.A graph is quicker to use but is not as accurate as the equation. A pressure gauge can be calibrated (according to this relationship) to read Q directly rather than Δp .

Example 1.3

Determine the flow rate through a flow-control valve that has a capacity coefficient of $2.2\,\text{LPM}/\sqrt{\text{kPa}}$ and a pressure drop of 687 kPa. The fluid is hydraulic oil with a specific gravity of 0.90.

Solution: For a sharp-edged orifice, we can write

$$Q = 2.2\sqrt{\frac{\Delta p}{\text{SG}}} = 2.2\sqrt{\frac{687}{0.9}} = 60.8\,\text{LPM}$$

Example 1.4

The system shown in Fig. 1.12 has a hydraulic cylinder with a suspended load *W*. The cylinder piston and rod diameters are 50.8 and 25.4 mm, respectively. The pressure-relief valve setting is 5150 kPa. Determine the pressure p_2 for a constant cylinder speed:

(a) $W = 8890$ N
(b) $W = 0$ (load is removed)
(c) Determine the cylinder speeds for parts (a) and (b) if the flow-control valve has a capacity coefficient of $0.72\,\text{LPM}/\sqrt{\text{kPa}}$. The fluid is hydraulic oil with a specific gravity of 0.90.

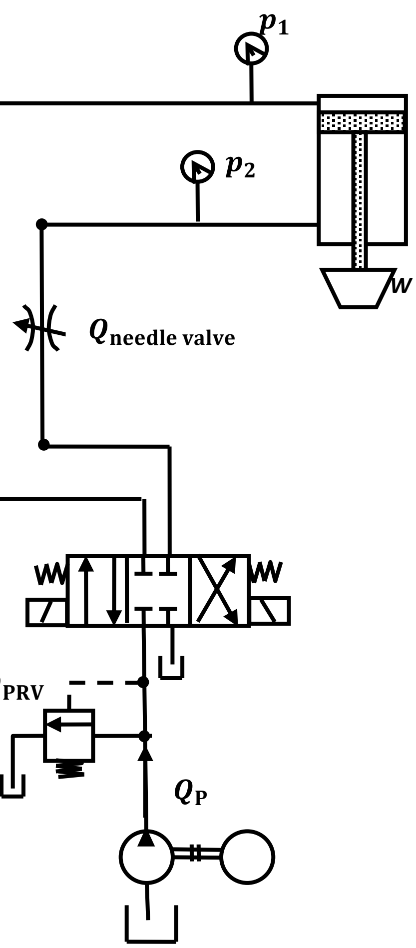


Figure 1.12 Hydraulic cylinder with a suspended weight.

Solution:
For a constant cylinder speed, the summation of the forces on the hydraulic cylinder must be equal to zero. Thus, we have

$$-W - p_1 A_{\text{p}} + p_2 (A_{\text{p}} - A_{\text{r}}) = 0$$

where $p_1 = \text{pressure-relief valve setting} = 5150\,\text{kPa}$. Now

$$A_p = \frac{\pi}{4}(D_P{}^2) = \frac{\pi}{4}(0.0508^2) = 0.00203\,\mathrm{m}^2$$

$$A_r = \frac{\pi}{4}(D_R{}^2) = \frac{\pi}{4}(0.0254^2) = 0.000506\,\mathrm{m}^2$$

So

$$A_p - A_r = 0.00152\ \mathrm{m}^2$$

Case 1: If W = 8890 N.

$$-W - p_1 A_p + p_2(A_p - A_r) = 0$$

$$\Rightarrow -8890 - 5150\times 10^3\,\mathrm{N/m}^2 \times 2.03\times 10^{-3}\ \mathrm{m}^2 + p_2(0.00152\ \mathrm{m}^2) = 0$$

$$\Rightarrow -8890 - 10450\,\mathrm{m}^2 + p_2(0.00152) = 0$$

$$\Rightarrow p_2 = 12700\,\mathrm{kPa}$$

Case 2: If W = 0.

$$0 - 5150\times 10^3\ \mathrm{N/m}^2 \times 2.03\times 10^{-3}\ \mathrm{m}^2 + p_2(0.00152\,\mathrm{m}^2) = 0$$

$$\Rightarrow p_2 = 6880\,\mathrm{kPa}$$

Case 3: Cylinder speed for case 1: For a sharp-edged orifice, we can write

$$Q = C_V\sqrt{\frac{\Delta p}{SG}} = 0.72\sqrt{\frac{12700}{0.9}} = 85.5\,\mathrm{LPM}$$

where $\Delta p = p_2$ because the flow-control valve discharges directly to the oil tank. This is the flow rate through the flow-control valve and thus the flow rate of the fluid leaving the hydraulic cylinder. Thus, we have

$$v_p(A_p - A_r) = Q$$

$$\Rightarrow v_p\ (\mathrm{m/s})(0.00152)\ \mathrm{m}^2 = 85.5\ \ \mathrm{L/min} \times \frac{1\,\mathrm{m}^3}{10^3\ \mathrm{L}} \times \frac{1\,\mathrm{min}}{60\,\mathrm{s}}$$

$$\Rightarrow v_p = 0.938\ \mathrm{m/s}$$

Case 4: Cylinder speed for case 2. We have

$$Q = C_V\sqrt{\frac{\Delta p}{SG}} = 0.72\sqrt{\frac{6880}{0.9}} = 63\mathrm{LPM}{=}63\mathrm{L/min} \times \frac{1\,\mathrm{m}^3}{10^3\ \mathrm{L}} \times \frac{1\,\mathrm{min}}{60\,\mathrm{s}} = 0.00105\frac{\mathrm{m}^3}{s}$$

Also we can write

$$\begin{aligned} Q &= \text{Velocity}\times\text{Area} \\ &= v_p \times A \\ &= 0.00105\frac{\mathrm{m}^3}{s} \end{aligned}$$

$$\Rightarrow v_p\ (\mathrm{m/s})(0.00152)\ \mathrm{m}^2 = 0.00105$$

$$\Rightarrow v_p = 0.691\ \mathrm{m/s}$$

Example 1.5

A cylinder has to exert a forward thrust of 100 kN and a reverse thrust of 10 kN. The effects of using various methods of regulating the extend speed will be considered. In all the cases, the retract speed should be approximately 5 m/min utilizing full pump flow. Assume that the

maximum pump pressure is 160 bar and the pressure drops over the following components and their associated pipe work (where they are used):
Filter = 3 bar
Directional control valve (DCV) = 2 bar
Flow-control valve (controlled flow) = 10 bar
Flow-control valve (check valve) = 3 bar
Determine the following:

(a) The cylinder size (assume the piston-to-rod area ratio to be 2:1).
(b) Pump size.
(c) Circuit efficiency when using the following:

Case 1: No flow controls (calculate the extend speed).
Case 2: Meter-in flow control for extend speed 0.5 m/min.
Case 3: Meter-out flow control for extend speed 0.05 m/min.

Solution:

Case 1: No flow controls (Fig. 1.13)
Part (a) No flow controls
Maximum available pressure at the full bore end of cylinder = 160 − 3 − 2 = 155 bar
Back pressure at the annulus side of cylinder = 2 bar.
This is equivalent to 1 bar at the full bore end because of the 2:1 area ratio. Therefore, the maximum pressure available to overcome load at the full bore end is 155 − 1 = 154 bar

Full bore area = Load/Pressure = $\dfrac{100 \times 103}{154 \times 105}$ =0.00649 m^2

Piston diameter = $\left(\dfrac{4 \times 0.00649}{\pi}\right)^{1/2}$

Select a standard cylinder, say with 100-mm bore and 70-mm rod diameter. Then

Full bore area = 7.85×10^{-3} m^2

Annulus area = 4.00×10^{-3} m^2

This is approximately a 2:1 ratio.

Part (b) No flow controls
Flow rate for a return speed of 5 m/min is given by

$$\text{Area} \times \text{Velocity} = 4.00 \times 10^{-3} \times 5 \text{ m}^3/\text{min} = 20 \text{ LPM}$$

Extend speed = $\dfrac{20 \times 10^{-3}}{7.85 \times 10^{-3}}$ = 2.55 m/min

Pressure to overcome load on extend = $\dfrac{100 \times 10^{3}}{7.85 \times 10^{-3}}$ = 12.7 MPa = 127 bar

Pressure to overcome load on retract = $\dfrac{10 \times 10^{3}}{4.00 \times 10^{-3}}$ = 2.5 MPa = 25 bar

(i) Pressure at pump on extend (working back from the DCV tank port)

Pressure drop over DCV B to T	2 × (1/2)	1
Load-induced pressure		127
Pressure drop over DCV P to A		2
Pressure drop over filter		3

Therefore, pressure drop required at the pump during extend stroke = 133 bar

Relief-valve setting = 133 + 10% = 146 bar

(ii) Pressure required at the pump on retract (working from the DCV port as before) is

$$(2 \times 2) + 25 + 2 + 3 = 34 \text{ bar}$$

Note: The relief valve will not be working other than at the extremities of the cylinder stroke. Also, when movement is not required, pump flow can be discharged to the tank at low pressure through the center condition of the DCV.

Part (c) No flow controls

System efficiency:

$$\frac{\text{Energy required to overcome load on the cylinder}}{\text{Total energy into fluid}} = \frac{\text{Flow to the cylinder} \times \text{ Pressure owing to load}}{\text{Flow from the pump} \times \text{ Pressure at the pump}}$$

Efficiency on extend stroke = $\frac{20 \times 127}{20 \times 133} \times 100 = 95.5\ \%$

Efficiency on retract stroke = $\frac{20 \times 25}{20 \times 34} \times 100 = 73.5\ \%$

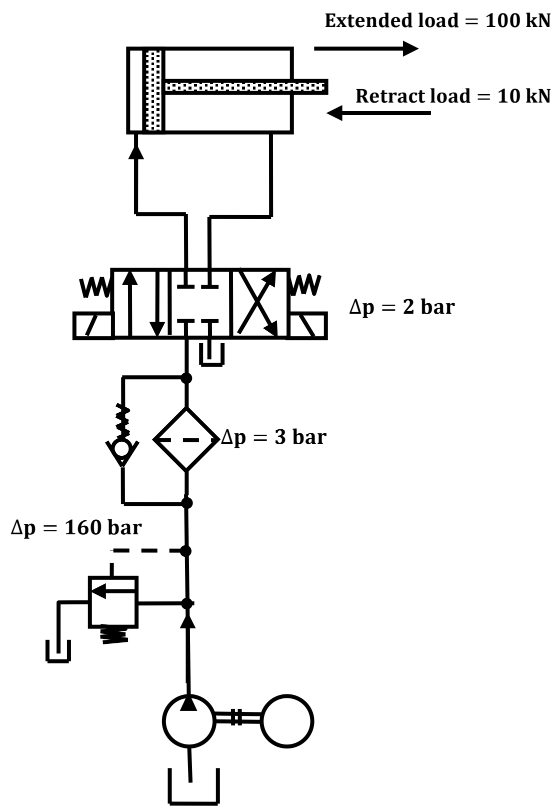


Figure 1.13 Hydraulic cylinder with no control.

Case 2: Meter-in flow control for the extend speed of 0.5 m/min(Fig. 1.14)

Part (a) meter in controls
From case 1,
Select a standard cylinder, say with 100-mm bore and 70-mm rod diameter.

Cylinder 100-mm bore diameter × 70-mm rod diameter
Full bore area 7.85×10^{-3} m^2
Annulus area = 4.00×10^{-3} m^2
Load-induced pressure on extend = 127 bar
Load-induced pressure on retract = 25 bar
Pump flow rate = 20 L/min
Part (b) meter in controls
Flow rate required for the extend speed of 0.5 m/min is

$$7.85 \times 10^{-3} \times 0.5 = 3.93 \times 10^{-3}\ \text{m}^3/\text{min} = 3.93\ \text{L/min}$$

Working back from the DCV tank port:
Pressure required at the pump on retract is

$$(2 \times 2) + (2 \times 3) + 25 + 2 + 3 = 40\ \text{bar}$$

Pressure required on the pump at extend is

$$2 \times (1/2) + 127 + 10 + 2 + 3 = 143\ \text{bar}$$

Relief-valve setting = 143 + 10% = 157 bar

This is close to the maximum working pressure of the pump (160 bar). In practice, it would be advisable to select either a pump with a higher working pressure (210 bar) or use the next standard size of the cylinder. In the latter case, the working pressure would be lower but a higher flow rate pump would be necessary to meet the speed requirements.

Part (c) meter in controls

Now that a flow-control valve has been introduced when the cylinder is on the extend stroke, the excess fluid will be discharged over the relief valve.

$$\text{System efficiency on extend} = \frac{3.93 \times 127}{20 \times 157} \times 100 = 15.9\%$$

$$\text{System efficiency on retract} = \frac{20 \times 25}{20 \times 40} \times 100 = 62.5\%$$

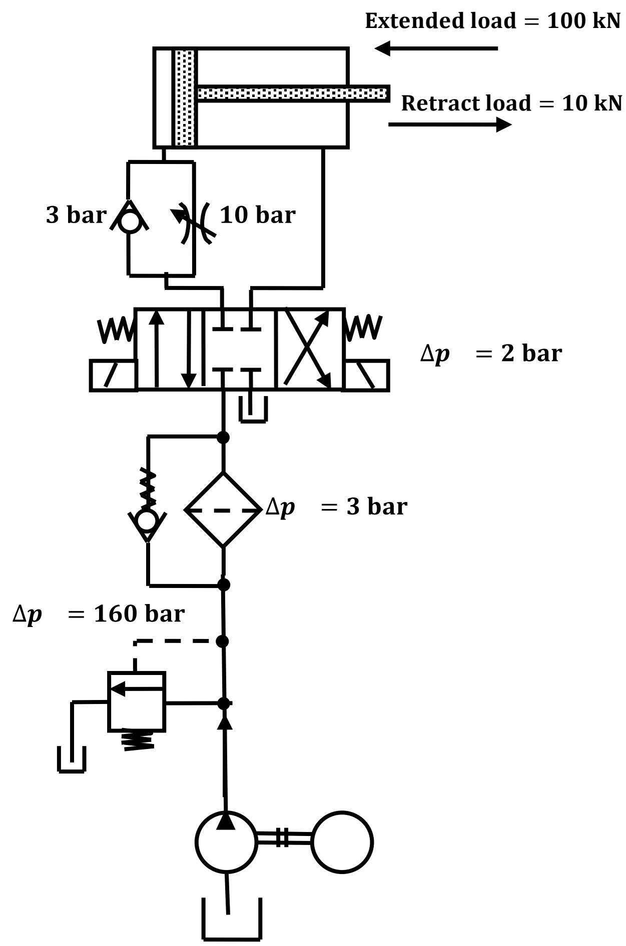


Figure 1.14 Hydraulic cylinder with meter-in control

Case 3: Meter-out flow control for the extend speed of 0.5 m/min(Fig. 1.15)

Cylinder, load, flow rate and pump details are as before (partsa and b of meter in control).

Part (c) meter out controls

Working back from the DCV tank port:

Pressure required at the pump on retract is

$$(2 \times 2) + 25 + 3 + 2 + 3 = 37 \text{ bar}$$

Pressure required at the pump on extend is

$$[2 \times (1/2)] + [10 \times (1/2)] + 127 + 2 + 3 = 138 \text{ bar}$$

Relief-valve setting = 38 + 10 % = 152 bar

$$\text{System efficiency on extend} = \frac{3.93 \times 127}{20 \times 152} \times 100 = 16.4\%$$

$$\text{System efficiency on retract} = \frac{20 \times 25}{20 \times 37} \times 100 = 67.6\%$$

Discussion of results of all three cases: No control, meter-in and meter-out.
As can be seen, meter-out is marginally more efficient than meter-in owing to the ratio of piston to piston rod area. Both systems are equally efficient when used with through-rod cylinders or hydraulic motors. It must be remembered that meter-out should prevent any tendency of the load to run away.

In both cases, if the system runs light, that is, extends against a low load, excessive heat is generated over the flow controls in addition to the heat generated over the relief valve. Consequently, there is further reduction in the efficiency. Also, in these circumstances, with meter-out flow control, very high pressure intensification can occur on the annulus side of the cylinder and within the pipe work between the cylinder and the flow-control valve. Take a situation where meter-out circuit is just considered. The load on extend is reduced to 5 kN without any corresponding reduction in the relief-valve settings.

Flow into the full bore end is 3.91 L/min.
Therefore, excess flow from the pump is

$$20 - 3.93 = 16.07 \text{ L/min}$$

that passes over the relief valve at 152 bar.
The pressure at the full bore end of the cylinder is = 152 – 3 – 2 = 147 bar
This exerts a force that is resisted by the load and the reactive back pressure on the annulus side:

$$147 - \left(\frac{5 \times 10^3}{7.85 \times 10^{-3} \times 10^5}\right) = (2 + 10 + p) \times \frac{4.00}{7.85}$$

where$p$ is the pressure within the annulus side of the cylinder and between the cylinder ant the flow-control valve. So

$$p = [(147 - 6.4) \times 7.85/4.00] - 12 = 264 \text{ bar}$$

The system efficiency on extend is

$$\frac{3.93 \times 6.4}{20 \times 152} \times 100 = 0.83\%$$

Almost all of the input power is wasted and dissipates as heat into the fluid, mainly across the relief and flow-control valves.

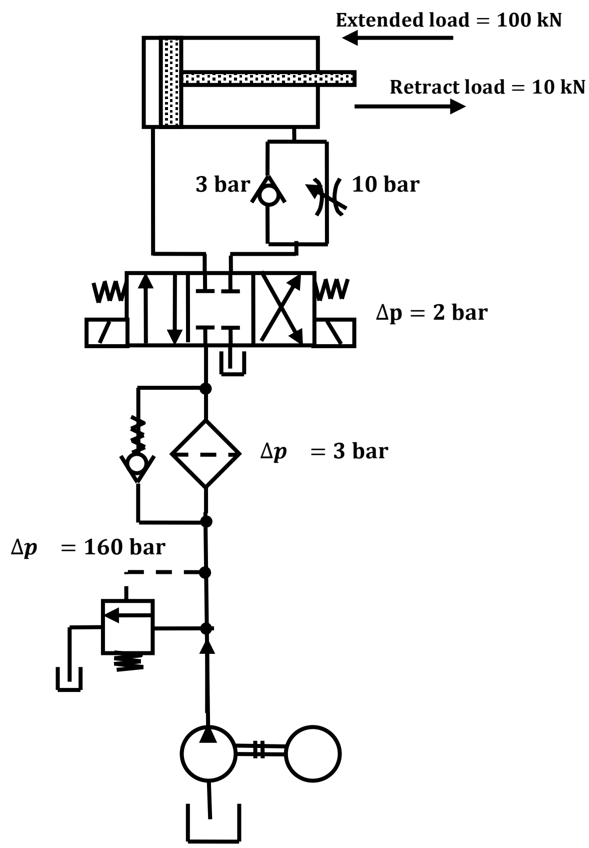


Figure 1.15 Hydrauliccylinder with meter-out control.

Example 1.6

Figure 1.16 shows a hydraulic circuit where the actuator speed is controlled by a meter-in system employing a series pressure-compensated valve. Determine the power input to the pump under a steady-state condition. If the series compensation is replaced by parallel compensation, and the load and speed of the actuator remain unchanged, determine the change of overall efficiency of the circuit.

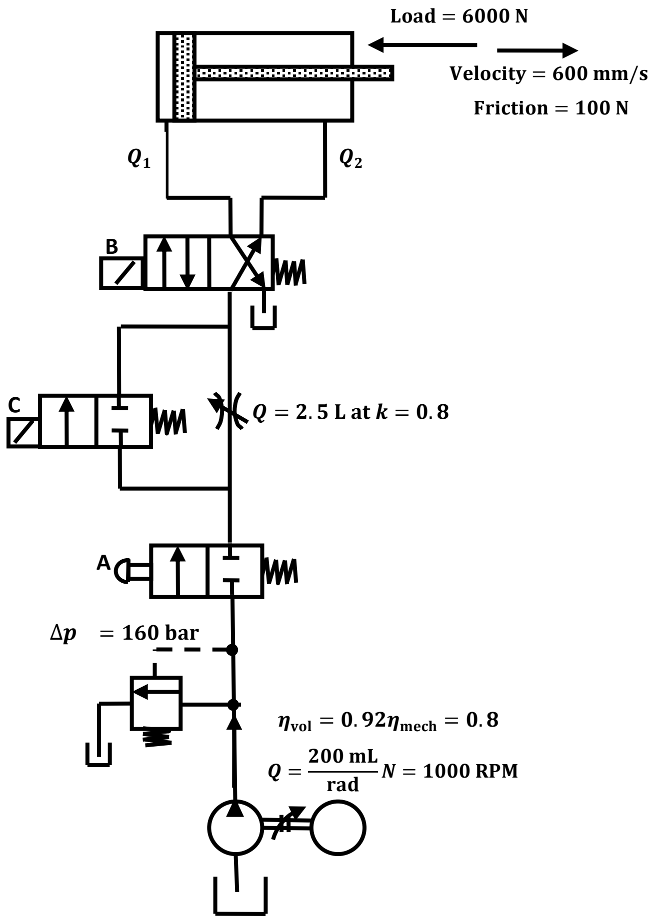


Figure 1.16 Hydraulic cylinder with a pressure-compensated valve.

Solution:

For valve A we have $Q = 0.5\sqrt{\Delta p}$

For valve B we have $Q = 0.4\sqrt{\Delta p}$

For valve C we have $Q = 0.3\sqrt{\Delta p}$

Now referring to Fig. 1.16, let us calculate the flow to piston side of the cylinder:

$$Q_1 = \frac{\pi}{4} \times 60^2 \times 600 = 1.7 \text{ L/s}$$

Flow from the return side of the cylinder is

$$Q_2 = 1.4 \text{ L/s}$$

Pump flow is given by

$$Q_p = 2\pi \times 1000/60 \times 20 = 2.1 \text{ L/s}$$

$$Q_3 = \eta_{vol} \times Q_p = 1.93 \text{ L/s}$$

Power input to the pump = $150 \times 10^5 \times 2.1 \times 10^{-3} \times 1/0.8$ W = 39.4 kW

Power output to the actuator = $6100/1000 \times 600/1000$ = 3.6 kW

Therefore, system efficiency = $3.66/39.4 \times 100$ = 9%

Pressure loss at valve B due to $Q_2 = 1 \times (1.4/0.4)^2 = 12.2$ bar

Pressure at the head end of actuator

$$p \times \frac{\pi}{4} \times 60^2 \times 10^{-6} = 6100 + 12.2 \times 10^5 \times \frac{\pi}{4} \times (602 - 252) \times 10^{-6}$$

$$\Rightarrow p = 29 \text{ bar}$$

Pressure losses at B due to $Q_1 = (1.7/0.4)^2 = 18$ bar
Pressure losses at valve A = $(1.7/0.5)^2 = 11.6$ bar
Therefore, the total pressure, excluding that lost in the pressure-compensated valve if it is of series type, is

$$29 + 18 + 11.6 + 4 = 62.6 \text{ bar}$$

Hence, 150 – 62.6 = 87.4 bar is dropped in the pressure-compensated valve if it is of series type.
For a parallel pressure-compensated valve, the excess oil $Q - Q_1$ would bypass at

$$62.6 \text{ bar} - 11.6 \text{ bar} = 51 \text{ bar}$$

The pump delivery would be at 62.6 bar and hence the total power consumption is

$$62.6 \times 10^5 \times 2.1 \times 10^{-3} \times 1/0.8 \text{ W} = 16.5 \text{ kW}$$

System efficiency = 22.2%

Example 1.7

A flow-control valve is used to control the speed of the actuator as shown in Fig. 1.17 and the characteristics of the system are given in Table 1.1. Determine the variable flow area A_v, the pressure downstream of the valve fixed orifice p_2, the valve displacement x and the spring preload F for the given motor operating conditions.

Table 1.1

Parameters	**Value**
Valve flow constant (C_d)	0.6
Length h	7.8 mm
Valve area gradient for flow area (A_v), b	1.25 mm^2/mm
Fixed orifice flow area (A_o)	4.9 mm^2
Valve face area (A)	125 mm^2
Spring constant	57 kN/m
Motor displacement (D_m)	40 cm^3/rev
Motor torque	60 Nm
Motor speed	350 RPM
Motor volumetric efficiency (η_v)	96 %
Motor mechanical efficiency (η_m)	97 5

System pressure (p_1)	14.5 MPa
Return pressure (p_4)	1 MPa
Fluid density	840 kg/m^3

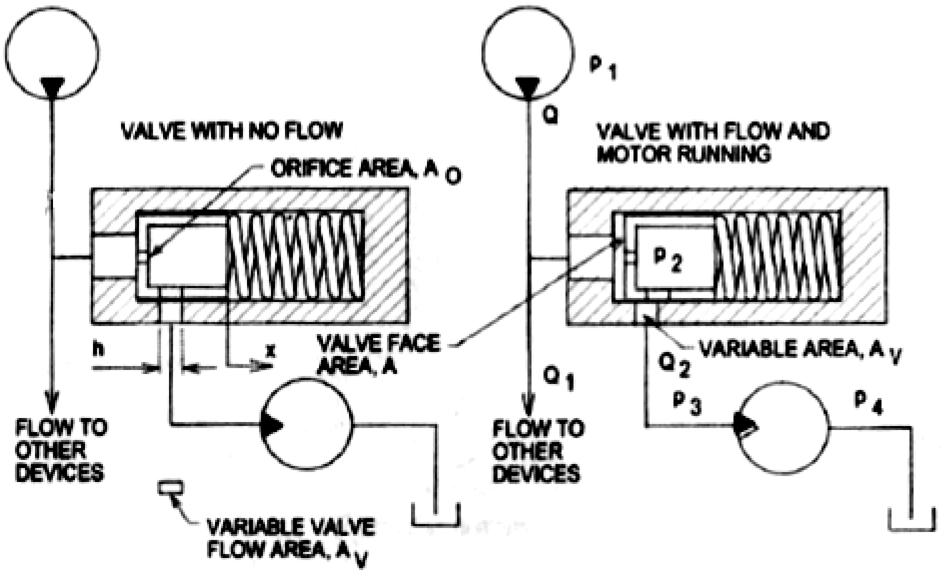


Figure 1.17

Solution: Refer to Fig. 1.17, flow from the pump divides as Q_1 and Q_2 . The pressure drop $p_1 - p_2$ occurs across orifice A_o. This makes the valve to move to right against the spring force *F*. The area of orifice A_v then adjusts to control the flow to the motor:

Let us first convert all the given variables to appropriate units

$$h = \frac{7.8}{1000}\ \text{m},\ w = \frac{1.25}{1000}\ \text{mm},\ k = 57000\ \text{N/m},\ A_o = 4.9\times10^{-6}\ \text{m}^2,\ A = 125\times10^{-6}\ \text{m}^2$$

$$D_m = \frac{\frac{40}{100^3}\ \text{m}^3}{\text{rev}},\ T = 60\ \text{N m},\ n = 350\,\text{RPM},\ \rho = 840\ \text{kg/m}^3,\ p_1 = 145000\times1000\ \text{N/m}^2$$

$$\eta_V = 96\%,\ \eta_m = 97\%,\ p_4 = 1000\times1000\ \text{Pa},\ \theta = \frac{2\times\pi\times350}{60} = 36.652\,\text{rad/s}$$

First let us calculate the discharge Q_2 through valve

$$Q_2 = \frac{\text{Motor displacement}}{\text{Revolution}} \times \frac{\text{Speed}}{\text{Volumetric efficiency}}$$
$$= \frac{(40/100^3)\ \text{m}^3}{0.96} \times \frac{350}{60}$$
$$= 2.431 \times 10^{-4}\ \text{m}^3/\text{s}$$

Pressure drop across motor is $p_3 - p_4$. So

$$p_3 - p_4 = \frac{\text{Torque}}{\text{Flow rate} \times \text{Mechanical efficiency}}$$

$$p_3 = \frac{T}{0.97} \times \frac{2\pi}{(40/100^3)\ \text{m}^3} + p_4$$
$$= \frac{60}{0.97} \times \frac{2\pi}{(40/100^3)\ \text{m}^3} + 10^6$$
$$= 1.072 \times 10^7\ \text{Pa}$$

Also flow through orifice from motor side is given by

$$Q_2 = c_d A_o \sqrt{\frac{2}{840}} \times \sqrt{p_2 - p_1}$$
$$2.431 \times 10^{-4} = 0.6 \times 4.9 \times 10^{-6} \sqrt{\frac{2}{840}} \times \sqrt{p_2 - 14500000}$$

Substituting all the known values we can get p_2

$$p_2 = 1.163 \times 10^7\ \text{Pa}$$

Also flow through orifice from pump side is given by

$$Q_2 = c_d A_V \sqrt{\frac{2}{840}} \times \sqrt{p_2 - p_3} = 0.6 \times A_V \sqrt{\frac{2}{840}} \times \sqrt{1.163 \times 10^7 - 1.072 \times 10^7}$$

Solving we get the value of A_V as

$$A_V = 8.688 \times 10^{-6}\ \text{m}^2 \text{ or } 8.688\,\text{mm}^2$$

But area of the valve is

$$A_V = (h - x)w$$

Solving we get

$$x = \frac{h \times w - A_V}{w} = 8.499 \times 10^{-4}\ \text{m}$$

Now let us write the force balance we get (see Fig. 1.18)

$$p_1 A = p_2 A + (K x + F)$$

Substituting all the values we get

$$14500 \times 1000 \text{ N/m}^2 \times 125 \times 10^{-6} = 1.163 \times 10^7 \times 125 \times 10^{-6} + 57000 \times 8.499 \times 10^{-4} + F$$

$$F = 310.4\,\text{N}$$

Figure 1.18

Objective-Type Questions

Fill in the Blanks

1. Non-pressure-compensated flow-control valves are used when the system pressure is relatively ________ and motoring speeds are not too critical.
2. A pressure compensator acts to keep the ________ nearly constant.
3. In ________ flow-control valve consists of a main spool and a compensator spool.
4. Speed control by controlling the rate of flow ________ the cylinder is called meter-in control.
5. In a meter-________ circuit, only the speed during the extend stroke is controlled.

State True or False

1. The speed of a piston can be defined accurately using non-pressure-compensated flow-control valves when the working load varies.
2. Speed control by controlling the rate of flow out of the cylinder is called meter-in control.
3. In a meter-in circuit, the extending speed of the cylinder is controlled whereas the retracing speed is not.
4. Compared to the meter-in and meter-out circuits, the bleed-off circuit is more commonly used.

5. In a meter-in circuit, flow coming from the pump into the cylinder is not controlled directly. However, the flow out of the cylinder is controlled using a flow-control valve.

Review Questions

1. What is the function of a flow-control valve?
2. What are the three ways of applying flow-control valves?
3. What is meant when a flow-control valve is said to be pressure compensated?
4. What is a meter-in circuit and where is it used?
5. What is a meter-out circuit and where is it used?
6. What are the advantages of a meter-in circuit?
7. What are the disadvantages of a meter-in circuit?
8. What are the advantages of a meter-out circuit?
9. What are the disadvantages of a meter-out circuit?
10. What are the advantages of a by-pass or bleed-off circuit?
11.What are the disadvantages of a by-pass or bleed-off circuit?
12. What is a modular valve and what are its benefits?
13. What is a hydraulic fuse?
14.What is the need for temperature compensation in a flow-control valve?
15. What is the difference between a hydraulic fuse and a pressure-relief valve?

Answers
Fill in the Blanks

1.Constant
2.Pressure drop
3.Pressure compensated
4.Into
5.Out

State True or False

1.False
2.False
3.True
4.False
5.False

Lecture 21

FLOW AND FORCE ANALYSIS OF VALVES

Learning Objectives

Upon completion of this chapter, the student should be able to:

- Understand the mathematical equation for flow through anoverlapped four-way spool valve.
- Understand the mathematical equation for flow through an underlappedfour-way spool valve.
- Understand the mathematical equation for flow through athree-way critical center valve.
- Understand the mathematical equation for flow through athree-way open center valve.
- Describe the working and construction of a flapper nozzle valve and performits mathematical analysis.
- Analyze poppet, single-stage relief and pressure-compensated valves.
- Carry out mathematical analysis of valves used inhydraulic systems.

1.1 Introduction

Most hydraulic servo mechanisms or other high-performance systems rely for their operation on the metering of fluid through a valve. This chapter deals with a linearized method of analysis for "four-way valves." They are called so because they have four connections, one for the supply pressure and another for the exhaust, and two control ports through both of which fluid may be metered, from the supply to either the system or the exhaust.

Metering valves are never fully open and their use is for accurately metering the flow of fluid through them. In this case of spool valves, longitudinal displacements of the spool are always small as compared with the spool's diameter.

1.2 Four-Way Spool Valves

A spool valve used for metering purposes controls flow rate by throttling.Each port in a valve that is partially closed by a land on the spool becomes a control throttle.

The rate of flow of fluid through such a valve depends on the spool displacement from the null position "*x*" and on the pressure upstream and downstream of the valve. One way of representing the flow rate *q* through a valve is

$$q = K_{\mathrm{q}} x - K_{\mathrm{c}} p_{\mathrm{m}} \tag{1.1}$$

where$q$is the volume flow rate of the oil, K_q is the flow gain, K_c is the pressureflow coefficient and $p_m$is the pressure difference across the load.Equation(1.1) implies that the flow rate is directly proportional to the valve opening and directly proportional to the pressure drop.

1.2.1 Critical Center Valve

In critical center valves, the lands of the spool are exactly of the same width as the annual ports of the valve body in the central or null position where the lands exactly cover the ports (Fig. 1.1).

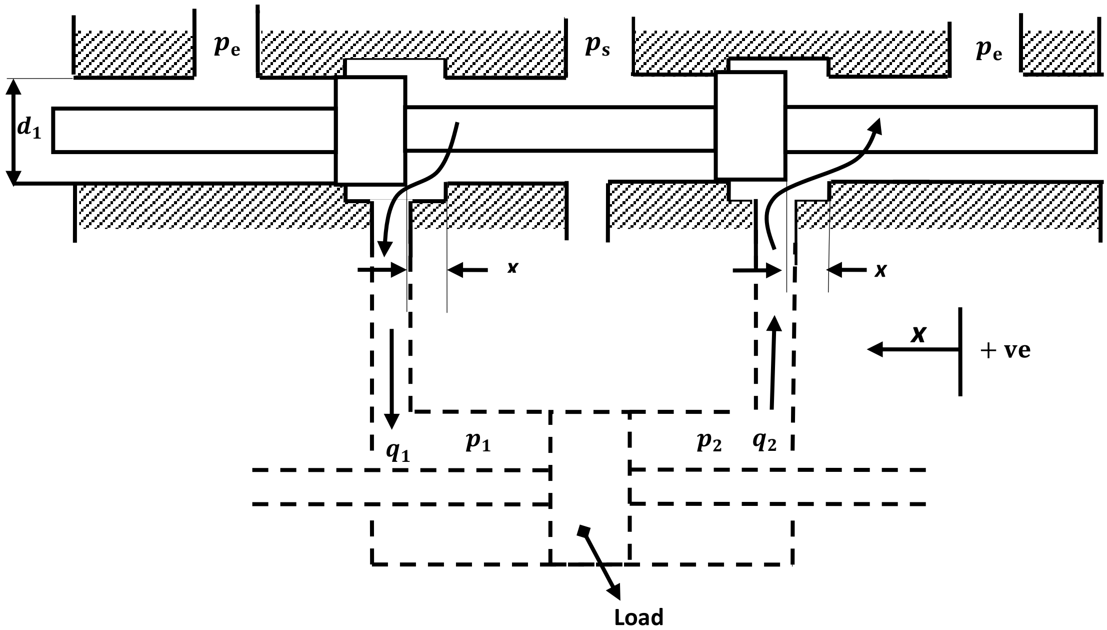


Figure 1.1 Four-way valve**.**

1.2.1.1 Flow Rate Prediction

Orifices are a basic means for the control of fluid power. Flow characteristics of orifices play a major role in the design of many hydraulic control devices. An orifice is a sudden restriction of short length in a flow passage and may have a fixed or variable area. Two types of flow regime exist depending on whether inertial or viscous forces dominate. The flow through orifice must increase above that in the upstream region to satisfy the law of continuity. At high Reynolds number, the pressure drop across the orifice is caused by the acceleration of the fluid particles from the upstream velocity to the higher jet velocity. At low Reynolds numbers, the pressure drop is caused by the internal shear forces resulting from fluid viscosity.

The pressure difference required to accelerate the fluid particles from the lower upstream velocity to higher upstream velocity is found by applying Bernoulli's equation and continuity equation. Here we will not derive the basic equations. Students are advised to go through any standard fluid mechanics books for the derivation of flow through orifice.

From fluid mechanics, flow through the orifice is given by

$$q_1 = C_d \pi d_1 x (p_s - p_1)^{1/2} (2/\rho)^{1/2} \tag{1.2}$$

$$q_2 = C_d \pi d_1 x (p_2 - p_e)^{1/2} (2/\rho)^{1/2} \tag{1.3}$$

Let us make following assumptions:

1. $q_1 = q_2$
2. $(p_s - p_1) = (p_2 - p_e)$
3. the supply pressure (p_s) is constant
4. p_e is negligible

By introducing the term $p_m = p_1 - p_2$, and using assumptions 2 and 4 we can write

$$(p_s - p_1) = (p_2 - p_e)$$
$$\Rightarrow (p_s - p_1) = (p_2 - 0)$$

$$\Rightarrow p_s = p_1 + p_2$$

Solving $p_s = p_1 + p_2$ and $p_m = p_1 - p_2$ simultaneously we get

$$p_s - p_m = 2p_2 \qquad (1.4)$$

$$p_s + p_m = 2p_1 \qquad (1.5)$$

Using all assumptions in Eqs. (1.2) and (1.3) we get

$$q = C_d \pi d_1 x (p_s - p_m)^{1/2} (1/\rho)^{1/2} \qquad (1.6)$$

With the type of configuration illustrated in Fig. 1.1, it is usually accepted that

$$C_d = 5/8 \qquad (1.7)$$

$$\rho = 870 \text{ kg/m}^3 \qquad (1.8)$$

Using values given in Eqs. (1.7) and (1.8) in Eq. (1.6) we get

$$q = 6.7\pi d_1 x (p_s - p_m)^{1/2} \qquad (1.9)$$

The assumption that the valve opening can be treated as orifices pre-supposes that x is small compared to d_1 so that the pressure drop across each orifice will be significant compared with p_s. In practice the pressure difference across the load p_m rarely exceeds 0.666 p_s. Allowing further simplification of Eq. (1.9) if $p_m < 0.666\, p_s$, then less than 10% error is involved using the binomial approximation. Using binomial approximation, we have

$$\sqrt{p_s - p_m} = \sqrt{p_s} \times \sqrt{1 - \left(\frac{p_m}{p_s}\right)} = \sqrt{p_s} \times \left[1 - \frac{1}{2}\left(\frac{p_m}{p_s}\right)\right] \qquad (1.10)$$

Hence Eq. (1.9) becomes

$$q = 6.7\pi d_1 x p_s^{1/2} - 6.7\pi d_1 x p_s^{1/2} p_m / 2p_s \qquad (1.11)$$

andEq. (1.11) is similar in form to Eq. (1.1). Now

$$K_q = 6.7\pi d_1 p_s^{1/2} \qquad (1.12)$$

$$K_c = 6.7\pi d_1 x p_s^{1/2} / 2p_s \qquad (1.13)$$

The above analysis predicts that the flow gain K_q can be treated as constant for a particular valve and supply pressure but the pressure flow coefficient K_c varies with the valve opening *x*. The variation of K_c is of minor significance for linear analysis.

In reality, spool lands never exactly match the annular ports in the valve body. Actual test results with a constant pressure drop across the valve ports show variations, particularly near the central or null position of the spool as those illustrated in Fig. 1.2.
The flow gain K_q is the slope of the approximate line in the figure, which can double its valve near null with negative lap. The magnitude of K_q is the most important parameter of a valve and often also of any system incorporating the valve.

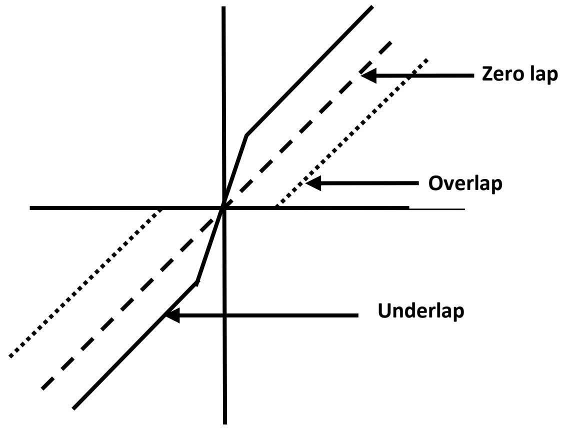


Figure 1.2 Flow rates versus valve displacement for constant pressure drop.

1.2.2 Open Center Valve (Underlapped Four-Way Valve)

A valve in which the land of the spool never completely covers the ports of the valve body is said to be underlapped(or to have negative lap)(Fig. 1.3).

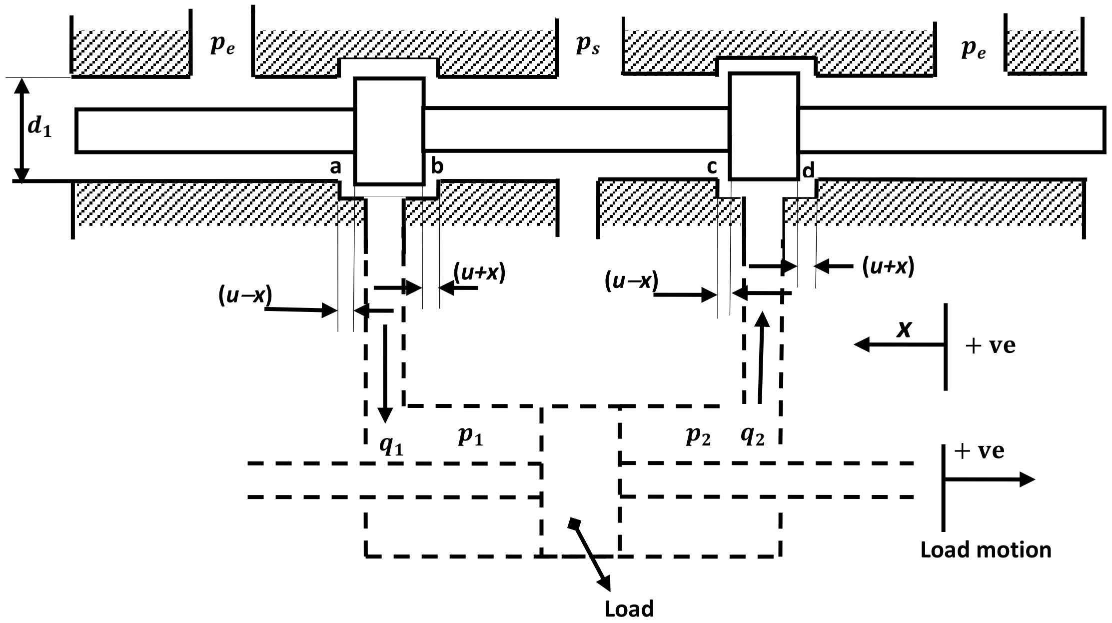


Figure 1.3 Open center–four-way valve.

Referring to Fig. 1.3,a displacement of x(say to the left) unbalances the symmetry of the ports. Two of the annular orifices increase in width from u to $u + x$ and two decrease from u to $u - x$.The flow ratesare estimated as follows:

$$q_1 = C_d \pi d_1 (u+x)(p_s - p_1)^{1/2} \left(\frac{2}{\rho}\right)^{1/2} - C_d \pi d_1 (u-x)(p_1 - p_e)^{1/2} \left(\frac{2}{\rho}\right)^{1/2} \quad (1.14)$$

$$q_2 = C_d \pi d_1 (u+x)(p_2 - p_e)^{1/2} \left(\frac{2}{\rho}\right)^{1/2} - C_d \pi d_1 (u-x)(p_s - p_2)^{1/2} \left(\frac{2}{\rho}\right)^{1/2} \quad (1.15)$$

Let us make following assumptions:

1. $q_1 = q_2$

2. $p_s = p_1 + p_2$

3. The supply pressure (p_s) is constant
4. p_e is negligible

Writing $p_m = p_1 - p_2$, we can use these assumptions in Eqs. (1.14) and (1.15) and get

$$q = C_d \pi d_1 \{(u+x)(p_s - p_m)^{1/2} - (u-x)(p_s + p_m)^{1/2}\} \left(\frac{1}{\rho}\right)^{1/2} \quad (1.16)$$

that may be approximated as

$$q_1 = C_d \pi d_1 \left\{ (u+x)(p_s)^{1/2} \left(1 - \frac{1}{2}\frac{p_m}{p_s}\right)^{1/2} - (u-x)(p_s)^{1/2} \left(1 + \frac{1}{2}\frac{p_m}{p_s}\right)^{1/2} \right\} \left(\frac{1}{\rho}\right)^{1/2} \quad (1.17)$$

$$q = C_d \pi d_1 \left(\frac{1}{\rho}\right)^{1/2} p_s^{1/2} 2x - C_d \pi d_1 u \left(\frac{1}{\rho}\right)^{1/2} \frac{p_s^{1/2} p_m}{p_s} \quad (1.18)$$

Equation (1.18) is similar in form to Eq. (1.1):

$$K_q = 13.4 \pi d_1 p_s^{1/2} \quad (1.19)$$

$$K_c = 6.7 \pi d_1 x p_s^{1/2} / p_s \quad (1.20)$$

The values refer to operation within the underlap region. Outside this region, these valves act as critical center valves with only two active ports. Note particularly that the flow gain K_q is double that for a comparable critical center valve (Fig. 152) in the underlap region. Note also significant leakage flow when the valve is centered (leakage flow at null when the load flow q is zero)becomes $13.4 \pi d_1 p_s^{1/2}$.

1.3 Three-Way Spool Valves

Three-way valves have only one critical length dimension which helps to ease manufacture. However they cannot be used for hydraulic motors requiring flow reversal and are usually used in differential ram and is discussed below.

1.3.1 Critical Center Valve

The central spool position just closes both the supply and the exhaust port of the valve. A displacement of the spool to the left causes fluid to be metered into the ram chamber from the supply, whereas one to the right causes metering of fluid from the chamber to the exhaust (Fig. 1.4).

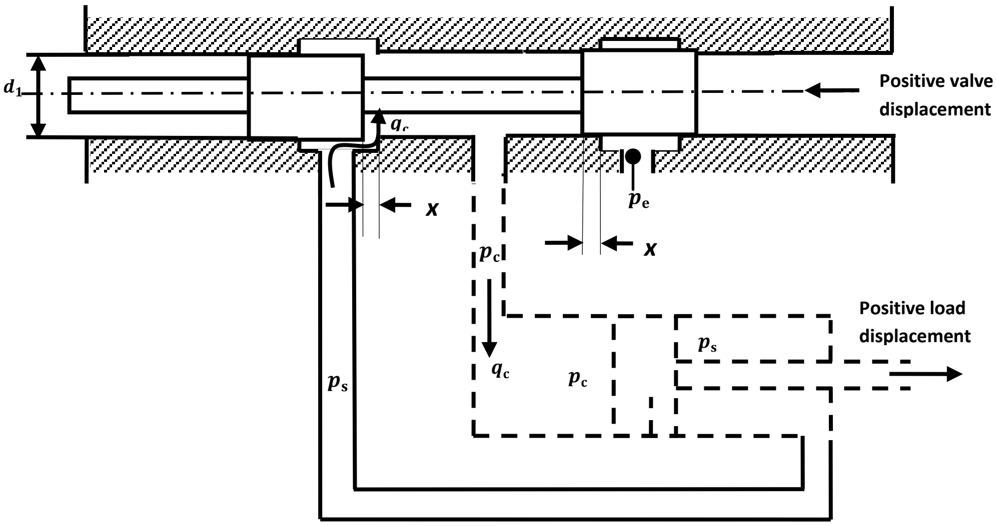


Figure 1.4 Three-way valve.

Oil flow rate is

$$q_c = C_d \pi\, d_1 x\,(p_s - p_c)^{1/2}\left(\frac{2}{\rho}\right)^{1/2} \text{ for} x \text{positive} \qquad (1.21)$$

$$q_c = C_d \pi\, d_1 x\,(p_c - p_e)^{1/2}\left(\frac{2}{\rho}\right)^{1/2} \text{ for } x \text{ negative} \qquad (1.22)$$

Substituting $p_m' = p_c - p_s/2$ we get

$$q_c = C_d \pi\, d_1 x\left(\frac{p_s}{2} - p_m'\right)^{1/2}\left(\frac{2}{\rho}\right)^{1/2} \text{ for } x \text{ positive} \qquad (1.23)$$

$$q_c = C_d \pi\, d_1 x\left(\frac{p_s}{2} + p_m'\right)^{1/2}\left(\frac{2}{\rho}\right)^{1/2} \text{ for } x \text{ negative} \qquad (1.24)$$

thatmay be approximated.Noting that

$$\left(\frac{p_s}{2} \pm p_m'\right)^{1/2} = \left\{\frac{p_s}{2}\left(1 \pm \frac{2p_m'}{p_s}\right)\right\}^{1/2} \approx \left(\frac{p_s}{2}\right)^{1/2}\left(1 \pm \frac{1}{2}\frac{2p_m'}{p_s}\right) \qquad (1.25)$$

we get

$$q_c = C_d \pi\, d_1 x\,(p_s)^{1/2}\left(\frac{1}{\rho}\right)^{1/2} \pm C_d \pi\, d_1 x\,(p_s)^{1/2}\,\frac{1}{p_s}\,p_m' \qquad (1.25)$$

where the positive sign is associated with negative values of x and negative sign with positive ones.Using $C_d(1/\rho)^{1/2} = 6.7$ for SI units with pressure in bars we get

$K_q = 6.7\,\pi\, d_1\,(p_s)^{1/2}$ (1.26)

$$K_c = 6.7\,\pi\, d_1 |x| \frac{\sqrt{\rho}\;\, p_m'}{\sqrt{p_s}} \text{ (1.27)}$$

1.3.2 Open Center Valve (Underlapped Three-Way Valve)

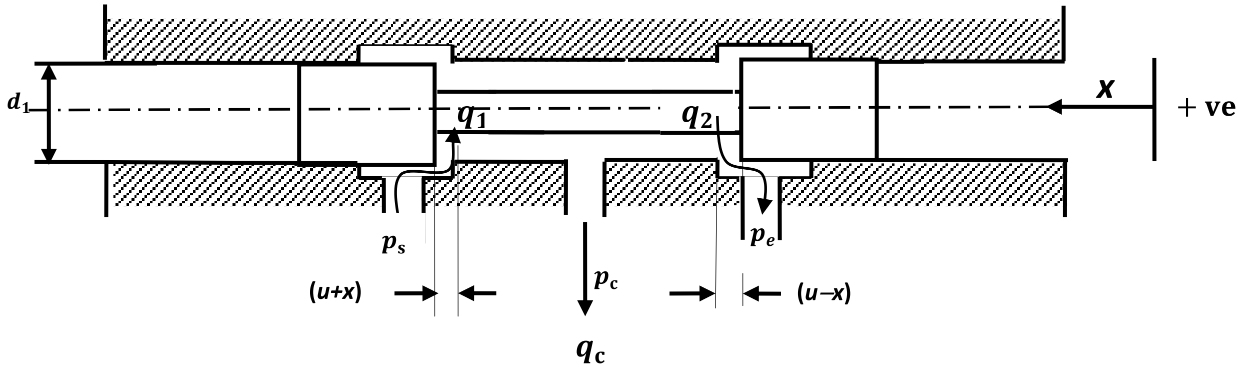


Figure 1.5 Open center–three-way valve.

Referring to Fig. 1.5, The orifice equation for q_1 can be written as

$$q_1 = C_d\pi\, d_1(u + x)\,(p_s - p_c)^{1/2}\left(\frac{2}{\rho}\right)^{1/2} \qquad (1.28)$$

Using binomial approximation we can write

$$q_2 \approx C_d\pi\, d_1(u + x)\left(p_s\right)^{1/2}\left(\frac{1}{\rho}\right)^{1/2}\left(1-\frac{p_m}{p_s}\right) \qquad (1.29)$$

Similarly orifice equation for q_2 can be written as

$$q_2 = C_d\pi\, d_1(u - x)\,(p_c)^{1/2}\left(\frac{2}{\rho}\right)^{1/2} \qquad (1.30)$$

Using binomial approximation we can write

$$q_2 \approx C_d\pi\, d_1(u - x)\left(p_s\right)^{1/2}\left(\frac{1}{\rho}\right)^{1/2}\left(1+\frac{p_m}{p_s}\right) \qquad (1.31)$$

The flow rateq_c is the difference of q_1 and q_2, that is

$$\begin{aligned} q_c &= q_1 - q_2 \\ &\approx C_d\pi\, d_1\left(p_s\right)^{1/2}\left(\frac{1}{\rho}\right)^{1/2} 2x - C_d\pi\, d_1\left(p_s\right)^{1/2}\left(\frac{1}{\rho}\right)^{1/2}\frac{2u}{p_s}\, p_m \qquad (1.32)\end{aligned}$$

Comparing with Eq. (1.1) we can write

$$K_q = 13.4\,\pi\, d_1\left(p_s\right)^{1/2} \quad (1.33)$$

$$K_q = 13.4\,\pi\, d_1 u\frac{\left(p_s\right)^{1/2}}{p_s} \qquad (1.34)$$

1.4 Flapper Nozzle Valve

Commonly used asthe first stage of two-stage servo valves, nozzles and the fixed upstream orifices used with them are made with diameters d_n and d_0 in the range 0.2–0.8 mm and the distance x_0 between each nozzle and the flapper in a double valve is often less than 0.2 mm.Each nozzle and orifice is as nearly a sharp-edged orifice as possible and treated as such for analytical purposes. The curtain area formed by the flapper at a nozzle exit modulates the control pressure caused by the fixed upstream orifice.

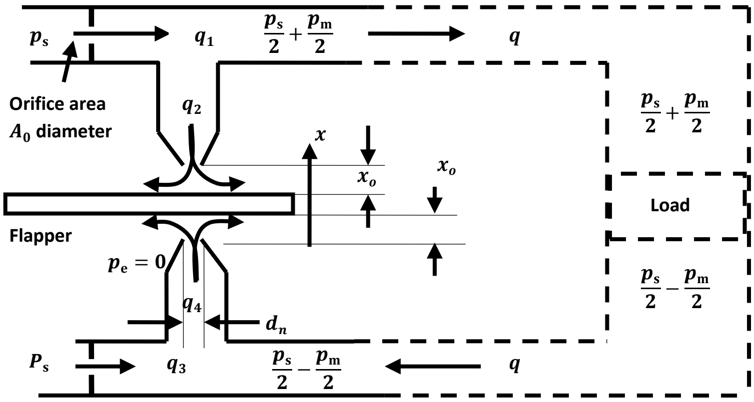


Figure 1.6 Flapper nozzle valve.

Consider a flopper nozzle valve indicated in Fig. 1.6. We assume that the valve has a balanced condition such that x = 0 and p_m = 0, q = 0. This occurs with the pressure downstream of each of the fixed orifices iequal top_s/2 and when the flow through each orifice equals that through each nozzle, that is

$$q_1 \text{ (steady state)} = q_3 \text{ (steady state)} = C_{d_0} A_0 (p_s - p_s/2)^{1/2}(2/\rho)^{1/2} \quad (1.35)$$

$$q_2 \text{ (steady state)} = q_4 \text{ (steady state)} = C_{d_n} \pi d_n x_0 (p_s/2)^{1/2}(2/\rho)^{1/2} \quad (1.36)$$

and
$$q_1 = q_2 = q_3 = q_4$$

This also implies that the orifice size and the curtain area in the null position are approximately equal

$$C_{d_0} A_0 = C_{d_n} \pi d_n x_0$$

Let us assume feasible ratio of

$$d_n = \left(\frac{3}{2}\right) d_o \Rightarrow d_n = 8x_0 \quad (1.37)$$

Consideringthat the valve is not in balance, that is,x has the same value as p_m has, we have

$$q = q_1 - q_2$$

$$= K_n \left[p_s - \left(\frac{p_s}{2} + \frac{p_m}{2} \right) \right]^{1/2} - K_n \left(\frac{x_0 + x}{x_0} \right) \left(\frac{p_s}{2} + \frac{p_m}{2} \right)^{1/2} \quad (1.38)$$

where

$$K_n = C_{d_0} A_0 (2/\rho)^{1/2} = C_{d_n} \pi d_n x_0 (2/\rho)^{1/2}$$

Also

$$q = -q_3 + q_4$$

$$= -K_n \left[p_s - \left(\frac{p_s}{2} - \frac{p_m}{2} \right) \right]^{1/2} + K_n \left(\frac{x_0 + x}{x_0} \right) \left(\frac{p_s}{2} - \frac{p_m}{2} \right)^{1/2} \quad (1.39)$$

Using binomial approximations, we have

$$q = K_{\mathrm{n}}\left(\frac{p_{\mathrm{s}}}{2}\right)^{1/2}\left(1-\frac{p_{\mathrm{m}}}{2p_{\mathrm{s}}}\right) - K_{\mathrm{n}}\left(\frac{p_{\mathrm{s}}}{2}\right)^{1/2}\left(1+\frac{p_{\mathrm{m}}}{2p_{\mathrm{s}}}\right) + K_{\mathrm{n}}\left(\frac{p_{\mathrm{s}}}{2}\right)^{1/2}\left(\frac{x}{x_0}\right)\left(\frac{p_{\mathrm{s}}}{2}+\frac{p_{\mathrm{m}}}{2}\right) \quad (1.40)$$

$$q = -K_{\mathrm{n}}\left(\frac{p_{\mathrm{s}}}{2}\right)^{1/2}\left(1+\frac{p_{\mathrm{m}}}{2p_{\mathrm{s}}}\right) + K_{\mathrm{n}}\left(\frac{p_{\mathrm{s}}}{2}\right)^{1/2}\left(1-\frac{p_{\mathrm{m}}}{2p_{\mathrm{s}}}\right) + K_{\mathrm{n}}\left(\frac{p_{\mathrm{s}}}{2}\right)^{1/2}\left(\frac{x}{x_0}\right)\left(\frac{p_{\mathrm{s}}}{2}-\frac{p_{\mathrm{m}}}{2}\right) \quad (1.41)$$

Adding and dividing by 2, we get

$$q = -K_{\mathrm{n}}(p_{\mathrm{s}}/2)^{1/2}\, p_{\mathrm{m}}/p_{\mathrm{s}} + K_{\mathrm{n}}(x/x_0)(p_{\mathrm{s}}/2)^{1/2} \quad (1.42)$$

thatis in the form of $q = K_{\mathrm{q}}x - K_{\mathrm{c}}p_{\mathrm{m}}$. Now

$$K_{\mathrm{q}} = K_{\mathrm{n}}/x_0(p_{\mathrm{s}}/2)^{1/2} = C_{d_{\mathrm{n}}}\pi d_{\mathrm{n}}(2/\rho)^{1/2}(p_{\mathrm{s}}/2)^{1/2}$$

$$K_{\mathrm{c}} = K_{\mathrm{n}}/p_{\mathrm{s}}(p_{\mathrm{s}}/2)^{1/2} = (C_{d_{\mathrm{n}}}\pi d_{\mathrm{n}}/p_{\mathrm{s}})(2/\rho)^{1/2}(p_{\mathrm{s}}/2)^{1/2}$$

$$= (C_{d_0}A_0/p_{\mathrm{s}})(2/\rho)^{1/2}(p_{\mathrm{s}}/2)^{1/2}$$

1.5 Special-Purpose Valves

The dynamic characteristics of hydraulic devices such as relief valves or flow-control valves are not well understood. For design purposes, it is useful to know the likely influence of spool mass, spring rate, orifice size or other parameter on the response of any particular valve to changes in pressure or flow rate or to other disturbances. A pressure-control valve and a flow-control valve are considered below.

1.5.1 Poppet Valves

Pressure control valve and flow control valves employs a poppet valve (Fig. 1.7) and their characteristics will be influenced by the flow pattern existing at or near the valve seat.

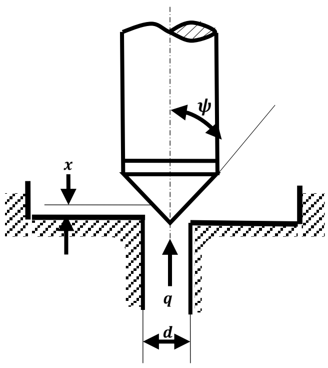


Figure 1.7 Poppet valve.

For a poppet displacement x, the area of flow is $C_d x\, \pi\, d \sin\psi$, where C_d is the flow coefficient. For a pressure drop of Δp across the orifice formed between the seat and poppet, the fluid velocity v is taken as equal to

$$\left(\frac{2\Delta p}{\rho}\right)^{1/2} \quad (1.43)$$

and the volume flow rate as

$$q = C_d x\, \pi\, d \sin\psi\left(\frac{2\Delta p}{\rho}\right)^{1/2} \quad (1.44)$$

The momentum of the jet has an axial component equal to $\dot{m}v\cos\psi$ (where $\dot{m}$ is the mass flow rate) that may be written as

Axial component of momentum of jet ($\mathbf{m}_{axial}$) = Mass flow rate ×Velocity

= Density ×Volume flow rate ×Velocity component

$$(m_{\text{axial}}) = \rho\left[C_d x\ \pi\ d \sin\psi\left(\frac{2\Delta p}{\rho}\right)^{1/2}\right]\left\{\left(\frac{2\Delta p}{\rho}\right)^{1/2}\cos\psi\right\} \qquad (1.45)$$

Please checkEq. (1.45)

where

$$\lambda = C_d\pi\, d\ \sin 2\psi \qquad (1.46)$$

The axial component of jet momentum is $\lambda\Delta p\, x$.

1.5.2 Single-Stage Relief Valve

Spring loaded valve is illustrated in Fig. 1.8. Let the pressure applied to valve to open is pIt is the effects of small changes in this pressure from p_o to $p_o + \bar{p}$ that are to be considered. The outlet (exhaust) pressure is assumed to be zero throughout.

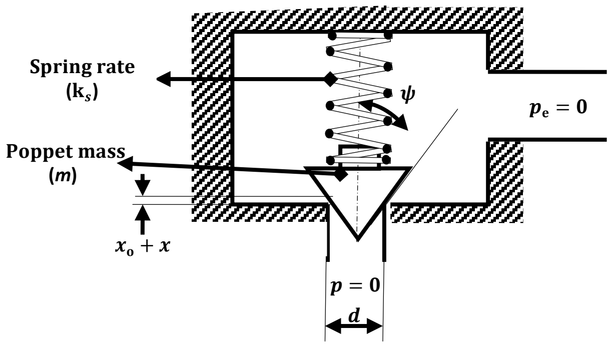


Figure 1.8 Relief valve.

With the poppet closed (i.e.,x = 0), there is some spring-related preload force holding it down that is designated as F. Under a steady-state condition with a greater pressure applied than that needed to overcome the preload, the valve partly opened and the poppet stationary, there is a steady-state balance of forces relating this pressure to the poppet displacement. For the applied pressure p_0 and the poppet displacement x_0, the relation is

$$p_o \frac{\pi d^2}{4} = F + k_s x_0 + \lambda\ p_o x_0 \qquad (1.47)$$

If the approach velocity is negligible then λ, given in Eq. (1.46), and the fluid frictional drag force across the poppet face is neglected.

For some other steady-state position of the poppet (displacement $x_o + \bar{x}$), another pressure $p_o + \bar{p}$ would occur according to

$$(p_o + \bar{p})\ \frac{\pi d^2}{4} = F + k_s(x_o + \bar{x}) + \lambda\ (p_o + \bar{p})(x_o + \bar{x}) \qquad (1.48)$$

Note that Eqs. (1.47) and (1.48) refer to steady state with poppet stationary. Under dynamic conditions with the poppet moving, the balance of forces has to take into account the effective mass of the poppet m and any damping (assumed viscous of rate f) as well as spring, pressure and momentum forces. For a pressure $p_o + \bar{p}$ and a poppet displacement $x_o + \bar{x}$, the balance forces become

$$(p_o + \bar{p})\ \frac{\pi d^2}{4} = F + k_s x_0 + \lambda\ p_o x_0 + \lambda x_0 \bar{p}\ + (k_s + \lambda p_0)\, x + f\,Dx + mD^2x \qquad (1.49)$$

Assume that the terms involving $\bar{p}x$ are negligibly small. Subtracting Eq. (1.47) from Eq. (1.49), we get

$$\bar{p}\frac{\pi d^2}{4} - \bar{p}\lambda x_0 = (k_s + \lambda\ p_o)\ x + f\,Dx + mD^2x$$

Indicating a second-order relation between changes in pressure $\bar{p}$ and changes in poppet displacement x given by

$$\frac{x}{\bar{p}} = \frac{K}{\left(\frac{1}{\omega_n{}^2}\right)D^2 + \left(\frac{2\xi}{\omega_n}\right)D + 1} \tag{1.50}$$

where

$$K = \frac{(\pi d^2\ /\ 4) - \lambda\, x_0}{(ks + \lambda\, p_o)}$$

$$\omega_n^2 = (k_s + \lambda\ p_o)\ /m$$

This relation suggests that any oscillation of the poppet is associated with much stiffer spring than the physical spring constant would suggest. As a numerical example, consider a valve of diameter d = 6mm for use at a nominal pressure of p_o = 70 bar. The projected area equals 28 mm^2, so that the value of F is approximately 20 N andif this is obtained by initially compressing the spring 10mm, the spring rate would have a valve of k_s = 20 N/mm. Assume that the poppet has a 90° cone angle and flow coefficient c_d = 0.7, then λ for the valve is about 13.2 mm and $\lambda\ p_o$ = 92.4 N/mm. The effective spring rate is not 20 but 112.4 N/mm. If the effective mass were 0.001 kg, then the natural frequency ω_n of poppet oscillations would be 533 Hz.

1.6 Pressure-Compensated Flow-Control Valve

Figure 1.9 illustrates a pressure-compensated flow-control valve which is designed to pass a constant flow rate of fluid despite fluctuations of the inlet and outlet pressures. The device has two orifices in series:one is preset manually to select the desired flow rate, while the other varies with the pressure difference across the valve. The aim is to keep the flow rate constant by maintaining a constant pressure difference across the present orifice.

For analysis, changes in the outlet pressure p_L represent the disturbances externally imposed on the device with the inlet (supply) pressure p_s assumed constant. This simulates a meter-in control with the poppet valve partly open and this analysis concerns small changes in this valve opening. The fully open or fully closed conditions are not dealt with. The datum for spring force acting on the spool is taken from some arbitrary position of the spool represented by the poppet (i.e., control orifice) opening of x_o.

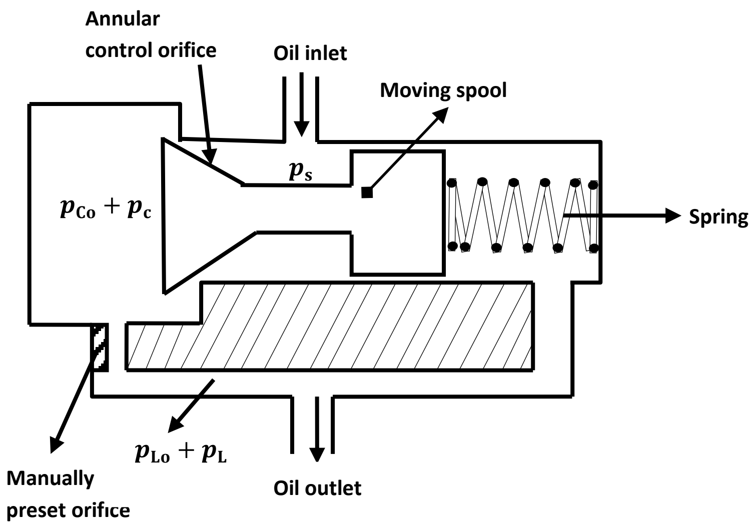


Figure 1.9 Pressure-compensated flow-control valve.

1.6.1 Forces

The steady-state balance of forces for some equilibrium operating position with the poppet stationary and open distance x_0, for a supply pressure p_s, outlet pressure p_{Lo} and consequent chamber pressure p_{Co} is given by

$$(p_{Co} - p_{Lo})A_p = F - \lambda(p_s - p_{Co})x_o \tag{1.51}$$

Under dynamic conditions with the spool in motion for the outlet pressure $p_{Lo} + p_L$, with the instantaneous spool position distance x to the left of its initial position, noting that $p_{Co} + p_c$ is the instantaneous chamber pressure, the balance of forces is given by

$$\{(p_{Co} + p_c) - (p_{Lo} + p_L\}A_p = F - k_s x - \lambda\{p_s - (p_{Co} + p_c)\}(x + x_o) - fDx - mD^2x \tag{1.52}$$

Assuming that p_c p_L and $x$are small and the terms $p_c x$ may be neglected, subtracting Eq. (1.52) from Eq. (1.51) gives

$$-p_c = \frac{k_s + \lambda(p_s - p_{Co})}{Ap - \lambda x_o}x + \frac{f}{Ap - \lambda x_o}Dx + \frac{m}{Ap - \lambda x_o}D^2x - \frac{Ap}{Ap - \lambda x_o}p_L \tag{1.53}$$

1.6.2 Flow Rates

Under steady-state condition, the flow rate through control orifice must be equal to that through the preset orifice. Hence, for the steady conditions previously used with the constant p_s, the outlet p_{Lo}, the chamber pressure p_{Co} and control orifice opening x_o using the subscript e for the preset orifice,

$$q_0 = C_d x_0 \pi d \left\{\frac{2(p_s - p_{Co})}{\rho}\right\}^{1/2} = C_{de} a_e \left\{\frac{2(p_{Co} - p_{Lo})}{\rho}\right\}^{1/2} \tag{1.54}$$

Under dynamic conditions, the flow rate through the control orifice will no longer be equal – the flow through the control orifice is $q_o + q_i$ and that through preset orifice $q_o + q_e$. The difference between the two flow rates (into and out from the valve chamber) causes compression of the oil in the valve chamber or

$$q_o + q_i = \frac{V}{\beta} D p_c \tag{1.55}$$

where V is the volume of oil and β is the effective bulk modules. In Eq. (1.55), V is assumed to be constant.The flow rate through the control orifice may be written as

$$q_o + q_i = Cd\,(x + x_o)\,\pi\, d\left(\frac{2}{\rho}\right)^{1/2}\{p_s - (p_{Co} + p_c)\}^2 + A_p Dx \quad (1.56)$$

Subtracting Eq.(1.54)from Eq. (1.56) neglecting the terms involving $p_c x$ we get

$$q_i = -\frac{q_o}{2}\frac{p_c}{p_s - p_{Co}} + \frac{q_o}{x_o}x + A_p Dx \quad (1.57)$$

and similarly for the change in flow rate through the preset orifice,

$$q_e = = \frac{q_o}{2}\frac{p_c}{p_s - p_{Lo}} - \frac{q_o}{2}\frac{p_L}{p_{Co} - p_{Lo}} \quad (1.58)$$

Now subtracting Eq. (1.58) from Eq.(1.57) gives

$$q_i - q_e = A_p Dx + \frac{q_o}{x_o}x + \frac{q_o}{2}\frac{p_L}{p_{Co} - p_{Lo}} - \frac{q_o}{2}\left(\frac{1}{p_s - p_{Co}} + \frac{1}{p_{Co} - p_{Lo}}\right)p_c \quad (1.59)$$

Also equating Eqs. (1.55) and (1.59) after substituting Eq. (1.53) for p_c and after differentiating, for $D\,p_c$ to obtain p_c and $D\,p_c$ terms of p_L and x leads to (closed loop) relation between displacements of the spool and changes of the outlet pressure

$$a_o D^3 x + a_1 D^2 x + a_2 D x + a_3 x = K\,p_L + TD\,p_L \quad (1.60)$$

$$\Rightarrow \frac{x}{p_L} = \frac{K + TD}{a_o D^3 + a_1 D^2 + a_2 D + a_3} \quad (1.61)$$

where

$$a_3 = \overline{k_s}\bar{Q} + \frac{q_o}{x_o}(A_p - \lambda x_o)$$

$$a_2 = f\bar{Q} + \overline{k_s}\left(\frac{V}{\beta}\right) + A_p(A_p - \lambda x_o)$$

$$a_1 = m\bar{Q} + f\left(\frac{V}{\beta}\right)$$

$$a_0 = m\left(\frac{V}{\beta}\right)$$

$$T = A_p\left(\frac{V}{\beta}\right)$$

$$K = A_p\frac{q_o}{2(p_s - p_{Co})} + \lambda x_o\frac{q_o}{2(p_{Co} - p_{Lo})}$$

$$\bar{Q} = \frac{q_o}{2(p_s - p_{Co})} + \frac{q_o}{2(p_{Co} - p_{Lo})}$$

$$\overline{k_s} = k_s + \lambda(P_s - P_{Co})$$

This linear equation is obviously difficult to apply particularly as the coefficients depend on the initial conditions that might be classed as the "normal" operating conditions. In practice,the valves of this type can be unsatisfactory at initial stage. The relation does indicate that a value is stable if $a_1 a_2$ is greater than $a_3 a_o$ and confirms that increasing the viscous damping helps to stabilize the valve, increasing both a_1 and a_2.

Example 1.1

A four-way valve with full periphery annular ports has a 6 mm diameter spool and it may be assumed that the spool lands fully cover the valve ports in zero or mid-position. Estimate the flow rate through one port when the pressure drop across it is 70 bar for every mm of spool displacement.

Solution

Spool diameter (d) = 6mm
Pressure drop (Δp) = 70 bar
Spool displacement (x) = 1mm
By the orifice flow equation, theflow rate is

$$q = C_{\mathrm{d}} \pi d x \sqrt{2\Delta p / \rho}$$

Letting C_{d} = 5/8 and ρ = 870 kg/m^3 and Δpin bar, we have

$$q = 6.7\pi d x \sqrt{2\Delta p}$$

Therefore,

$$q = 6.7 \times \pi(6 \times 10^{-3}) \times (1 \times 10^{-3}) \times \sqrt{2 \times 70}$$
$$\Rightarrow q = 1.494 \times 10^{-3} \text{ m}^3/\text{s}$$

Example1.2

What would be the flow coefficient K_{q} of the above valve if it is used as part of a servo system having oil supply pressure: (a) 140 bar, (b) 210 bar?

Solution:

(a) Supply pressure (p_{s}) = 140 bar. We know $K_{\mathrm{q}} = 6.7\pi d\sqrt{p_{\mathrm{s}}}$. Therefore,

$$K_{\mathrm{q}} = 6.7 \times \pi(6 \times 10^{-3}) \times \sqrt{140} = 1.494 \text{ m}^3/\text{s/mm}$$

(b)Supply pressure (p_{s}) = 210 bar. We know $K_{\mathrm{q}} = 6.7\pi d\sqrt{p_{\mathrm{s}}}$. Therefore,

$$K_{\mathrm{q}} = 6.7 \times \pi (6 \times 10^{-3}) \times \sqrt{210} = 1.83 \text{ m}^3/\text{s/mm}$$

Example1.3

A three-way spool valve with half the annular periphery of the valve port blocked off and spool diameter 9 mm is used in a system supplied with oil at 120 bar pressure. The half-area piston has areas 0.004 m^2 and 0.002 m^2 and a maximum required velocity 0.3 m/s. Estimate the maximum spool displacement required. (Assume one-third pressure drop through valve.)

Solution: Supply pressure (p_{s}) = 120 bar. Spool diameter (d) = 9mm. Piston area (A_{p}) = 0.004m^2.
Maximum velocity (v) = 0.3m/s. Maximum valve flow rate is

$$Q = A_{\mathrm{p}} \times v = 0.004 \times 0.3 = 1.2 \times 10^{-3} \text{ m}^3/\text{s}$$

Pressure difference across load (p_{m}) = 40 bar (one-third of supply pressure).
For a three-way valve with half-area piston, the pressure drop is

$$\Delta p = (p_{\mathrm{s}}/2) - p_{\mathrm{m}} = 20 \text{ bar}$$

Also from the orifice equation we can write

$q = 6.7 \times \pi\,(9 \times 10^{-3}) \times x \times \sqrt{2 \times 20}$

$\Rightarrow q = 1.198 \times x$

Equating two values of *Q and q*, we have

$$1.2 \times 10^{-3} = 1.198 \times x$$

$$\Rightarrow x = 1.0016\text{mm}.$$

As half the annular periphery of valve is blocked, for the same flow rate, the displacement required is twice the calculated value.Therefore, the valve displacement is

$$y = 2 \times x = 2.003\text{mm}$$

Example 1.4

In a 240-bar servo system employing a four-way valve, a valve underlap is used to assist in damping system oscillations. The valve has a 4-mm-diameter spool, full periphery ports and nominal underlap of 0.0127 mm. Estimate the pressureflow coefficient K_c for the valve.

Solution: Supply pressure (p_s) = 240 bar. Spool diameter (d) = 4mm. Underlap(u) = 0.0127mm. We know

$$K_c = \frac{6.7\,pdu}{\sqrt{p_s}} = \frac{6.7 \times \pi \times (4 \times 10^{-3})(0.0127 \times 10^{-3})}{\sqrt{240}} = 6.9 \times 10^{-9}\ \text{m}^3/\text{s/bar}$$

Objective-Type Questions
Fill in the Blanks

1. In critical center valves, the lands of the spool are of the ______ as the annual ports of the valve body in the central or null position where the lands exactly cover the ports.
2. A valve in which the land of the spool never completely covers the ports of the valve body is said to be ______.
3. ______ is commonly used as the first stage of two-stage servo valves.
4. A pressure-compensated flow control valve is designed to pass a ______ of fluid despite fluctuations of the inlet and outlet pressures.
5. Flow gain can be treated as ______ for a particular valve and supply pressure.

State True or False

1. The central spool position just closes both the supply and the exhaust port of the valve.
2. Most hydraulic servo mechanisms or other high-performance systems rely for their operation on the metering of fluid through a valve.
3. Metering valves are always fully open and their use is for accurately metering the flow of fluid through them.
4. A spool valve used for metering purposes controls flow rate by throttling.
5. A pressure flow coefficient will vary with valve opening and the variation of pressure flow coefficient are of major significance for linear analysis.

Review Questions

1. Discuss in detail the flow–displacement relationship of a critical center spool valve, giving its possible applications in hydraulic control valves.
2. Discuss in detail the flow forces acting on apressure-compensated flow control valve and derive an expression for these forces.
3. Derive an expression for the flow–displacement relations of anunderlapped four-way valve.
4. Derive an expression for the flow for a flapper nozzle valve.
5.What is the major limitation of a flapper nozzle amplifier and how it can be overcome?
6.Why is a feedback system used in a multistage servo valve?
7. What is a critical center valve?
8. What does an underlapped four-way valve mean?
9. Give two applications of a flapper nozzle valve.
10. How is a higher flow rate achieved in an electrohydraulic servo valve?
11. Why is a feedback system used in a multistage servo valve?
12. Illustrate the two different types of feedbacks used in a multistage electrohydraulic valve?
13. What is the major limitation of a flapper nozzle amplifier and how it can be overcome?

Answers
Fill in the Blanks

1.Same width
2.Underlapped
3.Flapper nozzle valve
4.Constant flow rate
5.Constant

State True or False

1.True
2.True
3.False
4.True
5.False

Lecture 22
PROPORTIONAL CONTROL VALVES

Learning Objectives

Upon completion of this chapter, the student should be able to:

- Appreciate the history of proportional control valves.
- Explain the operation of proportional solenoids.
- Describe various design considerations for proportional control valves.
- Describe various proportional direction and proportional pressure control valves.
- Explain the working of two-stage proportional valves.
- Compare a proportional valve and a servo valve.
- Describe the various applications of proportional valves.

1.1 Introduction

Proportional control valves can be operated easily using a solenoid. Solenoid controls have a digital control system: A valve is opened when the solenoid is energized and is closed when it is de-energized or vice versa. They are very quick in their operation and thus give rise to pressure and flow surges in the fluid power control units. If the control valves can be gradually opened or closed as a manually operated house tap, it results in a gradual transition between a fully opened and a fully closed position. These valves are operated by the application of electronics rather than just electrical switching. The advantage of these valves is that they give greater flexibility in the system design and operation. They also decrease fluid power circuit complexity especially for processes requiring multiple speed or force outputs.

1.2. History of Proportional Control Valves

A proportional valve is a valve that produces an output (direction, pressure, flow) that is proportional to an electronic control input. The output force exerted by the armature of a DC solenoid depends on the current flowing through it. This can be utilized in the design of a proportional DC solenoid in which the force exerted by the armature is proportional to the current flowing and independent of the armature over the working range of the solenoid.

In earlier days, there were only two types of electrically operated valves – solenoid valves and servo valves – that had a huge performance gap. Solenoid valves were either actuated or unactuated (i.e., fully open or fully closed) and had no intermediate position; thus, solenoid valves facilitated very little control. These were simply ON–OFF valves and their maximum frequency was 5 Hz or less. Servo valves, in contrast, were continuously controlled, high-frequency response devices that received commands through their electronic control systems that provided a high degree of control over position, velocity, acceleration, etc. They had high accuracy. They could accept and accurately respond to command signals at frequencies exceeding 100 Hz. The continuous feedback from electronic transducers ensured high accuracy. Between these extremes, there was nothing just a huge gap in performance, control capability and cost.

With the advent of proportional control valves, the situation changed. The design of their actuating device allowed the spool to be stopped at intermediate positions rather than only at the ends of the solenoid stroke. The associated electronics controlled the spool positions and offered a high degree of flexibility compared with the operation of the solenoid valve. The new valves had a maximum frequency response of 10 Hz that was better than solenoids but less when compared to servo valves. Thus, these valves were an intermediate between solenoids and servo valves. There was no feedback from the circuit, so the controllability and control accuracy were poor as compared to the parameters of the servo valves but greatly exceeded anything that could be achieved by solenoid valves. The final result was a valve that stood comfortably between a solenoid and a servo valve in performance, cost and complexity.

With the evolution of performance and application of proportional control valves, the efficiency increased. First a spool position feedback loop was added; next came improvements in the design of spools and the electronics; then came the external feedback systems, high-frequency responses, better performance in accuracy, hysteresis, dead band, threshold and other parameters. In short, the proportional valves began to look more and more like servo valves in capability. This was accompanied by an increase in cost and thus blurred the distinction between servo valves and proportional valves. As a result, performance and control are no longer distinguishing criteria. Rather physical features such as design and manufacturing processes are the defining characteristics. For instance, proportional valves are operated by proportional solenoids whereas servo valves are operated by torque motors. The spools in proportional valves are almost entirely machine produced, while the spools in servo valves require a great deal of manual lapping and finishing. The clearances and tolerances in servo valves are much tighter than in proportional valves. These differences mean that servo valves are still more expensive than proportional valves and also that they outperform proportional valves in terms of accuracy, hysteresis, leakage, etc. It is fair to say that a proportional valve can be linked to a low-cost, low-performance-range servo valve. These valves are divided into three types – directional, pressure and flow controls.

1.3 Proportional Solenoids

A directional control valve is the most common electrohydraulic proportional control valves (EHPV). The general aspects of its operation can also be applied to pressure and throttle valves. Though they look like solenoid valves, there are significant differences between the two. Both types have solenoids, and both have a valve body with a movable spool port and other components. We will look at the differences beginning with solenoids.

1.3.1 Proportional Solenoids

All standard solenoids have no intermediate positions; rather they are always at one end or the other of the solenoid stroke. The magnetic flux attempts to drive the plunger to its fully closed position when the coil is energized. The force developed by the solenoid is a function of square of the solenoid current and inverse function of square of the air gap. The result is that the force increases as the air gap closes as well as when the current increases. A typical force–displacement curve is shown in Fig. 1.1.

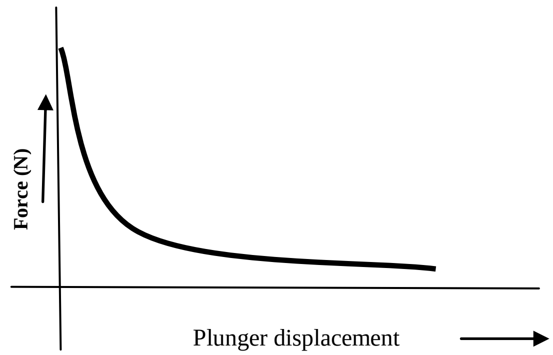


Figure 1.1 Solenoid force versus stroke atconstant current.

The solenoid force is at its minimum when the plunger is at the maximum position. By incrementally increasing the current in a particular solenoid, we can generate a family of curves (Fig. 1.2).

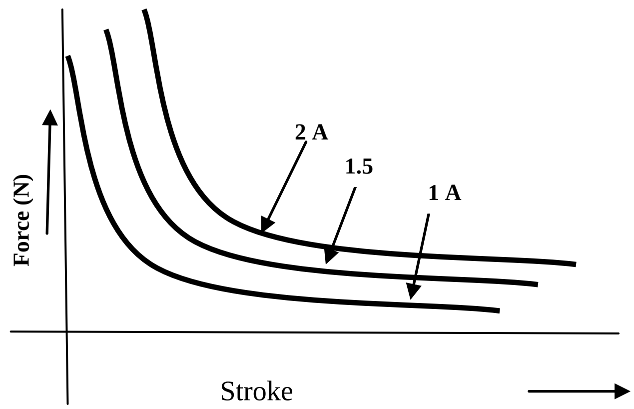


Figure 1.2 Solenoid force versus stoke curves with increasing current.

If there is a spring, then it is an additional design requirement. Let us assume a spring whose force is a linear function of its compressed distance ($F = kx$). If we plot the spring force versus air gap dimension, we get a graph shown in Fig. 1.3. From this plot, we can see that for a valve to operate at all, we must provide a current that ensures sufficient force to overcome the spring force throughout the plunger stroke. If the solenoid force ever drops below the spring force, the solenoid stops. The dependence of spring force on its compressed distance is shown in Fig. 1.3.

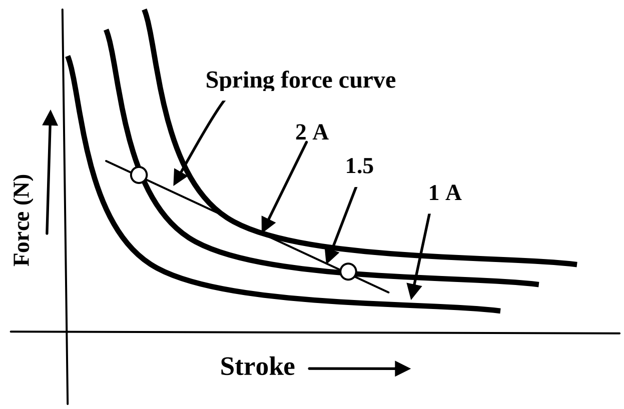


Figure 1.3 Solenoid force versus stroke with spring force.

A proportional solenoid differs from a standard solenoid in the design of the area near the end of the plunger stroke. In the design of a basic solenoid, the air gap is closed at a uniform rate as the plunger moves in. Because the square of the air gap dimension appears in the denominator of the force equation, the solenoid force increases exponentially as the gap closes. The design of a proportional solenoid eliminates the effect of diminishing air gap dimension at the end of the plunger stroke by utilizing constructional features that maintain a constant effective air gap or by using a magnetic impervious material to which the solenoid appears as a constant air gap. The results for any given current are a force curve similar to that in Fig. 1.4.

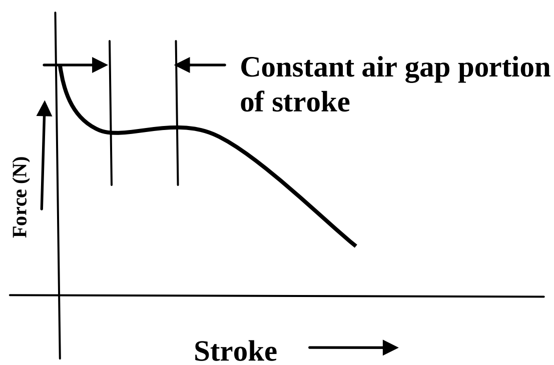


Figure 1.4 Proportional solenoid force versus stroke at constant current**.**

The flat portion of the curve occurs in the constant air-gap portion of the stroke. Figure 1.5 shows the variation of force versus stroke for various values of current.

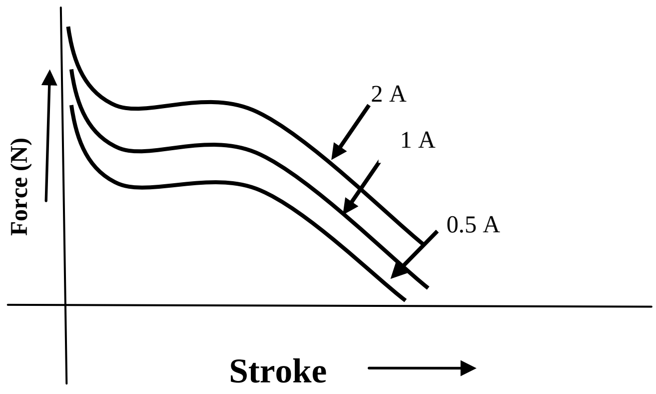


Figure 1.5 Proportional solenoid force versus stroke for varying current.

Use of a carefully designed, calibrated spring to oppose the solenoid force results in a solenoid force versus spring force arrangement as shown in Fig. 1.6. The concept is to have the spring force curve intersect the solenoid force lines in the flat portion of the solenoid force lines. Thus, the solenoid plunger position (and, subsequently, the valve spool position) can be controlled by the current applied. A higher current produces a higher solenoid force that compresses the spring until the spring force balances the solenoid force. When this force is balanced, the plunger stops. The position at which the plunger (spool) stops determines the size of the flow path through the valve. This along with the pressure differential across the valve determines the fluid flow rate through the valve. The functional result is that flow direction and flow rate can be controlled with a single valve. This portion of the solenoid stroke is known as the control zone. The length of this zone is only about 0.06–0.08 inch (0.15–0.20 cm). The total plunger stroke is also small, usually about 0.120 inch (0.3 cm).

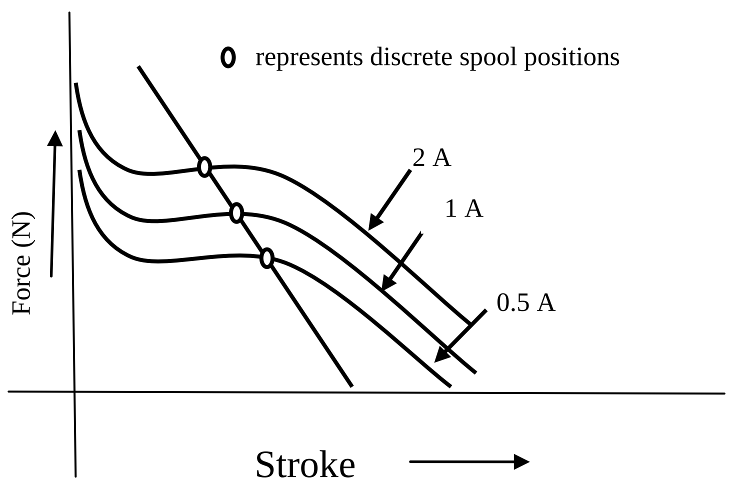


Figure 1.6 Proportional solenoid force versus stroke with spring force overlaid.

1.4 Design Considerations of Proportional Control Valves

The output force exerted by the armature of a DC solenoid depends on the current flowing through it (Fig. 1.7). This fundamental concept can be used in the design of a proportional DC solenoid in which the force exerted by the armature is proportional to the current flowing through it and independent of the armature movement over the working range of the solenoid. A typical characteristic is shown in Fig. 1.7.

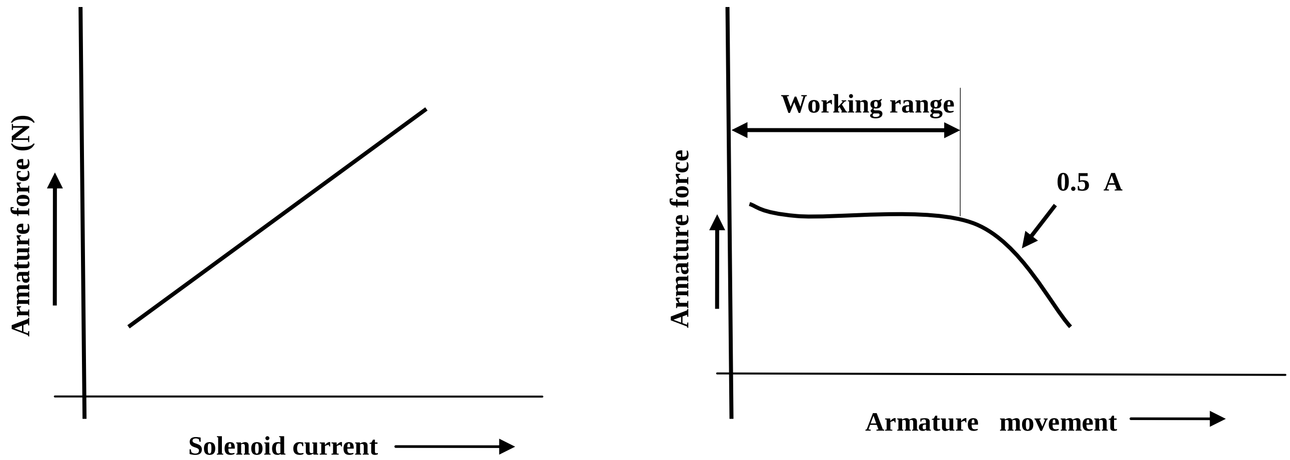


Figure 1.7 Proportional solenoid characteristics.

1.4.1 Force Position Control

The electrical control to the proportional valve normally uses a variable current rather than a variable voltage. If a voltage control system is adopted, any variation in coil resistance caused by temperature change will result in a change in current. This problem is eliminated by using a current control system. It is possible to control a force electrically. By applying the force to a compression spring, its deflection can be controlled. If the spool in a valve (as in Fig. 1.8) is acted on by a spring at one end and a proportional solenoid on the other, the orifice size can be varied along with the control current.

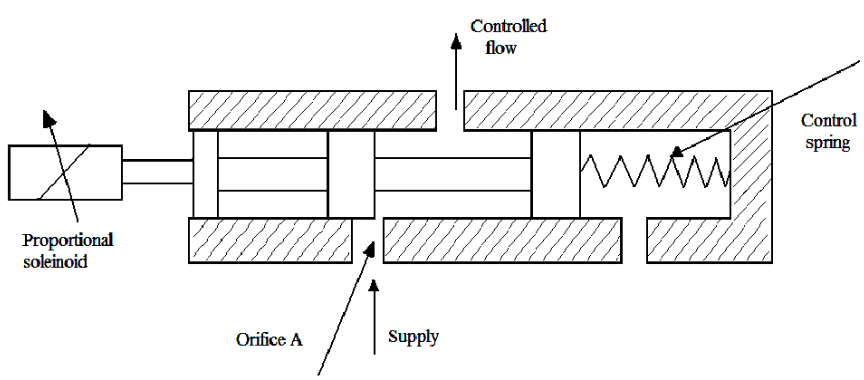


Figure 1.8 Diagrammatic section of a proportional control valve.

The flow from the valve is proportional to the current flowing through the solenoid. Because of the difficulties in manufacturing a zero lap spool, overlapped spools are used in proportional spool valves. This means that the spool has to move a distance equal to the overlap before any flow occurs through the valve, giving rise to a dead zone as shown in Fig. 1.9.

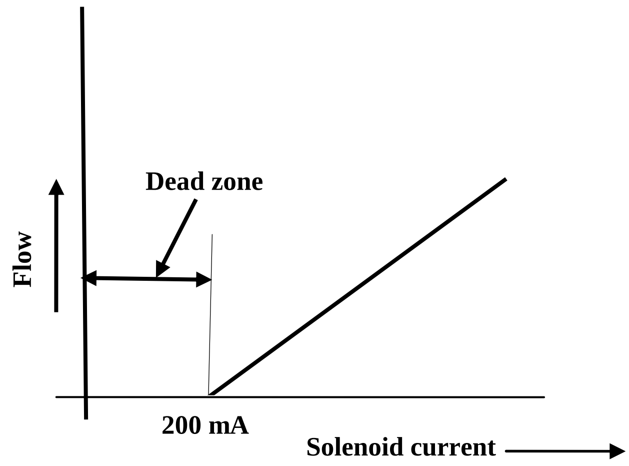


Figure 1.9 Flow current characteristics of a proportional control valve.

Notched spools gives better control of the flow rate because the orifice is progressively opened. Notch shape determines the amount of maximum flow. A diagrammatic representation of the notched spool valve is shown in Fig. 1.10(a)together with an electrical control diagram in Fig. 1.10(b). A proportional directional control valve with a double solenoid and spring centered is similar to the notched spool valve except that it has a solenoid at each end of the spool and spring-centering device. The symbol for such a valve is shown in Fig. 1.11as either a five-position [Fig. 1.11(a)] or a three-position valve [Fig. 1.11(b)]; both symbols are in common use. The extremes on a five position valve represent fully operated conditions.

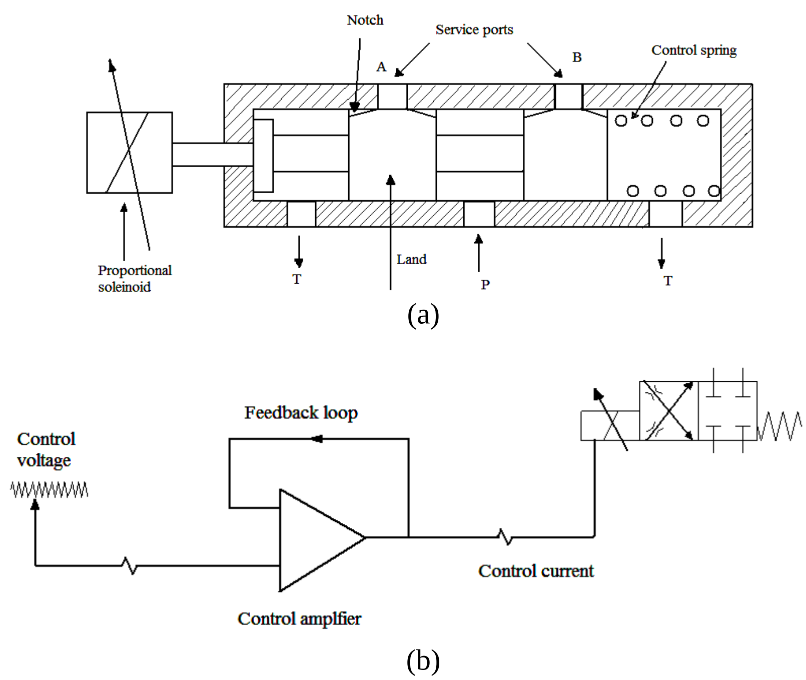
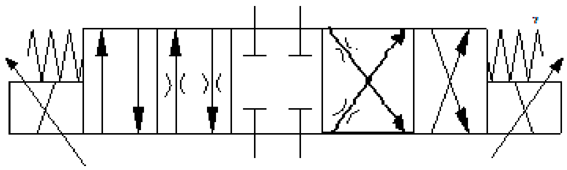
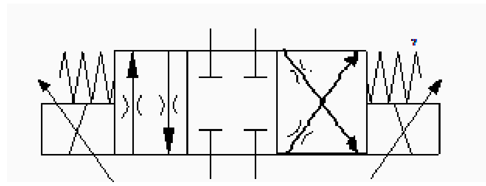


Figure 1.10 Notched spool proportional valve**.**(a) Valve construction;(b) electrical control diagram.



(a)



(b)

Figure 1.11 The symbols of a proportional directional control valve: (a) Five position;(b) three position .

1.4.2 Spool Positional Control

In order to increase the accuracy and extend the range of applications of proportional control valves, a linear transducer may be fitted to measure the spool position. The output from the transducer is a voltage that is proportional to the spool displacement and it continuously varies through the total spool movement. The actual position of the spool is fed back via the transducer to the electrical control system and then compared with the required position, the control current being adjusted accordingly. Such a system is shown in Fig. 1.12.

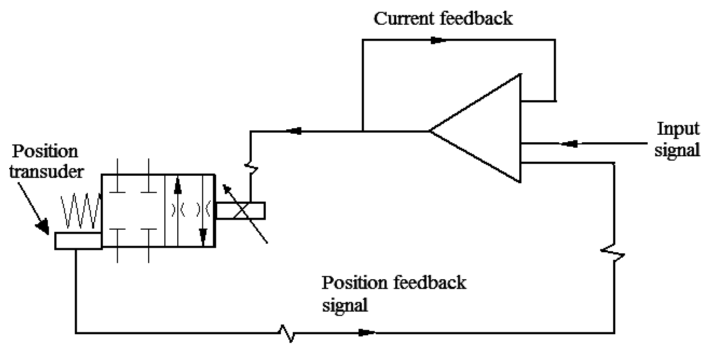


Figure 1.12 The symbols of a proportional directional control valve.

, In such valves spool opening and the flow rate is controlled in both forward and return direction. The transducer used for the position feedback of the spool does not monitor the quantity of fluid flowing through the valve. So it is an open-loop control system. If additional accuracy is required, it is possible to use a transducer to measure the system output and feed this back to the control circuit. In the speed control circuit of a hydraulic motor shown in Fig. 1.13, a tachogenerator or a similar device is used to measure the speed, in which case the effect of the “dead zone” must be considered. This will be more critical in the case of position control rather than speed control.

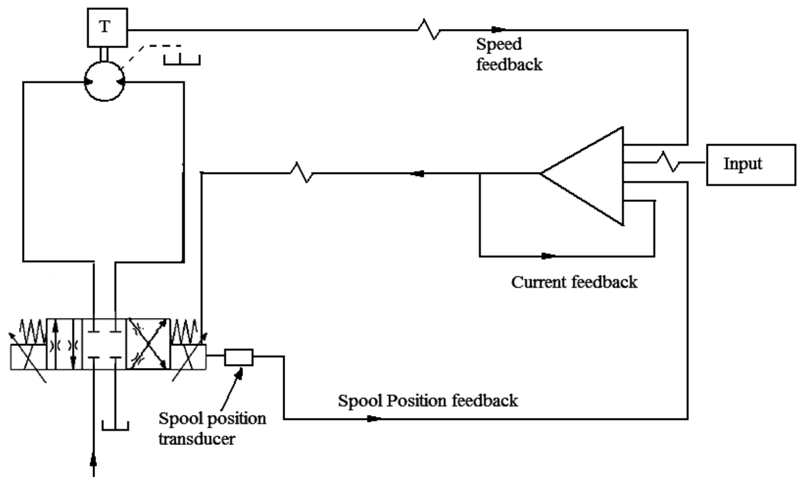


Figure 1.13 Closed-loop speed control with a proportional valve.

1.4.3 Proportional Pressure Control

In a conventional pressure control valve, a spring is used to control the pressure at which the valve operates. The spring is replaced by a DC solenoid in the case of proportional valves; the force set up by the solenoid is controlled by being dependent on the current flowing through it.

1.4.3.1 Single-Stage ProportionalRelief Valves

Direct-acting proportional relief valves are shown in Fig. 1.14. The proportional solenoid exerts a force on the poppet keeping the valve closed, until the hydraulic pressure at port P overcomes this force and opens the valve. In the design of the relief valve, the proportional solenoid acts directly on the valve poppet.

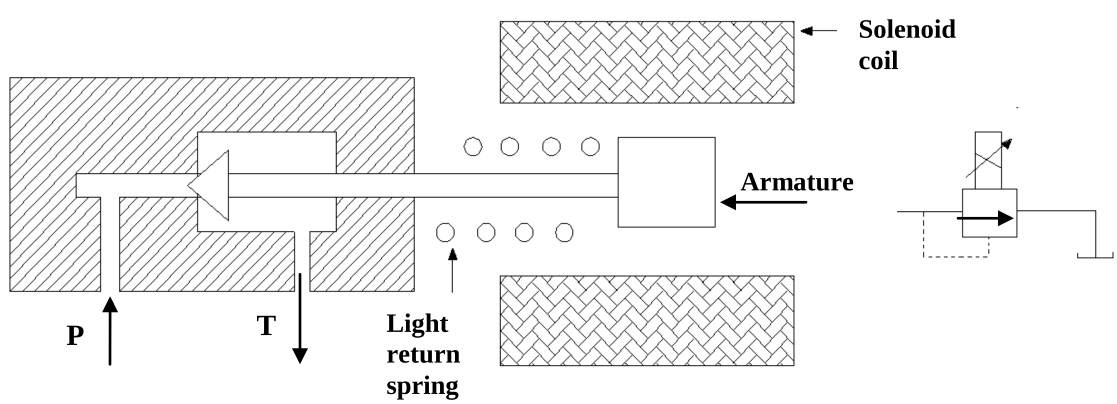


Figure 1.14 Direct-acting proportional relief valve.

1.4.3.2 Proportional Pressure-Reducing Valves

This operates in a manner similar to a conventional pressure-regulating valve, with the control spring being replaced by a proportional solenoid. When this solenoid is not energized, the proportional valve is closed unlike the conventional pressure reducing, which is normally open.

The output pressure of the valve, shown diagrammatically in Fig.1.15 is proportional to the current flowing through the solenoid.

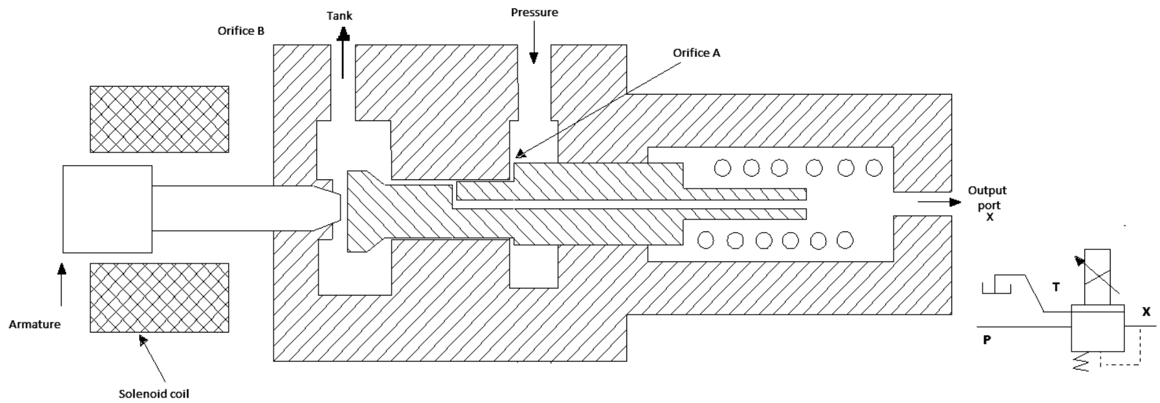


Figure 1.15 Proportional pressure-reducing valve.

When the solenoid is energized, it moves the spool to the right, the control orifice A opens and allows fluid to flow to the output port X. As the aperture of orifice A increases, the aperture of orifice B reduces; the pressure at the control output X is dependent upon the openings of control orifices A and B. This is shown in Fig. 1.16.

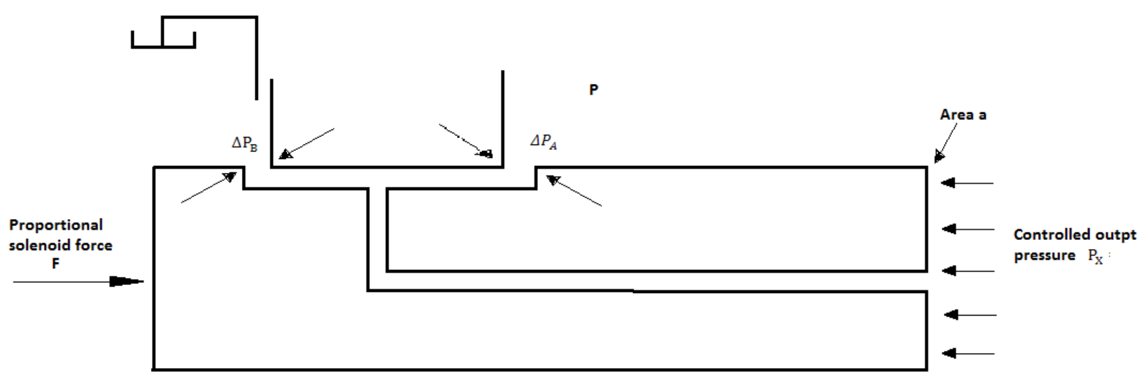


Figure 1.16 Principle of a pressure-reducing valve.

Let the supply pressure be p_1. The pressure drops across the control orifices A and B are p_A and p_B, respectively, and the output pressure is p_x.Then

$$p = \Delta p_A + \Delta p_B \quad \text{and} \quad p_X = \Delta p_B$$

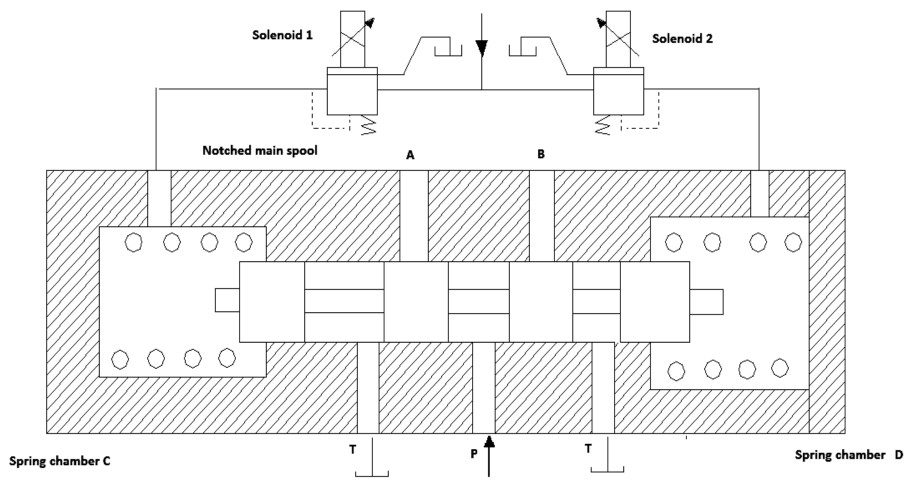
If the control orifice B is fully closed, p_x equals the supply pressure p_1.The output pressure is applied to the right-hand end of the spool and if this is greater than the equivalent pressure exerted by the proportional solenoid, the spool moves to the left. This increases the opening of orifice B and reduces that of orifice A, reducing the output $p_x a = F$. The output pressure is proportional to the current flowing in the proportional solenoid. There is always a flow to the tank from this type of valve if the output pressure p_x is less than the supply pressure p_1.It is essential that there is no back pressure in the tank line if the valve is to function properly.

1.4.4 Two-Stage Proportional Valves

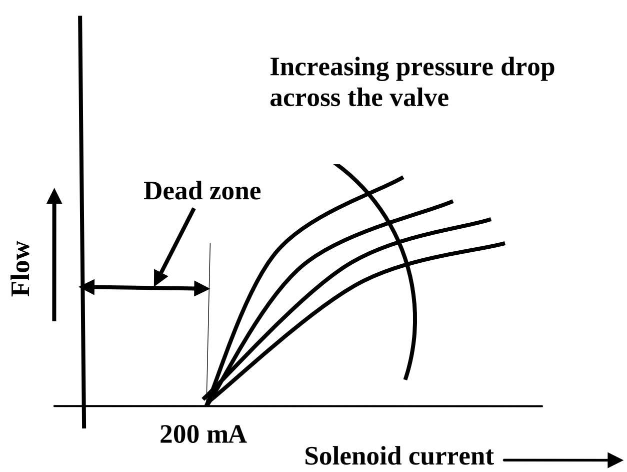
The valves already discussed have a maximum flow capacity of 5 LPM; to obtain higher flow rates in valves, two-stage versions are available. A single-stage proportional pressure control valve is used to pilot the main valve. These operate in a manner similar to conventional two-stage valves.

1.4.5 Two-Stage Proportional Directional Control Valves

The pressure output from a proportional pressure-reducing valve is directed to move the spool of the main valve against a control spring. Energizing solenoid 1 causes pressure to be applied to pilot port X and hence to current in solenoid 1. As the main spool lands are notched, a movement to the right progressively opens the flow paths from P to B and A to T. De-energizing solenoid 1 de-pressurizes spring chamber C and the control spring centralizes the spool.



(a)



(b)

Figure 1.17 Two-stage directional control**.**

Similarly, solenoid 2 controls the flow paths P to A and B to T. The symbol for such a valve is shown in Fig. 1.1. The operating time of valve from mid-position to one extreme is the minimum of 40–60 ms. The operating time can be dependent on the rate of increase or decrease of the control current. The output flow from the valve depends on the pressure drop across the valve and the control current in the solenoid.

1.4.6 Two-Stage Proportional Relief Valve

This is similar to a conventional two-stage relief valve but with a proportional pilot relief valve controlling the main spool. The system pressure is applied via the control orifice to the pilot stage.When the pressure exceeds the force generated by the proportional solenoid, the pilot stage opens. This causes a flow across the control orifice with a resultant pressure drop. The pressures on the main spool are out of balance and so the spool lifts thereby relieving the fluid. A small conventional direct-acting relief valve can be incorporated into the design as an overload pilot to protect the system from any possible malfunction of the proportional valve or electrical control circuit.

1.4.7 Proportional Flow Control

Small flows may be controlled by using one pair of ports of a four-port, two-position proportional directional control valve and higher flows may be controlled if two ports are coupled together. These methods of connections are illustrated in Fig. 1.18. The flow through the valve is proportional to the current flowing in the solenoid and to the pressure drop across the valve. The flow characteristic is not precisely linear because the flow opening is not exactly proportional to the applied current. By carefully designing notches in the spool, it is possible to obtain a variable sharp-edged orifice; this reduces the effect on the flow of variations in the fluid viscosity.

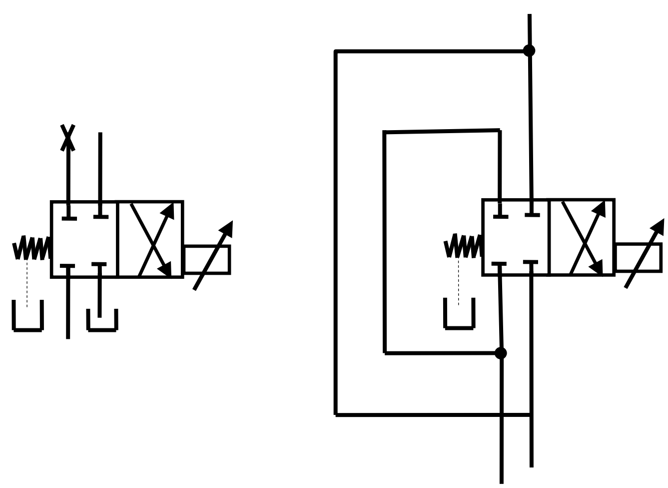


Figure 1.18 Connection of a four-port proportional direction control valve single-path flow and double-path flow capacity.

1.4.7.1 Pressure-Compensated Proportional Flow Control

If a constant pressure drop is maintained across the flow control valve orifice, the flow through the valve is independent of any upstream or downstream pressure variations. This is achieved by using a pressure-compensating cartridge, as employed in conventional flow control valves. The compensator can be considered as a remotely operated pressure control valve, which continuously varies its orifice opening to maintain a fixed pressure drop over the flow control orifice. The pressure difference between the two pilots on the compensator valve, that is the pressure drop across the flow control orifice, is equivalent to the fixed force set by the control spring of the compensator valve.

1.4.7.2 Electrical Control of Proportional Valves

A block schematic for a proportional amplifier together with a current–time graph showing ramping is shown in Fig. 1.19. The ramp-up control determines the rate at which the control signal increases and hence the acceleration of the actuator. The ramp-down control corresponds to the deceleration[Fig.1.19(b)]. The input level control determines the maximum value of the control signal. A low-level dither signal is superimposed onto the control signal. It is an AC level at 100 Hz. Its function is to keep the spool oscillating to overcome the effects of static friction[Fig. 1.19(a)].

The feedback signal is either from the output current or from the spool position in the valve. The feedback does not indicate the actuator conditions and it is still an open-loop system. In order to close the loop, the transducer has to measure of the actuator (either position or speed), feed the signal back and compare it with the output of the actuator, feed the signal back and compare with the desired values. The difference between these values is converted into a new input signal. A block diagram of a closed loop control system is shown in Fig. 1.20. Although closed-loop control can be achieved using proportional valves, it would not be as accurate or have as fast response as an electrohydraulic servo-valve-based system.

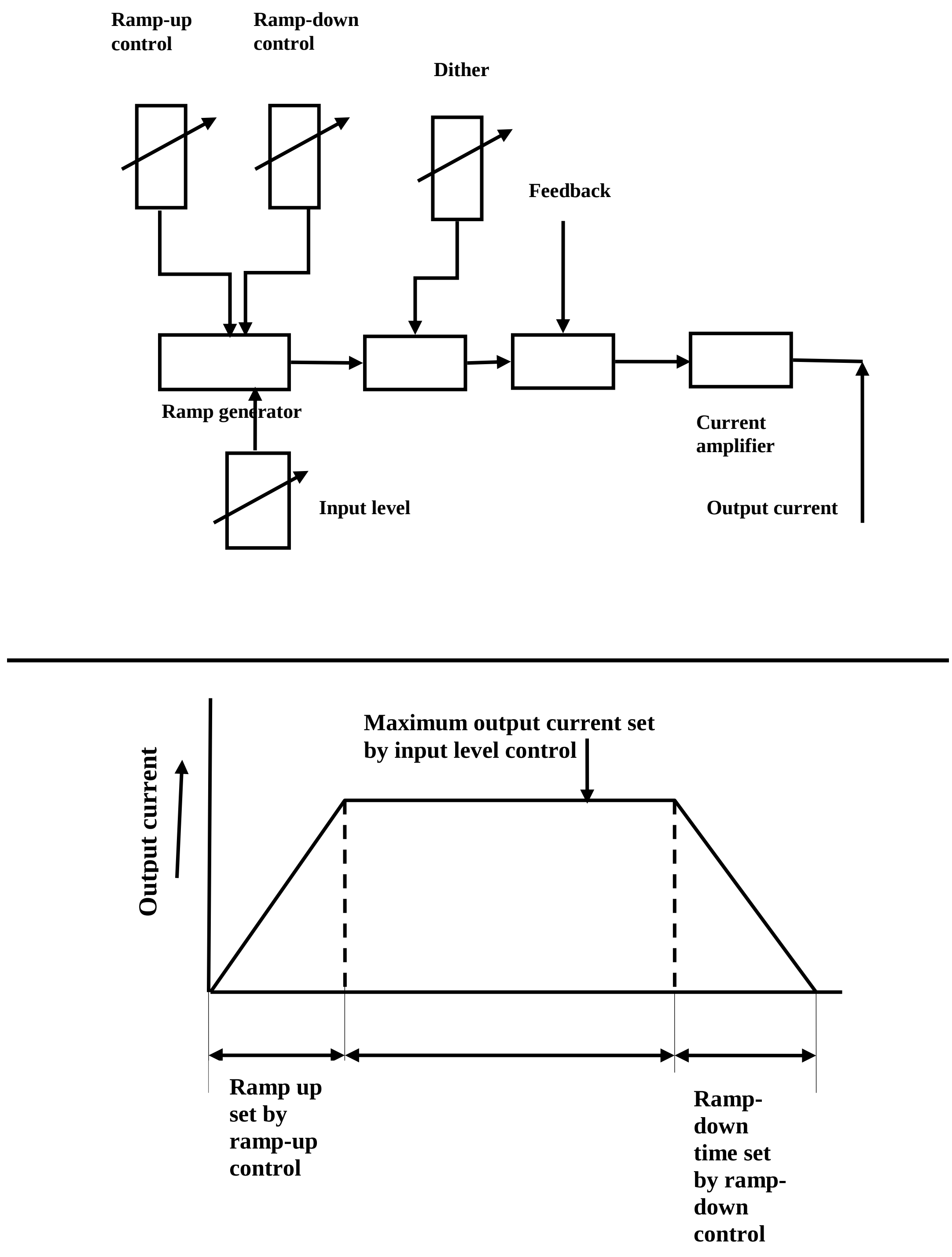


Figure 1.19 Proportional amplifier:(a) Block schematic diagram;(b) current–time characteristics.

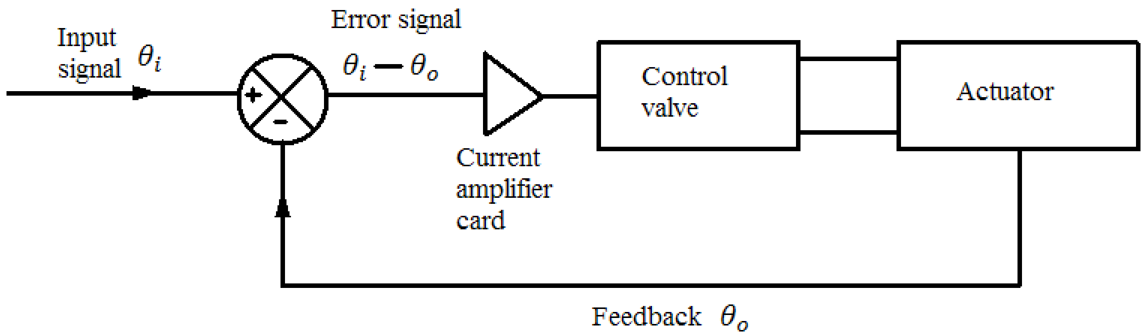


Figure 1.20 Closed-loop control.

1.5.1 Response Speed and Dynamic Characteristics

A short travel speed of minimum mass and consequently low inertia is used in servo valves, giving high response speeds and making servo valves suitable for dynamic applications. On the other hand, a proportional valve spool has a longer travel and is biased to one position by a control spring – the spool and spring combination have a much higher inertia than an equivalent servo valve. The application of a dither signal reduces the effects of spool "stiction" and inertia in both proportional and servo valves. Its value is usually adjustable and is set to give maximum response speed without any flow or pressure fluctuations being set up by the dither current.

1.5.2 Hysteresis Effect

Spool position in a servo valve is controlled by a nozzle and flapper or jet pipe system with a feedback link correction for the spool position. A proportional valve relies on the force exerted by a DC coil acting against a spring to position the spool. There is a considerable difference in the valve output depending on whether the current is increasing or decreasing. Proportional valves have higher hysteresis than servo valves.

1.5.3 Null Position

Because of the underlap spool, a very slight change in the control current varies the output of a servo valve about the zero flow position. In a proportional valve, there is no output until the control current exceeds 200 mA that is required to overcome the spring preload and spool overlap . Wherever fast response and accurate control are required, servo valves are best suited, whereas proportional valves are economic. The differences between proportional and hydraulic servo valves are given in Table 1.1. However, proportional valves are much more dirt-tolerant and provide economical and satisfactory alternative for many applications.

Table 1.1 Comparison of proportional and servo valves

Parameter	**Proportional Hydraulic Valve**	**Electrohydraulic Servo Valve**
Valve lap	Overlap spool, causing a “dead zone” on either side of the null position.	Zero or underlap valve spool. No dead zone.
Response time for the valve spool to move fully over	40–60 ms	5–10 ms
Maximum operating frequency	Approx. 10 Hz	Approx. 100 Hz
Hysteresis	Without armature feedback approx. 5% With armature position feedback approx. 1%	Approx. 0.1%

Lecture 23

PROPORTIONAL CONTROL VALVES [CONTINUED]

1.6 Some Applications of Proportional Control Valves

There are many applications in which the precision of servo valves (discussed in Chapter 18) is not needed, but more accuracy is needed than can be obtained with conventional valves. Proportional valves were developed to fill this gap. Their use has increased significantly since the mid-1970s. The following sub-sections present few applications of proportional control valves.

1.6.1 Control of Actuators

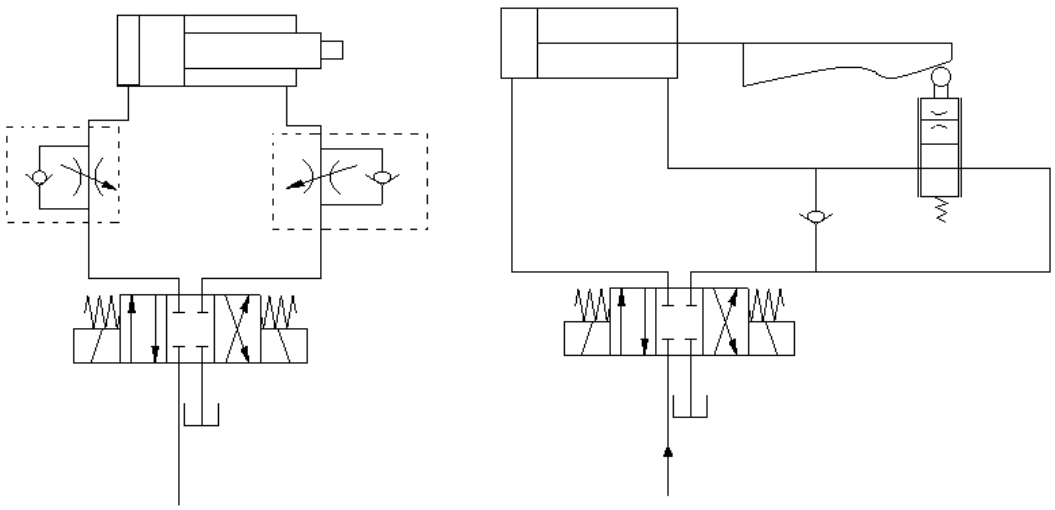
Proportional control valves are commonly used in speed control of actuators such as counterbalancing circuits in broaching machine, three-axis CNC(computer numerically controlled) machines, etc.

1.6.1.1 Speed Control of Cylinders

The conventional speed control of a cylinder is by meter-in, meter-out or spill-off flow control valves. This sets a cylinder speed that can be manually varied. Alternatively, a cam drive can progressively close or open an adjustable orifice in accordance with a preset speed profile that is altered by changing the cam profile. Examples of these types of circuits are shown in Fig. 1.21. The acceleration and retardation of a cylinder can be controlled by the following:

- Relief valves limiting the maximum pressure available to accelerate the load.
- Using a two-stage directional control valve with a choke pack to control the speed of movement of the main spool.
- Using a variable displacement pump.
- Using internal cylinder cushions or external shock absorbers to decelerate the cylinder.
- Building brake, deceleration and counterbalance valves into the circuit to control the deceleration and sometimes the acceleration of the actuator.

All these manual methods are incapable of continuous variations whilst the system operates. A proportional control valve in the cylinder circuit enables continuous regulation of speed, acceleration and retardation. If a proportional control card is used to drive the valve, any adjustments to the maximum current ramp-up and ramp-down have to be carried out by adjusting potentiometers on the card. However, a microprocessor or minicomputer may be employed to control the proportional valve by varying the solenoid current over different parts of the cycle.



(a) (b)

Figure 1.21 (a) Meter-out speed control. (b) Cam-operated speed control.

1.6.2 Speed Control of Hydraulic Motors

This is similar to the speed control of cylinders but it is relatively simple to monitor the motor speed and use a feedback system to control the proportional solenoid as seen in Fig. 1.13. The speed of response to changes in load or command limits the applications. For high response and accurate speeds, servo valves must be used.

1.6.2.1 Position Control of Hydraulic Cylinders

In order to control the position of the piston rod, a transducer has to be used to monitor the actual position. The output of the transducer is compared with the desired piston rod position and the difference is fed to the current amplifier and then to the proportional solenoid. The output of the current amplifier has to be biased so that any error signal that it receives provides a sufficiently large output to drive the proportional valve out of the dead zone. Otherwise the system will become unstable.

1.6.2.2 Pump Control Systems

In an ideal hydraulic system, the output of the power pack in terms of quantity flowing and maximum pressure is matched exactly to the system demand but this situation is rarely ever achieved. A proportional relief valve can be used as the main relief valve to set the maximum pressure and the setting remotely varies with the system sequence. A secondary relief valve should be fitted into the system as a safety feature in case of failure of the proportional relief valve. Supply pressure is matched to the system demand by controlling the proportional relief valve. It is only useful if the flow requirement of the system is constant.

In order to match the pump delivery to a system demand, a variable displacement pump has to be used. One possibility is to use a pressure-compensated pump with a proportional relief valve acting as a compensator (Fig. 1.22). The circuit has to include flow control valves. This method has limited versatility and can be adopted when there is a series of actuators demanding different pressures and the flows are operated sequentially.

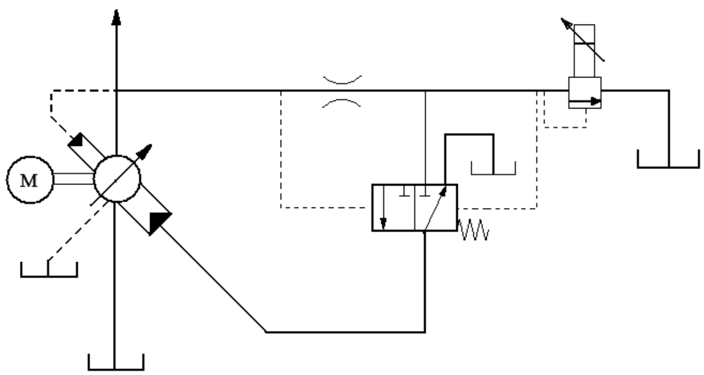


Figure 1.22 Pressure-compensated pump with proportional pressure control.

In any system using flow control valves, there are associated pressure drops that can be eliminated by removing the valves. In this case, the full pump output has to be utilized at all times. Wherever the flow demand varies within different parts of the sequence, this can be achieved by varying the pump displacement by electric motor drives and cam drives. But both have limitations – the motor drive in response speed and cam in versatility. An alternative is a proportional flow control valve and a proportional pressure control (Fig. 1.23). The pump output can be exactly matched to the system demand. This gives a power efficient system with little heat generation. This is shown in Fig. 1.23.

With valve A in the closed position, the directional valve C is piloted to the right which opens the large pump control piston to the tank. The small piston causes the pump to move to zero displacement. When a current is applied to the proportional solenoid on valve A, the orifice partially opens and the fluid flows and provides the system pressure that is below the set pressure of valve B. The pressures occurring across the orifice formed are applied to the pilots of valve C. Valve C spool centralizes locking the displacement of the pump. The system pressure is set by the proportional relief valve B. If the system pressure is greater than the setting pressure of B, the valve opens causing a reduced pressure on the right-hand pilot of valve C that moves to the right, thereby opening the large pump control piston to the tank. Pump displacement reduces until the system pressure matches that set by valve B that balances valve C locking the displacement. Thus, in this system the pump delivery and pressure can be matched to the system by remotely operating the proportional flow and proportional pressure control valves A and B.

The response time is of the order of 50–100 ms. If a pressure surge occurs in the system, the pump may not respond quickly enough to reduce the surge; therefore, the conventional pressure relief valve D is fitted to cater for pressure surges. This valve should be set 20% above the maximum operating pressure of the system.

The precise continuous regulation of flow pressure and displacement with consequential control of speed, thrust, position, etc., achievable from modern servo and proportional systems has made hydraulics indispensable in the field of modern drive and control techniques. The special characteristics of this sophisticated equipment involve electronic circuitry matched to the individual components. Therefore, system design and construction are naturally much more complex than electrohydraulic digital control. Here commissioning a prototype may be time consuming than in the case with conventional hydraulic equipment. Nevertheless, these developments have presented the hydraulic design engineer with many exciting opportunities and extended the application of the subject to new areas.

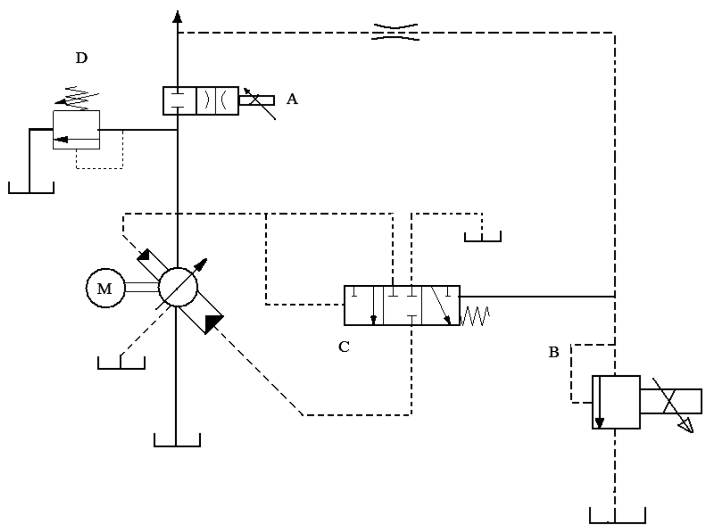


Figure 1.23 Pump with proportional pressure and flow control.

1.7 Analysis of Proportional Valves

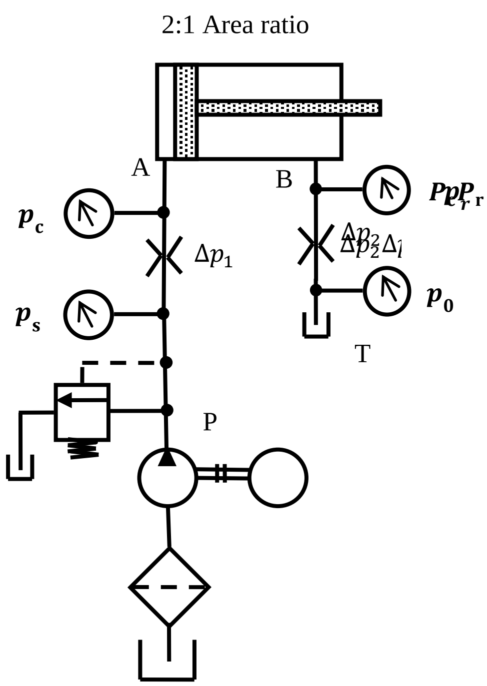


Figure 1.24 Cylinder circuit with a proportional valve.

The diagram of a proportional valve in a simple cylinder circuit is shown in Fig. 1.24. Let Δp_1 be the pressure drop between ports A and P, Δp_2 the pressure drop between ports B and T, as shown in Fig. 1.24. Q_1 the flow corresponding to the pressure drop Δp_1 and Q_2 the flow corresponding to the pressure drop Δp_2 .The cylinder has an area ratio of 2:1 and proportional valve with 2:1area ratio is used.

Using the orifice equations, we can write the expression for flow as

$$Q_1 = C_d A_1 \sqrt{\Delta p_1} \quad (1.1)$$

$$Q_2 = C_d A_2 \sqrt{\Delta p_2} \quad (1.2)$$

where Q_1 is the flow into the cap end of the cylinder, Q_2 is the flow out of the rod end of the cylinder, A_1 is the area of the orifice between ports P and A and A_2 is the area of the orifice between ports B and T.

The orifices have the same shape on both sides of the spool land. The number of grooves on one side is twice the other side for a 2:1 area ratio proportional valve. The same orifice coefficient for both sides shall be used.

The area ratio of the cylinder is 2:1; therefore, during the extension,

$$Q_2 = \frac{Q_1}{2}$$

Also

$$A_2 = \frac{A_1}{2}$$

Using the flow equations, we get

$$Q_1 = C_d A_1 \sqrt{\Delta p_1} \quad (1.3)$$

$$Q_2 = C_d A_2 \sqrt{\Delta p_2} = \frac{Q_1}{2} = C_d \left\{ \frac{A_1}{2} \right\} \sqrt{\Delta p_2} \quad (1.4)$$

or we can write Q_1 in terms of Δp_2 as

$$Q_1 = C_d A_1 \sqrt{\Delta p_2}$$

If $A_1 = A_2$, then we can show that

$$\Delta p_2 = \frac{\Delta p_1}{4}$$

(It is left as an exercise to the students.)

1.7.1 Overrunning Load

Suppose that the circuit shown in Fig. 1.24 has overrunning load during forward stroke. We can write the force balance on the cylinder as

$$p_c A_c = F_f + F_L + p_r A_r \qquad (1.5)$$

where $F_L = W$ = load on the cylinder (N),F_f is the frictional force (N), p_r is pressure at port B(Fig. 1.24). In this case, F_L is negative, since the load is overrunning, that is, it is acting in the direction of the movement of the cylinder. Solving for p_r we get

$$p_r = \frac{p_c A_c - F_f + F_L}{A_r} \qquad (1.6)$$

The pressure drop across port P to a orifice in the proportional valve is

$$\Delta p_1 = p_s - p_c \qquad (1.7)$$

Neglecting any drop between the proportional valve outlet and the tank, we can write $p_o = 0$. Then pressure drop from ports A to T is

$$\Delta p_2 = p_r - p_o = p_r \qquad (1.8)$$

If the area ratio is unity (i.e., $A_1 = A_2 = A$), the orifice equation becomes

$$Q_1 = C_d\, A\sqrt{\Delta p_1} \qquad (1.9)$$

$$Q_2 = C_d\, A\sqrt{\Delta p_2} \qquad (1.10)$$

Squaring both sides and eliminating $C_d\, A$ we get

$$\frac{Q_1^2}{Q_2^2} = \frac{\Delta p_1}{\Delta p_2} \qquad (1.11)$$

$$\Rightarrow \Delta p_2 = \Delta p_1 \times \frac{Q_2^2}{Q_1^2} \qquad (1.12)$$

Using Eqs. (1.7), (1.8) and (1.12) we can write

$$p_r = (p_s - p_c) \times \frac{Q_2^2}{Q_1^2} \qquad (1.13)$$

Equating Eqs.(1.6) and (1.13), we can now solve for the pressure at the cap end of the cylinder as

$$p_r = \frac{p_c A_c - F_f + F_L}{A_r} = (p_s - p_c) \times \frac{Q_2^2}{Q_1^2}$$

Rearranging we get relationship between p_c and other system parameters as

$$p_c = \frac{p_s\left[\frac{Q_2^2}{Q_1^2}\right] - \frac{[F_f - F_L]}{A_r}}{\left[\frac{A_c}{A_r}\right] + \left[\frac{Q_2^2}{Q_1^2}\right]}$$

Under certain conditions, p_c can be negative, which means that a vacuum exists in the cap end of the cylinder and the cylinder is not completely filled with oil. When this condition develops, the positive control of the load is lost during extension.

Example 1.1

Consider the hydraulic system shown in Fig. 1.24 The cylinder ratio is 2:1, pressure $p_s = 100\,\text{bar}, A_c = 0.002032\,\text{m}^2,\ A_r = 0.001070\,\text{m}^2,\ F_f = 290\,\text{N}$.

(a) Find the load that will cause negative pressure at the cap end of the cylinder and corresponding pressure at the rod end and pressure drop across port P to A and port B to T.

(b) If the load is 4450 N, find the pressure drop across port P to port A and corresponding pressure at the rod end. Is it possible to obtain this pressure drop valve area ratio 1:1?

(c) If the valve area ratio is 2:1, what is the overrunning load? Comment on the result.

Solution

(a) Because the cylinder area is 2:1, we have

$$Q_2 = \frac{Q_1}{2} \text{ or } \left[\frac{Q_1^2}{Q_2^2}\right] = 0.25$$

To find the load that causes the negative pressure on the cap end, set $p_c = 0$. We know that

$$p_c = \frac{p_s\left[\frac{Q_2^2}{Q_1^2}\right] - \frac{[F_f - F_L]}{A_r}}{\left[\frac{A_c}{A_r}\right] + \left[\frac{Q_2^2}{Q_1^2}\right]}$$

$$\Rightarrow 0 = p_s\left[\frac{Q_1^2}{Q_2^2}\right] - \frac{F_f - F_L}{A_r}$$

$$\Rightarrow F_L = p_s\left[\frac{Q_1^2}{Q_2^2}\right] A_r + [F_f]$$

Substituting the values we get $F_L = 100 \times 10^5 [0.25] \times 0.001070 + [290] = 2965\,\text{N}$

Any load greater than 2965 N will cause a negative pressure at the cap end of the cylinder. If the overrunning load is 2965 N, the pressure at the rod end is given by

$$p_r = \frac{p_c A_c - F_f + F_L}{A_r}$$
$$= \frac{0 - 290 + 2965}{0.001070} = 25 \text{ bar}$$

The pressure drop across the port P to port A orifice is

$$\Delta p_1 = p_s - p_c = 100 - 0 = 100\,\text{bar}$$
$$\Delta p_2 = p_r - p_o = p_r = 25\,\text{bar}$$

For any overrunning load greater than 2965 N, the valve will not create enough pressure drop across port B to port A orifice to maintain the control of load.

(b) When the load $F_L = 4450\,\text{N}$, we have

$$p_c = \frac{p_s\left[\dfrac{Q_2^2}{Q_1^2}\right] - \dfrac{[F_f - F_L]}{A_r}}{\left[\dfrac{A_c}{A_r}\right] + \left[\dfrac{Q_2^2}{Q_1^2}\right]}$$
$$= \frac{100 \times 10^5 [0.25] - \dfrac{[290 - 4450]}{0.001070}}{\left[\dfrac{0.002032}{0.001070}\right] + [0.25]}$$

$$= -\frac{13.8785 \times 10^5}{2.14907} = -6.4579 \times 10^5 = -6.4579\,\text{bar}$$

To create $p_c = -6.67$ bar, the pressure drop across port P to port A orifice must be

$$\Delta p_1 = p_s - p_c$$
$$= 100 - (-6.4579)$$
$$= 106.4579\,\text{bar}$$

which is not possible. Using Eq.(1.6) the required pressure at the rod end is

$$p_r = \frac{pA_c - F_f + F_L}{A_r}$$
$$= \frac{-6.4579 \times 10^5 \times 0.002032 - 290 + 4450}{0.001070} = 26.61 \text{ bar}$$

(c) Let us use the valve with the area ratio of 2:1. We have $A_1 = 2\,A_2$

$$Q_1 = C_d A_1 \sqrt{\Delta p_1} \tag{1.14}$$

$$Q_2 = C_d A_2 \sqrt{\Delta p_2} = C_d \frac{A_1}{2} \sqrt{\Delta p_2} \tag{1.15}$$

SolvingEqs. (1.14) and (1.15) we get

$$\frac{Q_1^2}{4Q_2^2} = \frac{\Delta p_1}{\Delta p_2}$$

$$\Rightarrow \Delta p_1 = \frac{Q_1^2}{4Q_2^2} \Delta p_2 \tag{1.16}$$

Using Eqs.(1.7) and (1.8) and substituting in Eq. (1.16). Solving for the pressure at the end of the cylinder, we can get

$$p_r = (p_s - p_c) \times \frac{4Q_2^2}{Q_1^2} \tag{1.17}$$

Equating Eqs.(1.6) and (1.17) , we can now solve for pressure at the cap end of the cylinder as

$$p_c = \frac{p_s \left[\frac{4Q_2^2}{Q_1^2}\right] - \frac{[F_f - F_L]}{A_r}}{\left[\frac{A_c}{A_r}\right] + \left[\frac{4Q_2^2}{Q_1^2}\right]}$$

Under certain conditions p_c can be negative. This means a vacuum will exist in the cap end of the cylinder; the cylinder will not be completely filled with oil. When this condition develops, positive control of load is lost during the extension. It is instructive to determine what load will cause p_c to go negative,

Setting $p_c = 0$, we can find the maximum overrunning load that can be controlled as

$$F_L = p_s A_r \left[\frac{4Q_2^2}{Q_1^2}\right] + [F_f]$$

For 2:1 area ratio,

$$Q_2 = \frac{Q_1}{2}$$

Therefore

$$\left[\frac{4Q_2^2}{Q_1^2}\right] = 1$$

So

$$F_L = p_s A_r\left[\frac{4Q_2^2}{Q_1^2}\right] + [F_f]$$
$$= 100\times10^5 \times 0.001070 + 290$$
$$= 10990\,\mathrm{N}$$

Therefore, the 2:1 area ratio valve can control the overrunning load more than three times the size load controlled with a 1:1 area ratio valve.Now the cap end pressure can be calculated using the equation

$$p_c = \frac{p_s\left[\frac{4Q_2^2}{Q_1^2}\right] - \frac{[F_f - F_L]}{A_r}}{\left[\frac{A_c}{A_r}\right] + \left[\frac{4Q_2^2}{Q_1^2}\right]}$$
$$p_c = \frac{100\times10^5[1] - [290 - 4450]/0.001070}{\left[\frac{0.002032}{0.001070}\right] + [1]}$$
$$= \frac{61.12\times10^5}{2.8990}$$
$$= 21.0826\,\mathrm{bar}$$

Also

$$p_r = \frac{p_c A_c - F_f + F_L}{A_r}$$
$$= \frac{21.0826\times10^5[0.002032] - 290 + 4450}{0.001070}$$
$$= 78.91\,\mathrm{bar}$$

The pressure drop across the valve is

$$\Delta p_1 = p_s - p_c$$
$$= 100 - (21.0826)$$
$$= 78.91\,\mathrm{bar}$$

$$\Delta p_2 = p_r - 0 = 78.91\,\mathrm{bar}$$

The total pressure drop across the valve is

$$\Delta p_1 + \Delta p_2 = 78.91 + 78.91 = 157.82\,\mathrm{bar}$$

Example 1.2

Consider the hydraulic circuit with resistive load shown in Fig. 1.25. The cylinder ratio is 2:1, pressure $P_s = 100\,\text{bar}, A_c = 0.002032\,\text{m}^2$, $A_r = 0.001070\,\text{m}^2$, $F_f = 290\,\text{N}$, $F_L = 4450\,\text{N}$. If the valve has 2:1 area ratio. Determine p_r, p_c and total pressure drop.

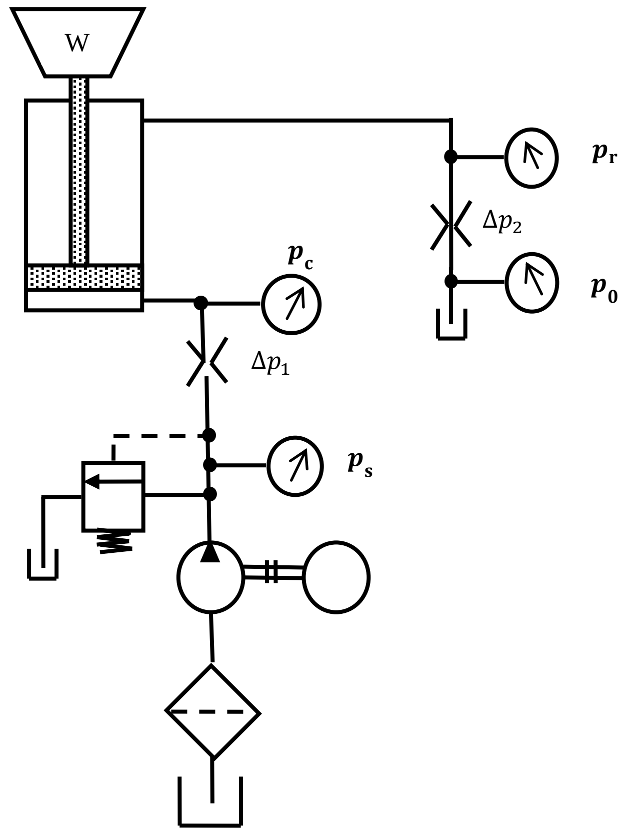


Figure 1.25 Control of resistive load.

Solution: We can write the force balance on the cylinder as

$$p_c A_c = F_f + F_L + p_r A_r$$

where $F_L = W$ is the load on the cylinder (N) and F_f is the frictional force (N). Solving for p_c we get

$$p_c = \frac{p_r A_r + F_f + F_L}{A_c} \tag{1.18}$$

If the area ratio is unity (i.e., $A_1 = A_2 = A$), the orifice equation becomes

$$Q_1 = C_d\, A\sqrt{\Delta p_1}$$
$$Q_2 = C_d\, A\sqrt{\Delta p_2}$$

Solving we get

$$\frac{Q_1^2}{Q_2^2} = \frac{\Delta p_1}{\Delta p_2}$$

$$\Rightarrow \Delta p_1 = \frac{Q_1^2}{Q_2^2}\Delta p_2$$

Also also using Eqs.(1.7) and (1.8) in Eq. (1.12) we get

$$p_{\mathrm{c}} = p_{\mathrm{s}} - p_{\mathrm{r}} \times \frac{Q_1^2}{Q_2^2} \qquad (1.19)$$

Equating Eqs.(1.18) and (1.19) and solving for p_{r} we get

$$p_{\mathrm{r}} = \frac{p_{\mathrm{s}} - [F_{\mathrm{f}} + F_{\mathrm{L}}]/A_{\mathrm{c}}}{\left[\frac{A_{\mathrm{r}}}{A_1}\right] + \left[\frac{Q_1^2}{Q_2^2}\right]}$$

If $F_{\mathrm{L}} = 4450\,\mathrm{N}$ and other parameters as the same as Example 1.1, then

$$p_{\mathrm{r}} = \frac{p_{\mathrm{s}} - [F_{\mathrm{f}} + F_{\mathrm{L}}]/A_{\mathrm{c}}}{\left[\frac{A_{\mathrm{r}}}{A_1}\right] + \left[\frac{Q_1^2}{Q_2^2}\right]} \qquad (1.20)$$

$$= \frac{100 \times 10^5 - [290 + 4450]/0.002032}{\left[\frac{0.001070}{0.002032}\right] + 4}$$

$$= 16.94\,\mathrm{bar}$$

Now substituting back into Eq.(1.18) we get

$$p_{\mathrm{c}} = \frac{p_{\mathrm{r}} A_{\mathrm{r}} + F_{\mathrm{f}} + F_{\mathrm{L}}}{A_{\mathrm{c}}}$$

$$= \frac{16.94 \times 10^5 \times 0.001070 + 290 + 4450}{0.002032} = 32.25\,\mathrm{bar}$$

The pressure drop across the valve are

$$\Delta p_1 = p_s - p_c = 100 - 32.25 = 67.75\,\mathrm{bar}$$

$$\Delta p_2 = p_{\mathrm{r}} = 16.94\,\mathrm{bar}$$

So

$$\begin{aligned}\Delta p_{\mathrm{total}} &= \Delta p_1 + \Delta p_2 \\ &= 67.75 + 16.94 \\ &= 84.69\,\mathrm{bar}\end{aligned}$$

If the valve has a 2:1 area ratio, we have

$$\Delta p_1 = \frac{Q_1^2}{4Q_2^2}\Delta p_2$$

Equation (1.20) becomes

$$p_r = \frac{p_s - \left[F_f + F_L\right]/A_c}{\left[\dfrac{A_r}{A_1}\right] + \left[\dfrac{Q_1^2}{4Q_2^2}\right]}$$

$$= \frac{100\times10^5 - [290+4450]/0.002032}{\left[\dfrac{0.001074}{0.002032}\right]+[1]}$$

$$= 50.16\,\text{bar}$$

Substituting back into Eq.(1.18), we get

$$p_c = \frac{p_r A_r + F_f + F_L}{A_c}$$

$$= \frac{50.16\times10^5\times0.001074+290+4450}{0.002032} = 49.84\text{ bar}$$

The pressure drop across the valve are

$$\Delta p_1 = p_s - p_c = 100 - 49.84 = 50.16\,\text{bar}$$

$$\Delta p_2 = p_r = 50.16\,\text{bar}$$

So

$$\begin{aligned}\Delta p_{\text{total}} &= \Delta p_1 + \Delta p_2\\ &= 50.16 + 50.16\\ &= 100.32\text{ bar}\end{aligned}$$

To select a valve for this application, the designer must look in manufacturer's literature and choose a valve with 2:1 area ratio spool having an operating curve for ~100 bar pressure drop. The best control is achieved if full spool stroke, or almost full spool stroke, is used to obtain the desired flow at the desired pressure drop. For large pressure drop like 100 bar, here, we may have to use less than the full spool stroke. Sample data for 2:1 spool area ration rated for 27 GPM with 10bar pressure drop is given in Fig. 1.26.

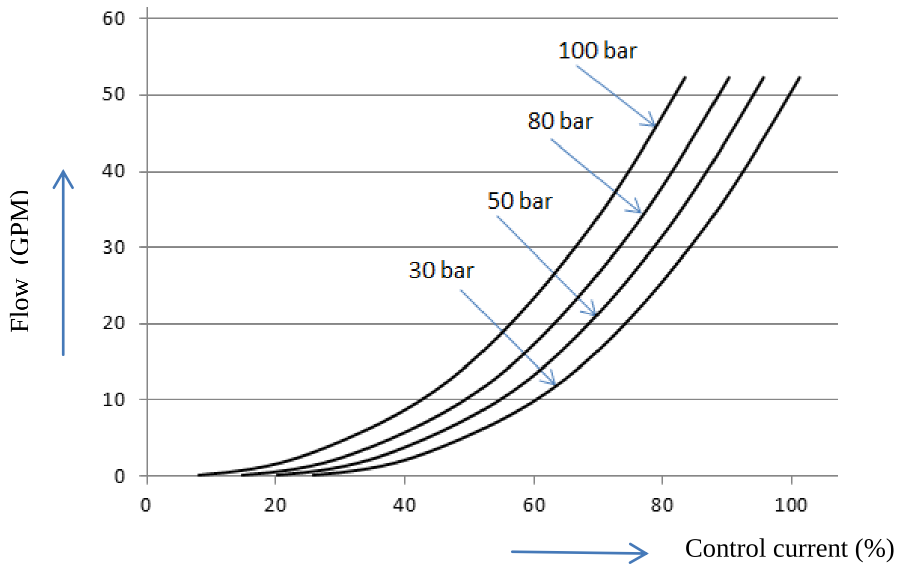


Figure 1.26

From Fig. 1.26 it is clear that, a 65% current will give 30 GPM flow at 100 bar pressure drop.

Objective-type questions

Fill in the Blanks

1. A proportional valve is a valve that produces an output (direction, pressure, flow) that is _____ to an electronic control input.

2. The performance of a proportional valve is a compromise between a conventional solenoid valve and _____ valve.

3. Proportional valves are operated by proportional _____, whereas servo valves are operated by torque motors.

4. A proportional valve has a maximum frequency response of _____.

5. In a conventional pressure control valve, a spring is used to control the pressure at which the valve operates. The spring is replaced by a _____ in the case of proportional valves.

State True or False

1. The response time for a proportional valve spool to move fully over is around 50 ms.

2. Notched spools give a better control of the flow rate.

3. All standard solenoids have no intermediate positions; rather they are always at one end or the otherof the solenoid stroke.

4. A proportional solenoid maintains a diminishing air gap dimension at the end of the plunger.

5. An electrical control to a proportional valve normally uses a variable voltage rather than a variable current.

Review Questions

1. Compare electrohydraulic servo valves with proportional hydraulic valves.
2. Where are proportional valves preferred?
3. Explain the principle of a proportional pressure-reducing valve.
4. With the help of a neat sketch, explain how the speed of a cylinder can be controlled using a proportional valve.
5. Discuss the various controls of proportional valves.
6. What is the difference between a standard solenoid and a proportional solenoid?
7. Explain the concept of operation of a proportional solenoid.
8. What is the purpose of dither in a proportional circuit?
9. Explain the difference between force control and position control in proportional control valves.
10. What is a proportional valve?
11. What is a proportional solenoid?
12. Draw the symbols of proportional 3/2-way DCV.
13. Define resolution, accuracy and repeatability as applied to proportional valves.
14. Name three applications of proportional valves.
15. What is the difference between force-controlled and stroke-controlled proportional valves?

Answers

Fill in the Blanks

1. Proportional
2. Servo
3. Solenoids
4. 100 Hz
5. DC solenoid

State True or False

1. True
2. True
3. True
4. False
5. True

Lecture 24

HYDRAULIC CIRCUIT DESIGN AND ANALYSIS

Learning Objectives

Upon completion of this chapter, the student should be able to:

- Identify the graphic symbols for various types of hydraulic components.
- Explain various hydraulic circuits to control single-acting and double-acting cylinders.
- Explain a regenerative circuit and determine the load-carrying capacities.
- Describe the working of a double-pump circuit along with its advantages.
- Explain the working of a sequencing circuit and a counterbalancing circuit.
- Differentiate between series and parallel synchronization circuits.
- Calculate the speed, pressure and load-carrying capacity of hydraulic circuits.
- Evaluate the performance of hydraulic circuits using various hydraulic elements.

1.1 Introduction

A hydraulic circuit is a group of components such as pumps, actuators, control valves, conductors and fittings arranged to perform useful work. There are three important considerations in designing a hydraulic circuit:

1. Safety of machine and personnel in the event of power failures.
2. Performance of given operation with minimum losses.
3. Cost of the component used in the circuit.

1.2Control of a Single-Acting Hydraulic Cylinder

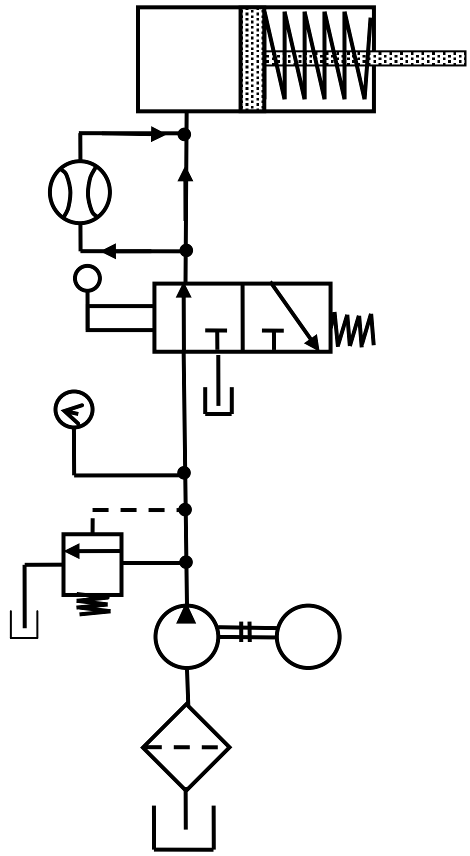


Figure 1.1 Control of a single-acting cylinder.

Figure 1.1 shows that the control of a single-acting,spring return cylinder using a three-way two-position manually actuated, spring offset direction-control valve (DCV). In the spring offset mode, full pump flow goes to the tank through the pressure-relief valve (PRV). The spring in the rod end of the cylinder retracts the piston as the oil from the blank end drains back into the tank. When the valve is manually actuated into its next position, pump flow extends the cylinder.
After full extension, pump flow goes through the relief valve. Deactivation of the DCV allows the cylinder to retract as the DCV shifts into its spring offset mode.

1.3Control of a Double-Acting Hydraulic Cylinder

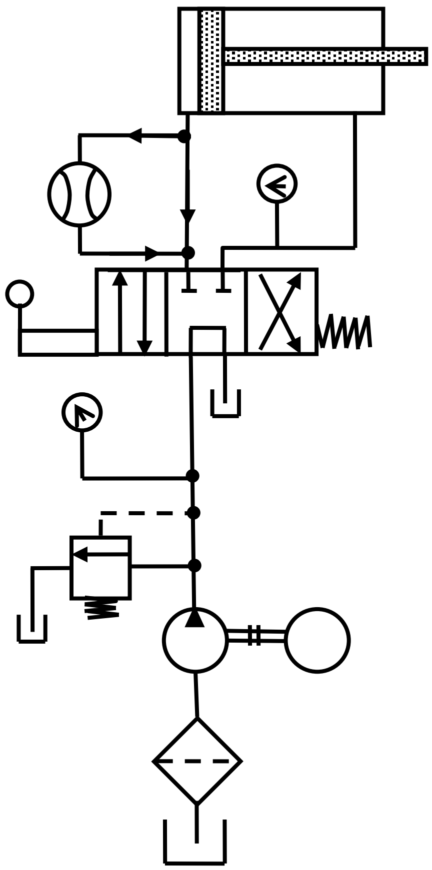


Figure 1.2 Control of a double-acting cylinder.

The circuit diagram to control double-acting cylinder is shown in Fig. 1.2. The control of a double-acting hydraulic cylinder is described as follows:

1. When the 4/3 valve is in its neutral position (tandem design), the cylinder is hydraulically locked and the pump is unloaded back to the tank.
2. When the 4/3 valve is actuated into the flow path, the cylinder is extended against its load as oil flows from port P through port A. Oil in the rod end of the cylinder is free to flow back to the tank through the four-way valve from portB through portT.
3. When the 4/3 valve is actuated into the right-envelope configuration, the cylinder retracts as oil flows from port P through port B. Oil in the blank end is returned to the tank via the flow path from port A to port T.

At the ends of the stroke, there is no system demand for oil. Thus, the pump flow goes through the relief valve at its pressure level setting unless the four-way valve is deactivated.

1.4Regenerative Cylinder Circuit

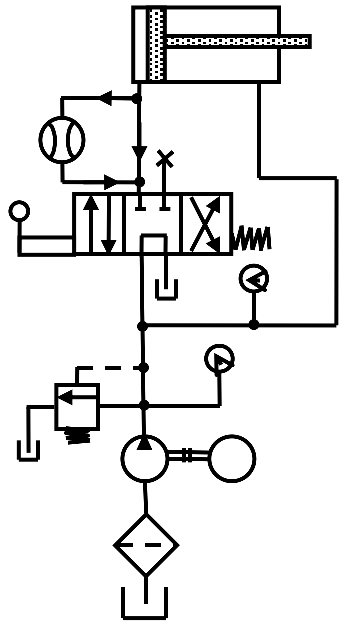


Figure 1.3 Regenerative circuit.

Figure 1.3 shows a regenerative circuit that is used to speed up the extending speed of a double-acting cylinder. The pipelines to both ends of the hydraulic cylinder are connected in parallel and one of the ports of the 4/3 valve is blockedby simply screwing a thread plug into the port opening. During retraction stroke, the 4/3 valve is configured to the right envelope. During this stroke, the pump flow bypasses the DCV and enters the rod end of the cylinder. Oil from the blank end then drains back to the tank through the DCV.

When the DCV is shifted in to its left-envelope configuration, the cylinder extends as shown in Fig. 1.3.The speed of extension is greater than that for a regular double-acting cylinder because the flow from the rod end regenerates with the pump flow Q_p to provide a total flow rate Q_T .

1.4.1 Expression for the Cylinder Extending Speed

The total flow rate Q_T entering the blank end of the cylinder is given by

$$Q_\mathrm{T} = Q_\mathrm{p} + Q_\mathrm{r}$$

where Q_p is the pump flow rate and Q_r is the regenerative flow or flow from the rod end.Hence,

$$\text{Pump flow rate } (Q_\mathrm{p}) = Q_\mathrm{T} - Q_\mathrm{r}$$

But the total flow rate acting on the blank rod end is given by

$$(Q_\mathrm{T}) = A_\mathrm{p} v_\mathrm{ext}$$

Similarly, theflow rate from the rod end is given by

$$(Q_\mathrm{r}) = (A_\mathrm{p} - A_\mathrm{r}) v_\mathrm{ext}$$

So pump flow rate is

$$Q_\mathrm{p} = A_\mathrm{p} v_\mathrm{ext} - (A_\mathrm{p} - A_\mathrm{r}) v_\mathrm{ext}$$

$$\Rightarrow Q_p = A_r v_{ext}$$

The extending speed of the piston is given as

$$v_{ext} = \frac{Q_p}{A_r}$$

Thus, a small area provides a large extending speed. The extending speed can be greater than the retracting speed if the rod area is made smaller. The retraction speed is given by

$$v_{ret} = \frac{Q_p}{A_p - A_r}$$

The ratio of extending and retracting speed is given as

$$\frac{v_{ext}}{v_{ret}} = \frac{Q_p / A_r}{Q_p / (A_p - A_r)} = \frac{A_p - A_r}{A_r} = \frac{A_p}{A_r} - 1$$

When the piston area equals two times the rod area, the extension and retraction speeds are equal. In general, the greater the ratio of the piston area to rod area, the greater is the ratio of the extending speed to retraction speed.

1.4.2 Load-Carrying Capacity During Extension

The load-carrying capacity of a regenerative cylinder during extension is less than that obtained from a regular double-acting cylinder.The load-carrying capacity $F_{load\text{-}extension}$ for a regenerative cylinder during extension equals pressure times the piston rod area. This is because system pressure acts on both sides of the piston during extension. Then

$$F_{load\text{-}extension} = pA_r$$

Thus, wedo not obtain more power from the regenerative cylinder during extension because the extension speed is increased at the expense of reduced load-carrying capacity.

1.5Pump-Unloading Circuit

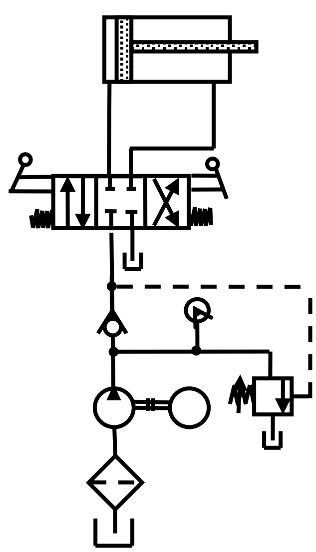


Figure 1.4 Pump-unloading circuit.

Figure 1.4 shows a hydraulic circuit to unload a pump using an unloading valve.When the cylinder reaches the end of its extension stroke, the pressure of oil rises because the check valve keeps the high-pressure oil. Due to high-pressure oil in the pilot line of the unloading valve, it opens and unloads the pump pressure to the tank.

When the DCV is shifted to retract the cylinder, the motion of the piston reduces the pressure in the pilot line of the unloading valve. This resets the unloading valve until the cylinder is fully retracted. When this happens, the unloading valve unloads the pump due to high-pressure oil. Thus, the unloading valve unloads the pump at the ends of the extending and retraction strokes as well as in the spring-centered position of the DCV.

1.6 Double-Pump Hydraulic System

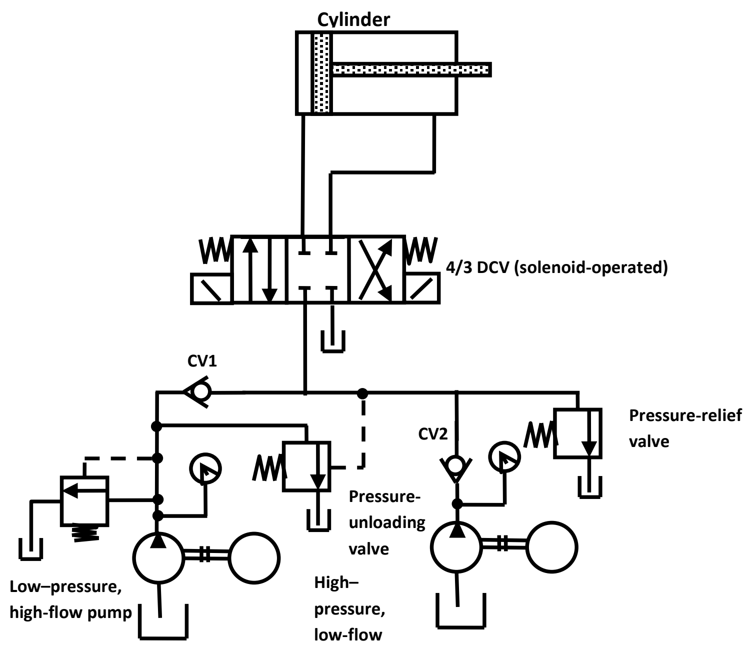


Figure 1.5 Double-pump circuit.

Figure 1.5 shows an application for an unloading valve. It is a circuit that uses a high-pressure, low-flow pump in conjunction with a low-pressure, high-flow pump. A typical application is a sheet metal punch press in which the hydraulic cylinder must extend rapidly over a great distance with low-pressure but high-flow requirements. This occurs under no load. However during the punching operation for short motion, the pressure requirements are high, but the cylinder travel is small and thus the flowrequirementsare low. The circuit in Fig. 1.5 eliminates the necessity of having a very expensive high-pressure, high-flow pump.
When the punching operation begins, the increased pressure opens the unloading valve to unload the low-pressure pump. The purpose of relief valve is to protect the high-pressure pump from over pressure at the end of cylinder stroke and when the DCV is in its spring-centered mode. The check valve protects the low-pressure pump from high pressure, which occurs during punching operation, at the ends of the cylinder stroke and when the DCV is in its spring-centered mode.

1.7Counterbalance Valve Application

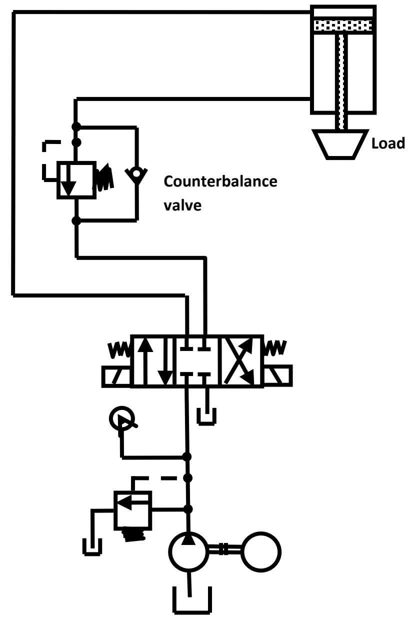


Figure 1.6 Counterbalance valvein circuit.

A counterbalance valve (Fig. 1.6) is applied to create a back pressure or cushioning pressure on the underside of a vertically moving piston to prevent the suspended load from free falling because of gravity while it is still being lowered.

1.7.1 Valve Operation (Lowering)

The pressure setting on the counterbalance valve is set slightly higher than the pressure required to prevent the load from free falling. Due to this back pressure in line A, the actuator piston must force down when the load is being lowered. This causes the pressure in line A to increase, which raises the spring-opposed spool, thus providing a flow path to discharge the exhaust flow from line A to the DCV and then to the tank. The spring-controlled discharge orifice maintains back pressure in line A during the entire downward piston stroke.

1.7.2Valve Operation (Lifting)

Asthe valve is normally closed, flow in the reverse direction (from port B to port A) cannot occur without a reverse free-flow check valve. When the load is raised again, the internal check valve opens to permit flow for the retraction of the actuator.

1.7.3 Valve Operation (Suspension)

When the valve is held in suspension, the valve remains closed. Therefore, its pressure setting must be slightly higher than the pressure caused by the load. Spool valves tend to leak internally under pressure. This makes it advisable to use a pilot-operated check valve in addition to the counterbalance valve if a load must be held in suspension for a prolonged time.

1.8 HydraulicCylinder Sequencing Circuits

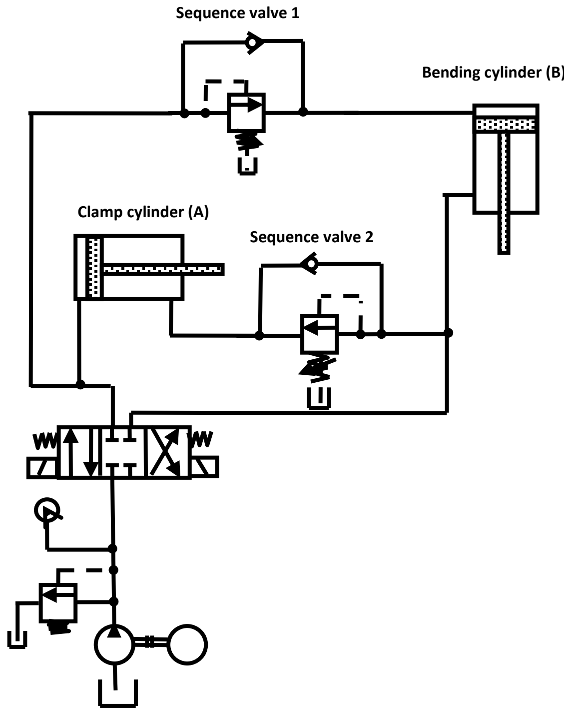


Figure 1.7 Sequencing circuit.

Hydraulic cylinders can be operated sequentially using a sequence valve. Figure 1.7 shows that two sequence valves are used to sequence the operation of two double-acting cylinders. When the DCV is actuated to its right-envelope mode, the bending cylinder (B) retracts fully and then the clamp cylinder (A) retracts.

This sequence of cylinder operation is controlled by sequence valves. This hydraulic circuit can be used in a production operation such as drilling. Cylinder A is used as a clamp cylinder and cylinder B as a drill cylinder. Cylinder A extendsand clamps a work piece. Then cylinder B extends to drive a spindle to drill a hole. Cylinder B retracts the drill spindle and then cylinder A retracts to release the work piece for removal.

1.9Automatic Cylinder Reciprocating System

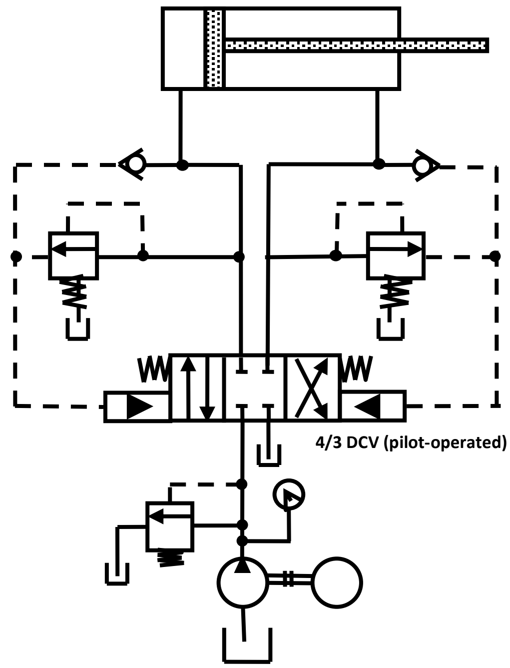


Figure 1.8 Sequencing circuit.

The hydraulic circuit shown in Fig. 1.8 produces continuous reciprocation of a double-acting cylinder using two sequence valves. Each sequence valve senses the completion of stroke by the corresponding build-up pressure. Each check valve and the corresponding pilot line prevent the shifting of the four-way valve until the particular stroke of the cylinder is completed.
The check valves are needed to allow pilot oil to leave either end of the DCV while the pilot pressure is applied to the opposite end. This permits the spool of the DCV to shift as required.

1.10Locked Cylinder Using Pilot Check Valves

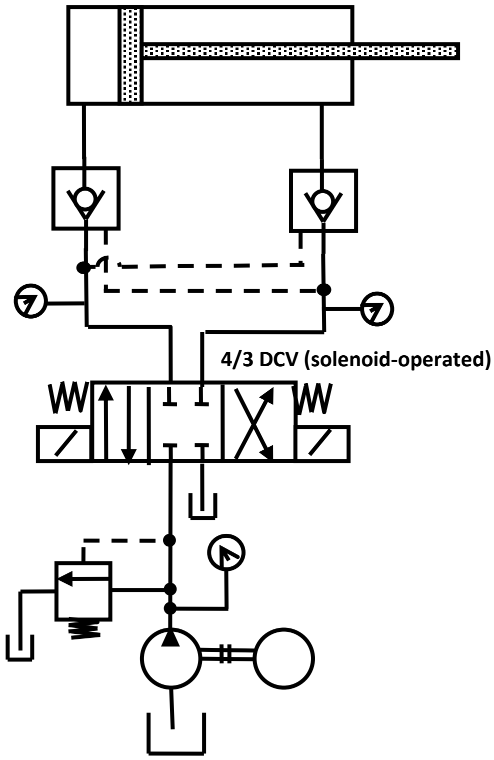


Figure 1.9 Locked cylinders with pilot check valves.

A check valve (Fig. 1.9) blocks flow in one direction but allows free flow in the opposite direction. A pilot-operated check valve permits flow in the normally blocked opposite direction when pilot pressure is applied at the pilot pressure port of the valve.

Pilot-operated check valves are used to lock the cylinder, so that its piston cannot be moved by an external force. The cylinder can be extended and retracted by the DCV. If regular check valves are used, the cylinder could not extend or retract. External force acting on the piston rod does not move the piston in either direction thus locking the cylinder.

1. 11Cylinder Synchronizing Circuits

In industry, there are instances when a large mass must be moved, and it is not feasible to move it with just one cylinder. In such cases we use two or more cylinders to prevent a moment or moments that might distort and damage the load. For example, in press used for molding and shearing parts, the platen used is very heavy. If the platen is several meter wide, it has to be of very heavy construction to prevent the damage when it is pressed down by a single cylinder in the middle. It can be designed with less material if it is pressed down with two or more cylinders. These cylinders must be synchronized. There are two ways that can be used to synchronize cylinders: Parallel and series.

1.11.1 Cylinders in Parallel

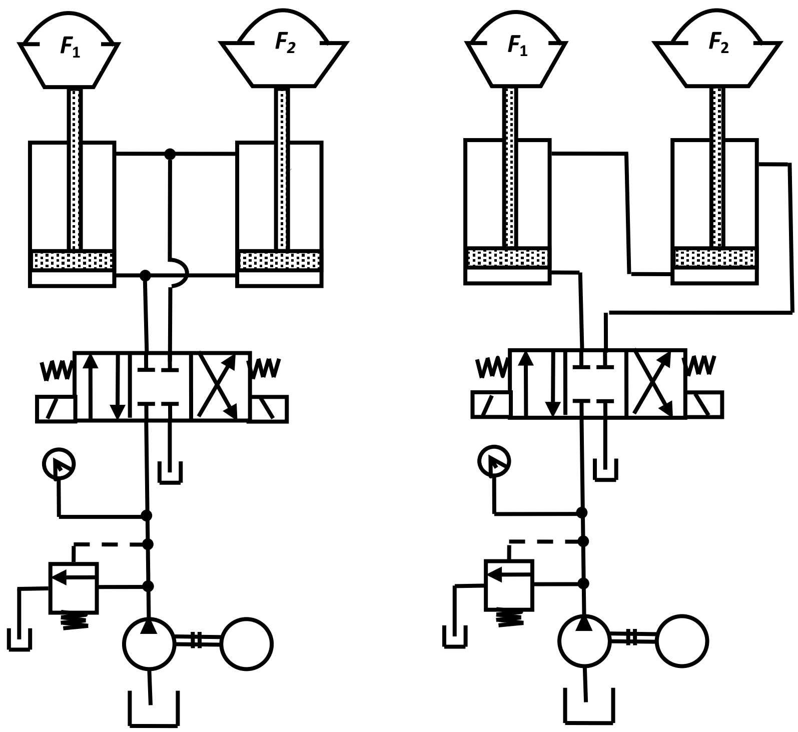


Figure 1.10 Cylinders in parallel and series.

Figure 1.10shows a hydraulic circuit in which two cylinders are arranged in parallel. When the two cylinders are identical, the loads on the cylinders are identical, and then extension and retraction are synchronized. If the loads are not identical, the cylinder with smaller load extends first. Thus, the two cylinders are not synchronized. Practically, no two cylinders are identical, because of packing(seals)friction differences. This prevents cylinder synchronization for this circuit.

1.11.2 Cylinders in Series

During the extending stroke of cylinders, fluid from the pump is delivered to the blank end of cylinder 1. As cylinder 1 extends, fluid from its rod end is delivered to the blank end of cylinder 2 causing the extension of cylinder 2. As cylinder 2 extends, fluid from its rod end reaches the tank.
For two cylinders to be synchronized, the piston area of cylinder 2 must be equal to the difference between the areas of piston and rod for cylinder 1. Thus, applying the continuity equation,

$$Q_{\text{out (cylinder1)}} = Q_{\text{in (cylinder 2)}}$$

we get

$$(A_{p1} - A_{r1})v_1 = A_{p2}v_2$$

For synchronization, $v_1 = v_2$. Therefore,

$$(A_{p1} - A_{r1}) = A_{p2} \qquad (1.1)$$

The pump must deliver a pressure equal to that required for the piston of cylinder 1 by itself to overcome loads acting on both extending cylinders. We know that the pressure acting at the blank end of cylinder 2 is equal to the pressure acting at the rod end of cylinder 1.

Forces acting on cylinder 1 give

$$p_1 A_{p1} - p_2 (A_{p1} - A_{r1}) = F_1$$

Forces acting on cylinder 2 give

$$p_2 A_{p2} - p_2 (A_{p2} - A_{r2}) = F_2$$

UsingEq. (1.1)and noting that $p_3 = 0$ (it is connected to the tank), we have

$$p_1 A_{p1} - p_2 (A_{p2}) = F_1 \qquad (1.2)$$

$$p_2 (A_{p2}) - 0 = F_2 \qquad (1.3)$$

Now, Eq. (1.2) + Eq. (1.3) gives

$$p_1 A_{p1} = F_1 + F_2 \qquad (1.4)$$

If Eqs. (1.1) and (1.4) are met in a hydraulic circuit, the cylinders hooked in series operate in synchronization.

1.12SpeedControl of a Hydraulic Cylinder

The speed control of a hydraulic cylinder circuit can be done during the extension stroke using a flow-control valve (FCV). This is done on a meter-in circuit and meter-out circuit as shown in Fig. 1.11. Refer to Fig. 1.11(a). When the DCV is actuated, oil flows through the FCV to extend the cylinder. The extending speed of the cylinder depends on the FCV setting. When the DCV is deactivated, the cylinder retracts as oil from the cylinder passes through the check valve. Thus, the retraction speed of a cylinder is not controlled. Figure 1.11(b) shows meter-out circuit; when DCV is actuated, oil flows through the rod end to retract the cylinder.

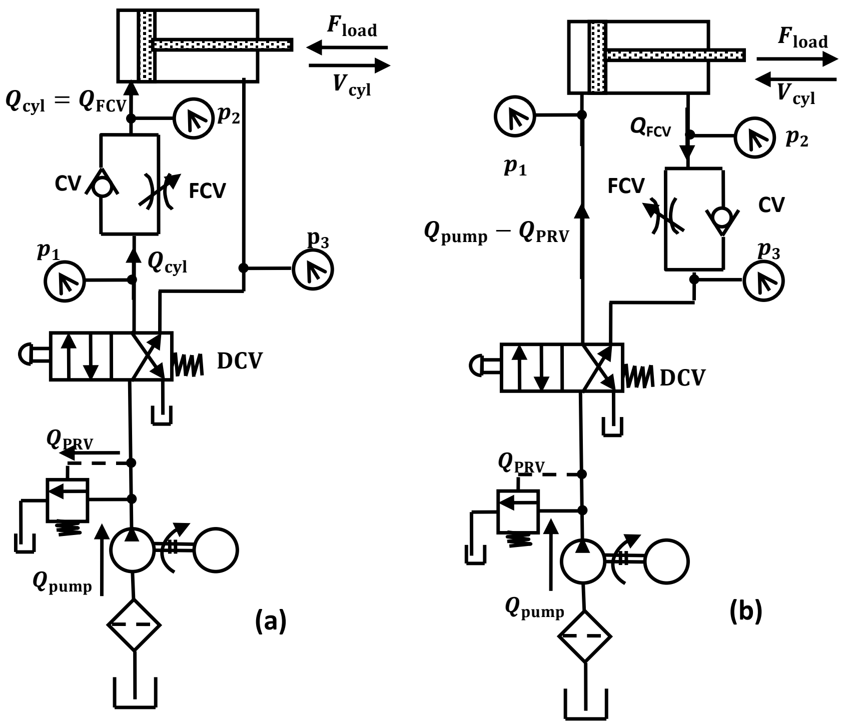


Figure 1.11 Speed control of cylinders:(a) Meter in and (b)meter out.

1.12.1 Analysis of Extending Speed of Cylinder (Controlled)

When theFCV is fully open during extension, all the flow from the pump goes to the cylinder to produce a maximum cylinder speed.When the FCV is partially closed, its pressure drop increases across the FCV. This causes an increase in pressure p_1. If closing is continued, pressure p_1 reaches and exceeds the cracking pressure of the PRV. This results in a slower cylinder speed because part of the pump flow goes through the PRV.

The flow rate to the cylinder Q_{cyl} is given by

$$Q_{cyl} = Q_{pump} - Q_{PRV} \tag{1.5}$$

where Q_{pump} is the flow rate of the pump and Q_{PRV} is the flow rate through the PRV. The flow rate through the FCV is given by

$$Q_{FCV} = C_V \sqrt{\frac{\Delta p}{SG}} = C_V \sqrt{\frac{p_1 - p_2}{SG}} \tag{1.6}$$

where Δp is the pressure drop across the FCV, C_V is the capacity coefficient of the FCV and SG is the specific gravity of oil.The pressure p_1 can be determined by summing forces on the hydraulic cylinder:

$$p_2 A_p = F_{load}$$

$$\Rightarrow p_2 = \frac{F_{load}}{A_p}$$

The extending velocity (controlled speed) of the cylinder is given by

$$v_{\text{cyl}} = \frac{Q_{\text{cyl}}}{A_{\text{p}}} = \frac{Q_{\text{FCV}}}{A_{\text{p}}}$$

$$\Rightarrow v_{\text{cyl}} = \frac{C_{\text{V}}}{A_{\text{p}}}\sqrt{\frac{p_{\text{PRV}} - \frac{F_{\text{load}}}{A_{\text{p}}}}{\text{SG}}}$$

1.12.2 Meter-In Versus Meter-Out Flow-Control Valve Systems

In Section 1.12, the FCV is placed in the line leading to the inlet port of the cylinder. Thus, it is called the meter-in control of speed. Meter-in flow controls the oil flow rate into the cylinder.

A meter-out flow control system is one in which the FCV is placed in the outlet line of the hydraulic cylinder. Thus, a meter-out flow control system controls the oil flow rate out of the cylinder.

Meter-in systems are used primarily when the external load opposes the direction of motion of the hydraulic cylinder. When a load is pulled downward due to gravity, a meter-out system is preferred. If a meter-in system is used in this case, the load would drop by pulling the piston rod, even if the FCV is completely closed.

One drawback of a meter-out system is the excessive pressure build-up in the rod end of the cylinder while it is extending. In addition, an excessive pressure in the rod end results in a large pressure drop across the FCV. This produces an undesirable effect of a high heat generation rate with a resulting increase in oil temperature.

1.12.3 Speed Control of a Hydraulic Motor

Figure 1.12shows the speed control circuit of a hydraulic motor using a pressure-compensated FCV.The operation is as follows:

- In a spring-centered position of the tandem four-way valve, the motor is hydraulically blocked.
- When the valve is actuated to the left envelope, the motor rotates in one direction. Its speed can be varied by adjusting the throttle of the FCV. Thus, the speed can be infinitely varied and the excess oil goes through the PRV.
- When the valve is deactivated, the motor stops suddenly and becomes locked.
- When the right envelope is in operation, the motor turns in the opposite direction. The PRV provides overload protection if, for example, the motor experiences an excessive torque load.

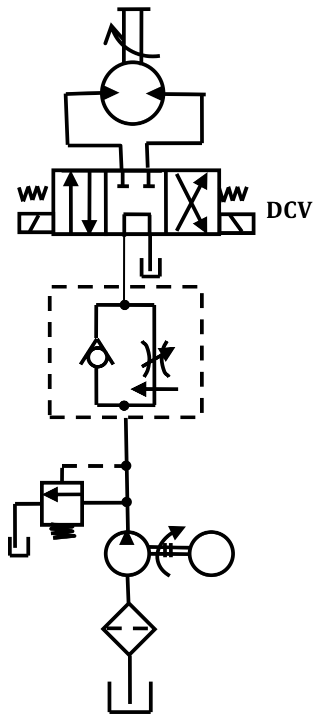


Figure 1.12 Speed control of a motor.

1.13Fail-Safe Circuits

Fail-safe circuits are those designed to prevent injury to the operator or damage to the equipment. In general, they prevent the system from accidentally falling on an operator and also prevent overloading of the system. In following sections we shall discuss two fail-safe circuits: One is protection from inadvertent cylinder extension and other is fail-safe overload protection.

1. **Protection from inadvertent cylinder extension:** Figure 1.13 shows a fail-safe circuit that is designed to prevent the cylinder from accidentally falling in the event when a hydraulic line ruptures or a person inadvertently operates the manual override on the pilot-actuated DCV when the pump is not working. To lower the cylinder, pilot pressure from the blank end of piston must pilot open the check valve to allow oil to return through the DCV to the tank. This happens when the push button is actuated to permit the pilot pressure actuation of DCV or when the DCV is directly manually actuated when the pump operates. The pilot-operated DCV allows free flow in the opposite direction to retract the cylinder when this DCV returns to its offset mode.

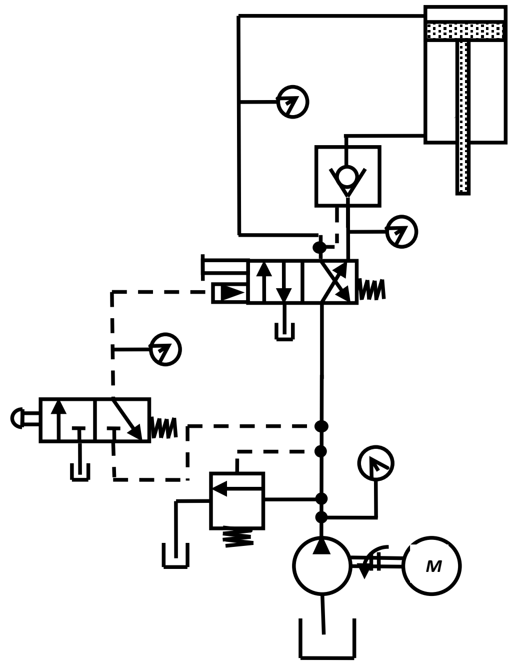


Figure 1.13 Fail-safe circuits – inadvertent cylinder extension.

2. Fail-Safe System with Overload Protection: Figure 1.14 shows a fail-safe system that provides overload protection for system components. The DCV V_1 is controlled by the push-button three-way valve V_2. When the overload valve V_3 is in its spring offset mode, it drains the pilot line of valve V_1. If the cylinder experiences excessive resistance during the extension stroke, sequence valve V_4 pilot-actuates overload valve V_3. This drains the pilot line of valve V_1 causing it to return to its spring offset mode. If a person then operates the push-button valve V_2 nothing happens unless overload valve V_3 is manually shifted into its blocked-port configuration. Thus, the system components are protected against excessive pressure due to an excessive cylinder load during its extension stroke.

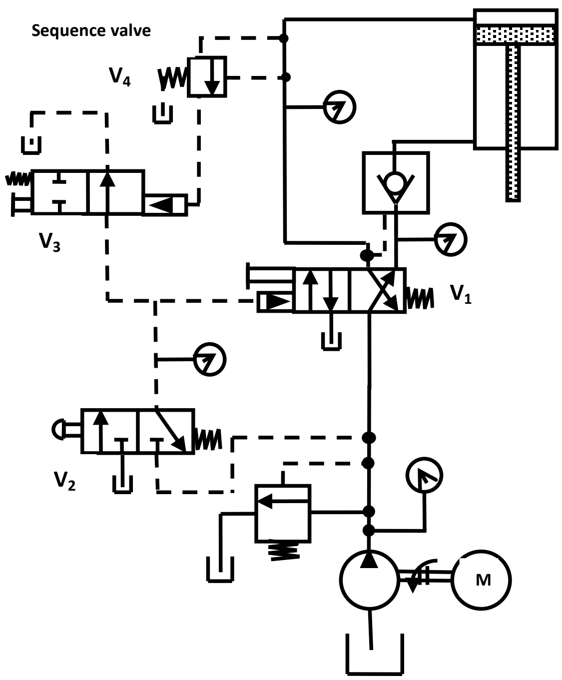


Figure 1.14 Fail-safe circuits –overload protection.

Lecture 25

HYDRAULIC CIRCUIT DESIGN AND ANALYSIS [CONTINUED]

1.14 Circuit for Fast Approach and Slow Die Closing

A machine intended for high volume production has a high piston velocity. If not controlled, the high-speed platen approaching the job instead of making a smooth contact will bang on the job. This is not desirable. In all such cases,**“**rapid traverse and feed circuits” are employed.

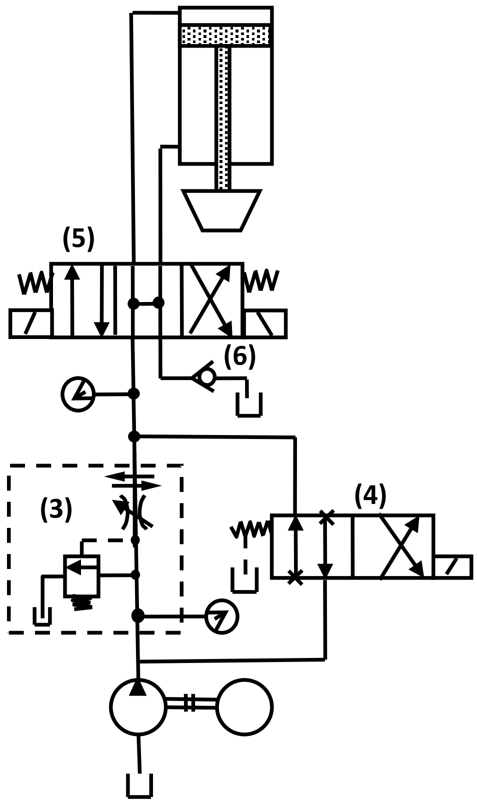


Figure 1.15 Rapid traverse and feed circuit.

In the circuit shown in Fig. 1.15, pump delivery normally passes through FCV(3). During fast approach,the solenoid-operated DCV (4) is energized. This diverts pump delivery to the cap end of the cylinder through valve (4). Full flow is thus available for the actuator to advance at the rated speed. A few millimeters before the platen makes contact with the die, solenoid valve (4) is de-energized forcing the pump delivery to pass through FCV (3). The platen now approaches the die at a controlled speed because the flow to cylinder (6) is now regulated. Directional valves (4) and (5), however, must be energized simultaneously for the approach phase to begin.

Valves (4) to (5) are solenoid-controlled pilot-operated valves intended for handling large flows with minimum pressure drop. While valve (5) requires a 4.5 bar check valve (6) in the return line to develop the pilot pressure required to move the main spool, no such facility is required in the case of valve(4) because the back pressure generated by valve (3) would serve as the pilot pressure for this valve.

1.15Rapid Traverse and Feed, Alternate Circuit

In this circuit (Fig. 1.16), full flow from the pump is allowed to the cap end through the directional valve (3)for fast approach and the rod end oil freely passes through the normally open deceleration valve back to the tank. Near about the end of the stroke, a cam depresses a roller attached to the deceleration valve spool and therefore the valve shifts blocking the flow from the rod end. The flow now has only one pathway back to the tank and that is through FCV (4). The approach speed is now governed by the setting of this valve. During piston retraction stroke, full flow is allowed to the rod end through check valve (6).

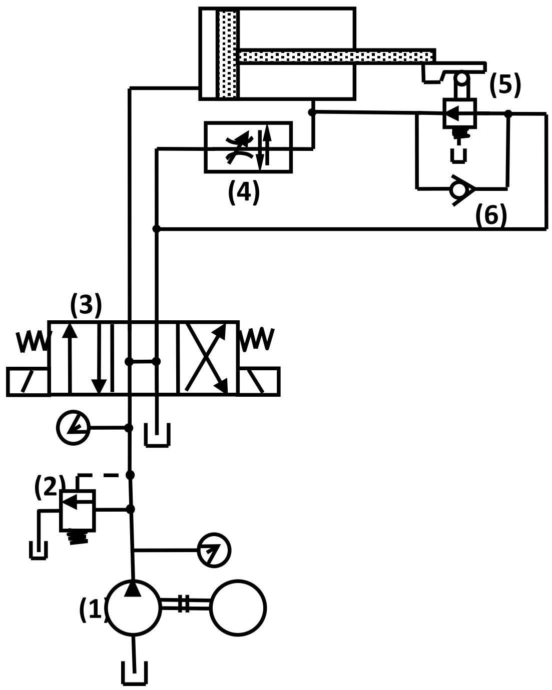


Figure 1.16 Rapid traverse and feed circuit – alternate circuit.

Example 1.1
A double-acting cylinder is hooked up in a regenerative circuit. The relief-valve setting is 105 bar. The piston area is 130 cm^2 and the rod area is 65 cm^2. If the pump flow is 0.0016 m^3/s, find the cylinder speed and load-carrying capacity for the

(a) Extending stroke.
(b) Retracting stroke.

Solution:

(a) We have

$$v_{ext} = \frac{Q_p}{A_r} = \frac{0.0016}{65 \times 10^{-4}} = 0.246 \text{ m/s}$$

$$F_{\text{load-extension}} = pA_r$$
$$= 105 \times 10^5 \times 65 \times 10^{-4}$$
$$= 68250 \text{ N}$$

(b) We have

$$v_{ret} = \frac{Q_p}{A_p - A_r} = \frac{0.0016}{(130-65) \times 10^{-4}} = 0.246 \text{ m/s}$$

$$F_{\text{load-extension}} = p(A_p - A_r)$$
$$= 105 \times 10^5 \times (130 - 65) \times 10^{-4}$$
$$= 68250 \text{ N}$$

Example 1.2

What is wrong with the circuit diagram given in Fig. 1.17?

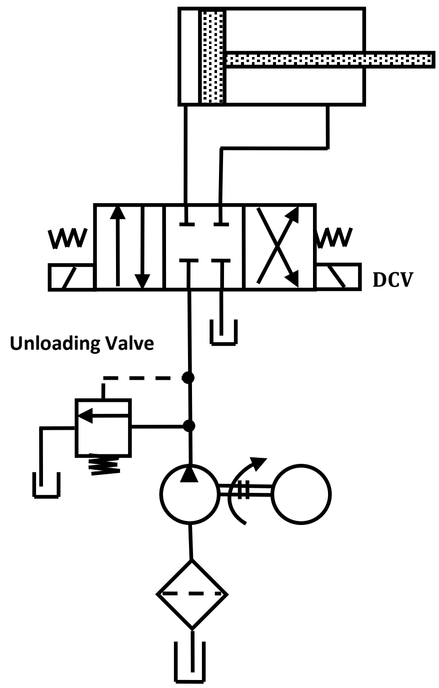


Figure 1.17

Solution: A check valve is needed in the hydraulic line just upstream from where the pilot line to the unloading valve is connected to the hydraulic line. Otherwise the unloading valve would behave like a pressure-relief valve and thus, valuable energy would be wasted.

Example 1.3

What unique feature does the circuit of Figure 1.18 provide in the operation of the hydraulic cylinder?

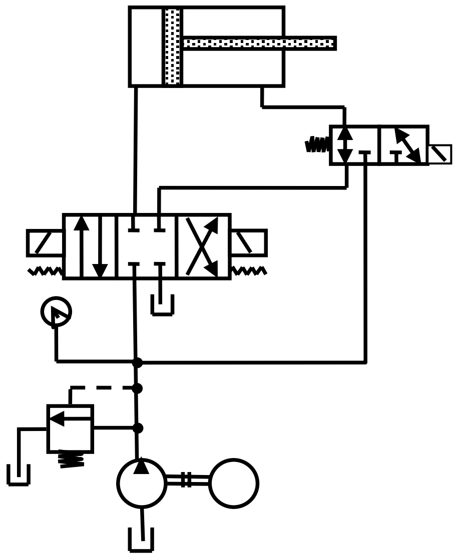


Figure 1.18

Solution:

1. It provides mid-stroke stop and hold of the hydraulic cylinder (during both the extension and retraction strokes) by deactivation of the four-way, three-position DCV.
2. It provides two speeds of the hydraulic cylinder during the extension stroke:

- When the three-way, two-position DCV is unactuated in spring offset mode, extension speed is normal.
- When this DCV is actuated, extension speed increases by the regenerative capability of the circuit.

Example 1.4

For the circuit of Fig. 1.19, give the sequence of operation of cylinders 1 and 2 when the pump is turned ON. Assume that both cylinders are initially fully retracted.

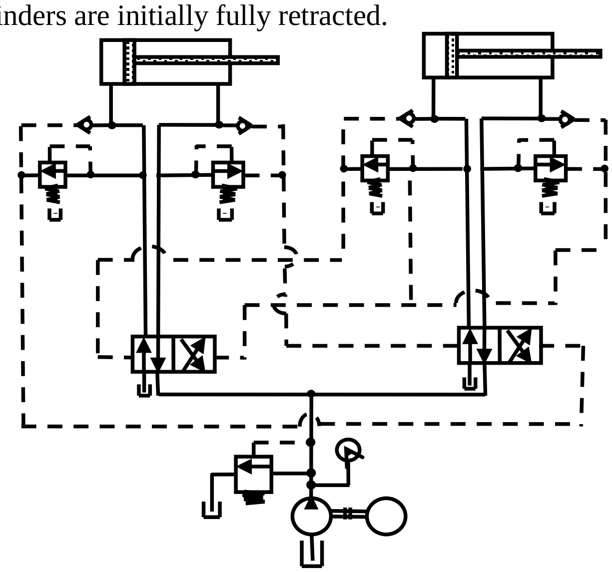


Figure 1.19

Solution: Cylinder 1 extends, cylinder 2 extends. Cylinder 1 retracts, cylinder 2 retracts.

Example 1.5

What safety features does Fig. 1.20possess in addition to a pressure-relief valve. If the load on cylinder 1 is greater than the load on cylinder 2, how will the cylinder move when DCV is shifted into the extending or retracting mode? Explain your answer.

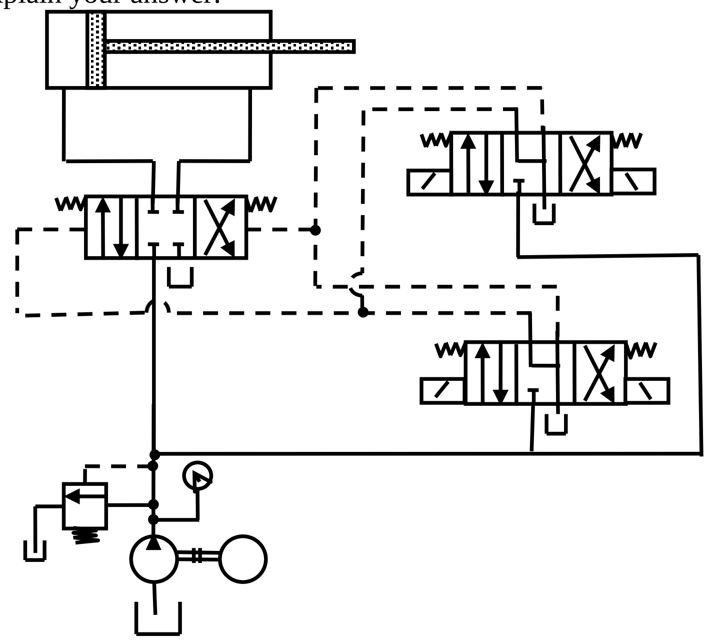


Figure 1.20

Solution: Both solenoid-actuated DCVs must be actuated in order to extend or retract the hydraulic cylinder.

Cylinder 2 will extend through its complete stroke receiving full pump flow while cylinder 1 will not move. The moment cylinder 2 will extend through its complete stroke, cylinder 1 will receive full pump flow and extend through its complete stroke. This is because the system pressure builds up until load resistance is overcome to move cylinder 2 with the smaller load. Then the pressure continues to increase until the load on cylinder 1 is overcome. This then causes cylinder 1 to extend. In retraction mode, the cylinders move in the same sequence.

Example 1.6

Assuming that the two double-rod cylinders of Fig. 1.21 are identical, what unique feature does the circuit in Fig. 1.21 possess.

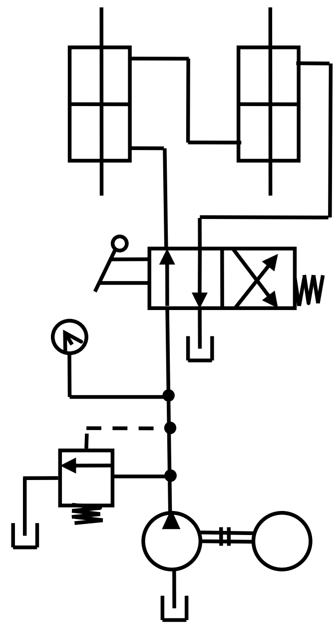


Figure 1.21

Solution: Both cylinder strokes would be synchronized.

Example 1.7
For the hydraulic system is shown in Fig. 1.22
(a) What is the pump pressure for forward stroke if the cylinder loads are 22000 N each and cylinder 1 has the piston area of 65 cm^2 and zero back pressure?
(b) What is pump pressure for retraction stroke (loads pull to right), if the piston and rod areas of cylinder 2 equal to 50 cm^2 and 15 cm^2, respectively, and zero back pressure?
(c) Solve using a back pressure p_3 of 300 kPa instead of zero, the piston area and rod area of cylinder 2 equal 50 and 15 cm^2, respectively.
Solution:

(a)Pressure acting during forward stroke is

$$p_1 = \frac{F_1 + F_2}{A_{p1}} = \frac{22000 + 22000}{65 \times 10^{-4}} = 6.77 \text{ MPa}$$

(b)For cylinder 2 we can write

$$p_3(A_{p2} - A_{r2}) - p_2 A_{p2} = F_2$$

For cylinder 1, force balance gives

$$p_2(A_{p1} - A_{r1}) = F_1$$

But $A_{p2} = A_{p1} - A_{r1}$. So we can write

$$p_2 A_{p2} = F_1$$

and rod side pressure of second cylinder is given by

$$p_3 = \frac{F_1 + F_2}{A_{p2} - A_{r2}} = \frac{22000 + 22000}{35 \times 10^{-4}} = 12570000 \text{ Pa} = 12.57 \text{ MPa}$$

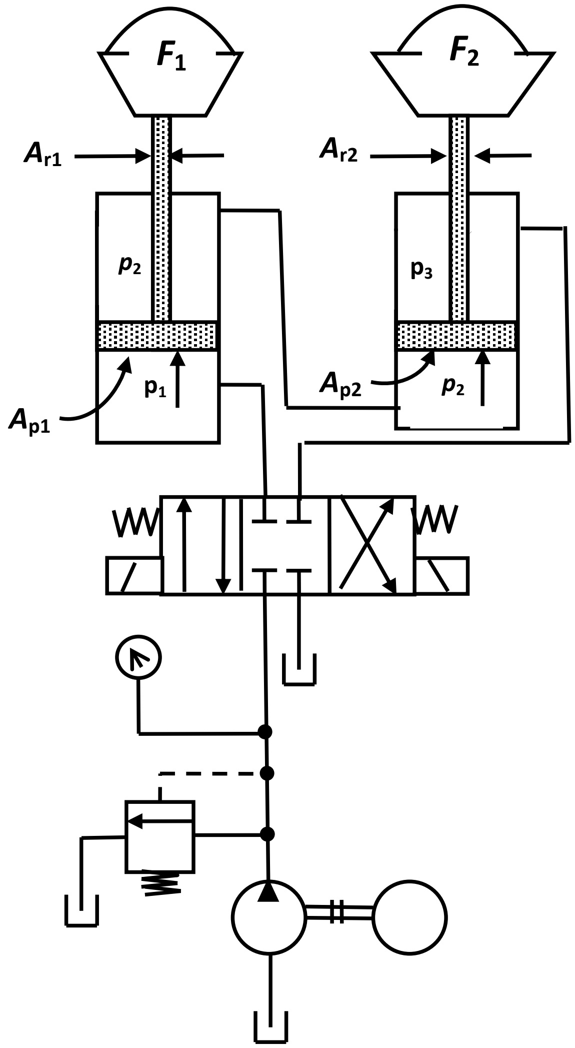


Figure 1.22

(c)For cylinder 1, we have

$$p_1 A_{p1} - p_2 (A_{p1} - A_{r1}) = F_1$$

Similarly for cylinder 2, we have

$$p_2 A_{p2} - p_3 (A_{p2} - A_{r2}) = F_2$$

Adding both equations and noting that $A_{p2} = A_{p1} - A_{r1}$ yield

$$p_1 A_{p1} - p_3 (A_{p2} - A_{r2}) = F_1 + F_2$$

$$\Rightarrow p_1 = \left\{ \frac{F_1 + F_2 + p_3 (A_{p2} - A_{r2})}{A_{p1}} \right\}$$

$$\Rightarrow p_1 = \left\{ \frac{22000 \text{ N} + 22000 \text{ N} + 300000 \text{ N/m}^2 (50 - 15) \text{ cm}^2 \times 10^{-4} \text{ m}^2}{65 \text{ cm}^2 \times 10^{-4} \text{ m}^2} \right\}$$

$$\Rightarrow p_1 = 6.93 \text{ MPa}$$

Example 1.8

For the double-pump system in Fig. 1.23, what should be pressure setting of the unloading valve and pressure-relief valve under the following conditions:

(a) Sheet metal punching operation requires a force of 8000 N.
(b) A hydraulic cylinder has a 3.75 cm diameter piston and a 1.25 cm diameter rod.
(c) During the rapid extension of the cylinder, a frictional pressure loss of 675 kPa occurs in the line from the high-flow pump to the blank end of the cylinder. During the same time, a 350 kPa pressure loss occurs in the return line from the rod end of the cylinder to the oil tank. Frictional pressure losses in these lines are negligibly small during the punching operation.
(d) Assume that the unloading valve and relief-valve pressure setting (for their full pump flow requirements) should be 50% higher than the pressure required to overcome frictional pressure losses and the cylinder punching load, respectively.

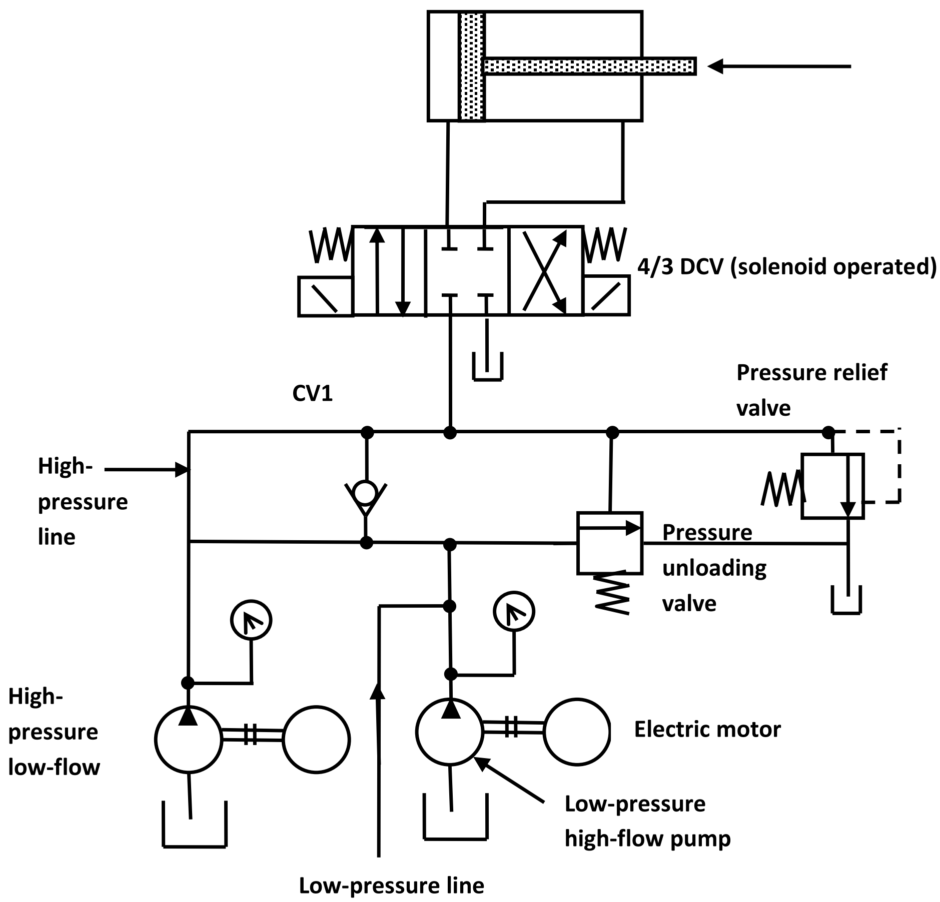


Figure 1.23

Solution:

Unloading valve: Back pressure force on the cylinder equals pressure loss in the return line times the effective area of the cylinder $(A_p - A_r)$:

$$F_{\text{back pressure}} = 350000 \frac{\text{N}}{\text{m}^2} \times \frac{\pi}{4}(0.0375^2 - 0.0125^2)\ \text{m}^2 = 344\ \text{N}$$

Pressure at the blank end of the cylinder required to overcome back pressure force equals the back pressure force divided by the area of the cylinder piston:

$$p_{\text{cyl blank end}} = \frac{344\ \text{N}}{\frac{\pi}{4}(0.0375^2)\text{m}^2} = 311\ \text{kPa}$$

Thus,the pressure setting of unloading valve equals

$$1.50(675 + 311)\ \text{kPa} = 1480\ \text{kPa}$$

Pressure relief valve: Pressure required to overcome the punching operation equals the punching load divided by the area of the cylinder piston:

$$p_{\text{punching}} = \frac{8000\ \text{N}}{\frac{\pi}{4}(0.0375^2)\text{m}^2} = 7240\ \text{kPa}$$

Thus, the pressure setting of pressure-relief valve equals

$$1.50 \times 7240\ \text{kPa} = 10860\ \text{kPa}$$

Example1.9

Design a suitable hydraulic circuit to raise and lower a load of magnitude 10000 kgf at a speed of 100 mm/s. The speed must be equal both during raising and lowering of the load. The load is essentially overrunning. The load must be lowered gradually onto the platform. Calculate the flow through the control valves and indicate the pressure gauge readings both at the cap end and at the rod end during raising and lowering. Explain your reasons for your choice of the hydraulic components. Neglect mechanical and hydraulic losses. Assume 100 mm bore for the cylinder and a rod diameter = 45 mm.

Solution: In any double-acting cylinder the rod end area is smaller than the cap end area to the extent of the piston rod cross-sectional area and so the pressure required to raise/lower the load is derived from the rod end area. Accordingly, the pressure required to raise the load is

$$\frac{10000}{\frac{\pi(10^2 - 4.5^2)}{4}} = 160\ \text{kgf/cm}^2 = 160\ \text{bar}$$

The relief-valve setting pressure is 175 bar.

The cap end area =.7584$(10)^2$ = 75.84 cm^2.

If the cylinder has to extend and retract at the rate of 100 mm/s,the flow required at the cap end of the cylinder is

$$(75.84 \times 10) = 758.4\ \text{cm}^3/\text{s or } 47\ \text{LPM}$$

Flow required at the rod end would be (62.6 ×10) = 626 cm^3/s or 37.5 LPM.

Referring to the circuit in Fig. 1.24, a constant delivery pump with a pressure rating of 175 bar capable of delivering 50 LPM has been chosen. A variable delivery pump would not help because both velocity and pressure are constant throughout the cycle. It is required that the piston must travel both during extension

and retraction at the same speed. Thus, flow control valve (1) is used on the cap end of the cylinder because pump supply is constant but cap end and rod end areas differ. Since it is an overrunning load, a flow control valve became necessary (2) on the rod end (meter-out flow control).
Because it is required that the load must be positioned on the platform gradually, flow control (3) and solenoid-operated DCV (4) become necessary. Toward the end of the stroke, the load makes contact with a limit switch. This energizes valve (4) to divert rod end flow through the flow control valve (3) so that the load is decelerated from 100 mm/s to 30 mm/s. In order to ensure accurate speed control pressure- and temperature-compensated flow, flow control valves were chosen. Flow through the flow control valve (3) during deceleration would be = 188 cm^3/s or 11 LPM.
While raising the load, the required flow to the rod end of the cylinder is 37.5 LPM. But the pump is supplying 50 LPM. The excess flow must pass over the relief valve which is set at 175 bar. The relief-valve setting pressure of 175 bar creates a retracting force on the rod end equaling

$$(175 \times 62.6) = 10955 \text{ kgf}$$

Of this, 10000 kgf is required just to balance the load.

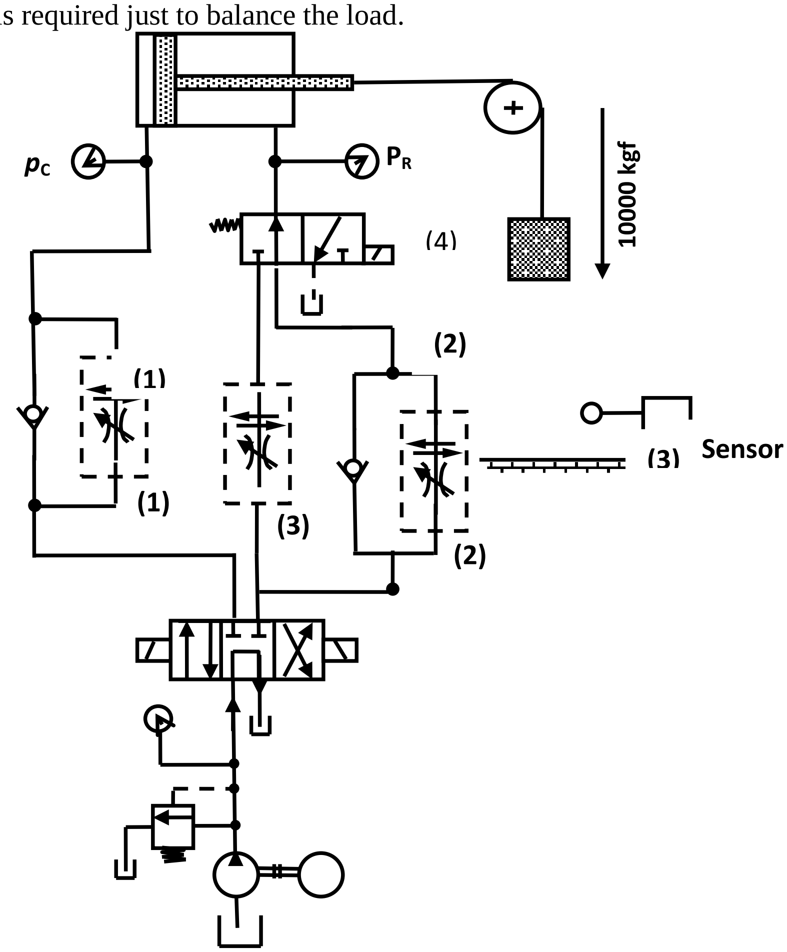


Figure 1.24

The remaining 955 kgf acting in the retracting direction has to be balanced by the backpressure due to the flow control valve (1). Consequently, the pressure gauge P_c at the cap end during retraction would read 955/62.6 = 15 bar. When the load is lowered at a speed of 100 mm/s, the cylinder extends at a velocity of 100 mm/s. The flow entering the cap end of the cylinder is 47 LPM, which is less than the pump delivery, which is 50 LPM.
The pressure gauge P_c would read 175 bar because the extra flow must be dumped over the relief valve. In this operating condition, the extension force (175 ×75.84) = 13744 kgf, together with the load force = 10000 kgf, tries to extend the cylinder.

According to Newton's first law of motion the net force must be equal to zero for an object moving at a constant velocity, neglecting friction. So the balancing force at the rod end should be 23744 kgf. Therefore, the pressure gauge P_r at the rod would read 23774 / 62.6 = 379 bar. At the end of high-speed extension solenoid valve (2) is energized to decelerate the load from 100 mm/s to 30 mm/s. The time duration around which this occurs depends on the valve response time that can be assumed as 20 ms or 0.020 s.
The deceleration

$$a = \frac{V_f - V_i}{\delta t}$$

where final velocity $V_f = 30$ mm/s, initial velocity $V_i = 100$ mm/s, δtis the time element = 0.020 s during which time the change occurs. Substituting the relevant values, we obtain the value of a as 3500 mm/s^2.Therefore, the force required to decelerate the cylinder is

$$F = ma$$

Now m= 10000/9.81 = 1019 kg . So

$$F = 1019 \times 3.5 = 23566 \text{ kgf}$$

Therefore

$$p_R = (23744 + 3566)/62.6 = 436 \text{ bar}$$

Example 1.10

For the fluid power system shown in Fig. 1.25,

(a) Determine the external loads F_1 and F_2 that each hydraulic cylinder can sustain while moving in an extending direction. Take frictional pressure losses into account. The pump produces a pressure of increase of 6.90 MPa from the inlet port to the discharge port and a flow rate of 0.00252 m^3/s. The following data are applicable:

Kinematic viscosity of oil	0.0000930 m^2/s
Specific weight of oil	7840 N/m^3
Cylinder piston diameter	0.203 m
Cylinder rod diameter	0.102 m
All elbows are 90° with k factor	0.75

Pipe lengths and diameters are given:

Pipe Number	Length (m)	Diameter	Pipe number	Length(m)	Diameter
1	1.83	0.0508	6	3.05	0.0254
2	9.15	0.0317	7	3.05	0.0254
3	6.10	0.0317	8	12.2	0.0317
4	3.05	0.0254	9	12.2	0.0317
5	3.05	0.0254			

(b)Determine the heat generation rate.

(c) Determine the extending and retracting speeds of cylinder.

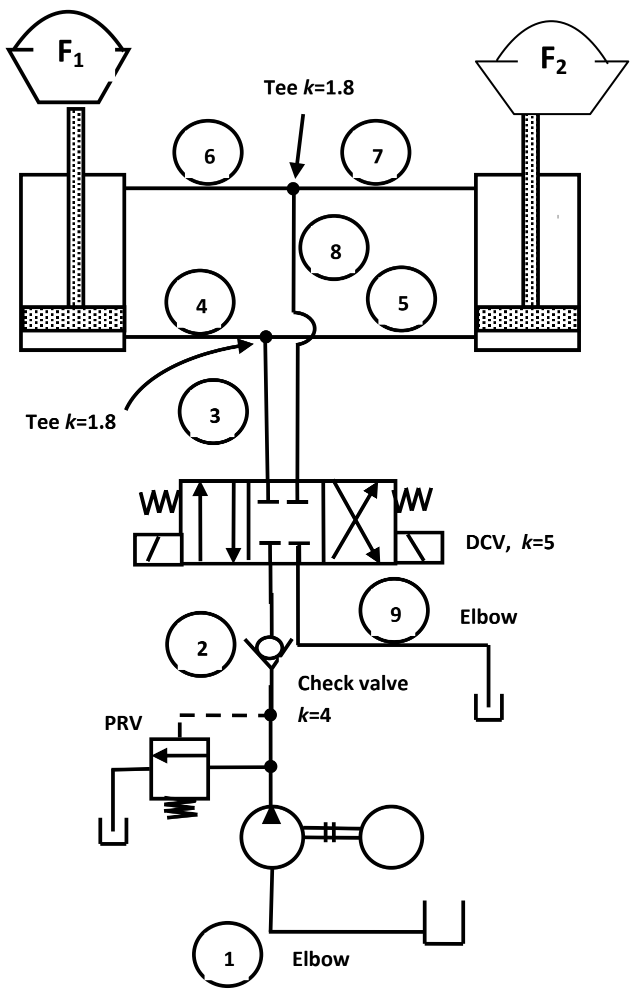


Figure 1.25

Solution:

(a) Cylinders 1 and 2 are identical and are connected by identical lines. Therefore, they receive equal flows and sustain equal loads ($F_1 = F_2$).

Velocity is calculated from discharge and area as

$$v = \frac{Q\,(\mathrm{m^3/s})}{A\,(\mathrm{m^2})}$$

Head loss in the systems is given by

$$H_\mathrm{L} = \sum_{1}^{13}\left(\frac{f\,L_\mathrm{p}}{D_\mathrm{p}} + K\right)\frac{v^2}{2g}$$

Reynolds number is given by

$$\mathrm{Re} = \frac{VD\rho}{\mu} = \frac{VD}{\mu/\rho} = \frac{VD}{\nu}$$

Flow through path 4 (Fig. 1.26) is given by

$$Q_4 = \frac{0.00252}{2} = 0.00126\,\mathrm{m^3/s}$$

Flow through path 6 (Fig 1.26) is given by

$$Q_6 = 0.00126 \times \frac{(0.203^2 - 0.102^2)}{0.203} = 0.000945\,\mathrm{m^3/s}$$

Similarly for paths 8 and 9 we can write

$$Q_8 = Q_9 = 2 \times 0.000945 = 0.00189\ \mathrm{m^3/s}$$

Velocity calculation:

$$v_1 = \frac{0.00252\ (\mathrm{m^3/s})}{\frac{\pi(0.0508)^2}{4}\ (\mathrm{m^2})} = 1.24\ \mathrm{m/s}$$

$$v_{2,3} = \frac{0.00252\ (\mathrm{m^3/s})}{\frac{\pi(0.0317)^2}{4}\ (\mathrm{m^2})} = 3.19\ \ \mathrm{m/s}$$

$$v_4 = \frac{0.00126\ (\mathrm{m^3/s})}{\frac{\pi(0.0254)^2}{4}\ (\mathrm{m^2})} = 2.49\ \mathrm{m/s}$$

$$v_6 = \frac{0.000945\ (\mathrm{m^3/s})}{\frac{\pi(0.0254)^2}{4}\ (\mathrm{m^2})} = 1.86\ \mathrm{m/s}$$

$$v_{8,9} = \frac{0.00189\ (\mathrm{m^3/s})}{\frac{\pi(0.0317)^2}{4}\ (\mathrm{m^2})} = 2.39\ \mathrm{m/s}$$

Reynolds number calculation:

$$\mathrm{Re}_{(1)} = \frac{1.24 \times 0.0508}{0.000093} = 677$$

$$\mathrm{Re}_{(2,3)} = \frac{3.19 \times 0.0317}{0.000093} = 1087$$

$$\mathrm{Re}_{(4)} = \frac{2.49 \times 0.0254}{0.000093} = 680$$

$$\mathrm{Re}_{(6)} = \frac{1.86 \times 0.0254}{0.000093} = 508$$

$$\mathrm{Re}_{(8,9)} = \frac{2.39 \times 0.0317}{0.000093} = 815$$

All flows are laminar; hence we can calculate the losses in each branch. The general formula is

$$H_\mathrm{L} = \sum_1^{13} \left(\frac{f\,L_\mathrm{p}}{D_\mathrm{p}} + K \right) \frac{v^2}{2g}$$

where

$$f = \frac{64}{\text{Re}}$$

Hence the losses are

$$H_{L(1)} = \left(\frac{64}{677}\frac{1.83}{0.0508} + 7.5\right)\frac{1.24^2}{2\times 9.81} = 0.33 \text{ m} = 7840\times 0.33\,\text{Pa} = 2560\,\text{Pa}$$

$$H_{L(2)} = \left(\frac{64}{1087}\frac{9.15}{0.0317} + 4\right)\frac{3.19^2}{2\times 9.81} = 10.9 \text{ m} = 85500\,\text{Pa}$$

$$H_{L(3)} = \left(\frac{64}{1087}\frac{12.2}{0.0317} + 6.8\right)\frac{3.19^2}{2\times 9.81} = 15.3 \text{ m} = 120000\,\text{Pa}$$

$$H_{L(4)} = \left(\frac{64}{680}\frac{3.05}{0.0254} + 1.8\right)\frac{2.49^2}{2\times 9.81} = 4.14 \text{ m} = 32500\,\text{Pa}$$

$$H_{L(6)} = \left(\frac{64}{508}\frac{3.05}{0.0254} + 0\right)\frac{1.86^2}{2\times 9.81} = 2.67 \text{ m} = 20900\,\text{Pa}$$

$$H_{L(8)} + H_{L(9)} = \left(\frac{64}{815}\frac{6.1+12.2}{0.0317} + 5.75\right)\frac{2.39^2}{2\times 9.81} = 12.8 \text{ m} = 100500\,\text{Pa}$$

Total force can now be calculated as

$$F_1 = F_2 = [(6900000) - (2560 + 85500 + 120000 + 32500)] \times \frac{\pi(0.203^2)}{4} - \left[(20900 + 100500) \times \frac{\pi(0.203^2 - 0.102^2)}{4}\right]$$

$$F_1 = F_2 = [216000] - [2940] = 213000 \text{ N}$$

(b) We have

Heat generation rate (power loss in W) = Pressure × Discharge

$$= \{(2560 + 85500 + 120000) \times (0.00252) + (2 \times 20900 \times 0.000945) + (2 \times 32500 \times 0.00126) + (100500 \times 0.00189)\}$$

= 524 + 39.5 + 81.9 + 190 = 835W = 0.835 kW

(c) Cylinder piston diameter = 0.203 m

Area of piston $(A_\text{p}) = \dfrac{\pi(0.203^2)}{4}\ \text{m}^2$

Cylinder rod diameter = 0.102 m

Area of rod = $\dfrac{\pi(0.102^2)}{4}\ \text{m}^2$

Annulus area $= A_\text{annulus} = \dfrac{\pi(0.203^2)}{4} - \dfrac{\pi(0.102^2)}{4}$

Now

$$v = \frac{Q_{\text{cyl}}\ (\text{m}^3/\text{s})}{A\,(\text{m}^2)}$$

where each cylinder receives one half of pump flow because of the configuration of cylinder. Extension velocity is given by

$$v_{\text{ext}} = \frac{Q_{\text{blank end}}\ (\text{m}^3/\text{s})}{A_{\text{p}}\ (\text{m}^2)}$$
$$= \frac{0.00126}{\frac{\pi(0.203^2)}{4}} = 0.0389\,\text{m/s}$$

Retracting velocity is given by

$$v_{\text{ret}} = \frac{Q\ (\text{m}^3/\text{s})}{A_{\text{annulus}}\ (\text{m}^2)}$$
$$= \frac{0.00253}{\frac{\pi(0.203^2)}{4} - \frac{\pi(0.102^2)}{4}}$$
$$= 0.0521\ \text{m/s}$$

Example 1.11

Figure 1.26 shows a regenerative circuit in which an 18.65 kW electric motor drives a 90% efficient pump. The pump discharge pressure is 6897 kPa. Take frictional pressure losses into account.
(a)Determine the external load *F* that the hydraulic cylinder can sustain in the regenerative mode (spring-centered position of DCV).
(b) Determine the heat generation rate due to frictional pressures losses in the regenerative mode.
(c) Determine the cylinder speed for each position of the DCV.
The following data are applicable:

Kinematic viscosity of oil	0.0000930 m^2/s
Specific weight of oil	7850 N/m^3
Cylinder piston diameter	0.203 m
Cylinder rod diameter	0.102 m
All elbows are 90° with *k* factor	0.75

Pipe lengths and diameters are given

Pipe number	Length (m)	Diameter
1	0.61	0.0508
2	6.10	0.0445
3	9.15	0.0445
4	9.15	0.0445
5	6.10	0.0445

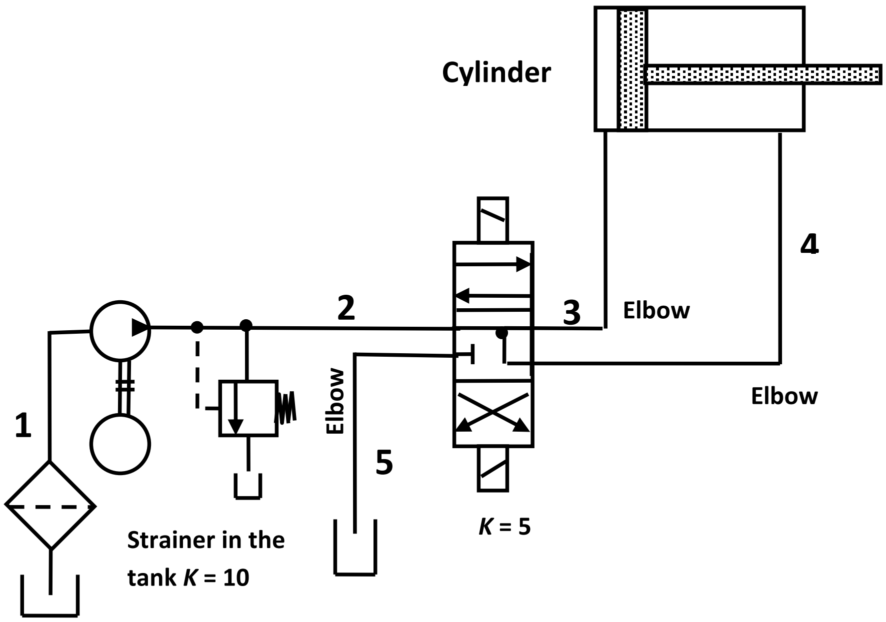


Figure 1.26

Solution:

(a) Determination of external load, considering all losses: Let us first calculate the flow rate at different branches as shown in Fig. 1.27. Before we calculate the losses, we calculate the pump power as

$$\text{Pump power} = \eta \times P_{\text{pump}} = 0.90 \times 18.65 = 16.79 \text{ kW}$$

The flow rate is given by

$$Q_{\text{pump}} = \frac{16.79 \text{ kW}}{6897 \text{ kPa}} = 0.00243 \text{ m}^3/\text{s}$$

We can write the force balance

$$F_{\text{regen}} = p_{\text{blank end}} A_{\text{P}} - p_{\text{rod end}} A_{\text{annulus}}$$

Now

$$Q_{\text{pump}} = Q_1 = Q_2 = 0.00243 \text{ m}^3/\text{s}$$

From the derivation of regenerative circuits, we can write

$$
\begin{aligned}
Q_3 &= \frac{A_{\text{p}}}{A_{\text{r}}} Q_{\text{pump}} \\
&= \frac{\pi(0.203)^2 / 4}{\pi(0.102)^2 / 4} \times 0.00243 \\
&= 0.00972 \text{ m}^3/\text{s}
\end{aligned}
$$

$$Q_4 = \frac{A_p - A_r}{A_r} Q_{\text{pump}}$$
$$= \frac{\pi[(0.203)^2 - (0.102)^2]/4}{\pi(0.102)^2/4} \times 0.00243$$
$$= 0.00729 \text{ m}^3/\text{s}$$

Velocity calculation

$$v_1 = \frac{0.00243 \text{ (m}^3/\text{s)}}{\frac{\pi(0.0508^2)}{4} \text{ (m}^2)} = 1.20 \text{ m/s}$$
$$v_2 = \frac{0.00243 \text{ (m}^3/\text{s)}}{\frac{\pi(0.0445^2)}{4} \text{ (m}^2)} = 1.56 \text{ m/s}$$
$$v_3 = \frac{0.00972 \text{ (m}^3/\text{s)}}{\frac{\pi(0.0445^2)}{4} \text{ (m}^2)} = 6.24 \text{ m/s}$$
$$v_4 = \frac{0.00729 \text{ (m}^3/\text{s)}}{\frac{\pi(0.0445^2)}{4} \text{ (m}^2)} = 4.69 \text{ m/s}$$

Reynolds number calculation

$$\text{Re}_{(1)} = \frac{1.20 \times 0.0508}{0.000093} = 655$$
$$\text{Re}_{(2)} = \frac{1.56 \times 0.0445}{0.000093} = 746$$
$$\text{Re}_{(3)} = \frac{6.24 \times 0.0445}{0.000093} = 2990$$
$$\text{Re}_{(4)} = \frac{4.69 \times 0.0445}{0.000093} = 2240$$

Assume that all flows are laminar; head losses can be calculated as follows:

$$H_{L(1)} = \left(\frac{64}{655}\frac{6.10}{0.0508} + 10\right)\frac{1.20^2}{2 \times 9.81} = 0.74 \text{ m} = 7850 \times 0.33 \text{ Pa} = 5840 \text{ Pa}$$

$$H_{L(2)} = \left(\frac{64}{746}\frac{6.10}{0.0445} + 5\right)\frac{1.56^2}{2 \times 9.81} = 2.08 \text{ m} = 16300 \text{ Pa}$$

$$H_{L(3)} = \left(\frac{64}{2990}\frac{9.15}{0.0445} + 0.75\right)\frac{6.24^2}{2 \times 9.81} = 10.2 \text{ m} = 80300 \text{ Pa}$$

$$H_{L(4)} = \left(\frac{64}{2240}\frac{9.15}{0.0445} + 0.75\right)\frac{4.69^2}{2\times 9.81} = 7.43\,\mathrm{m} = 58400\,\mathrm{Pa}$$

The force is given by

$$F = [(6897\,\mathrm{kPa}) - (5.84 + 16.3 + 80.3)] \times \frac{\pi(0.203)^2}{4} -$$
$$\left[[6897\,\mathrm{kPa} - (5.84 + 16.3 - 58.4)] \times \frac{\pi[(0.203)^2 - (0.102)^2]}{4} \right]$$

Solving we get

$$F = 220 - 168 \; = 52 \text{ kN}$$

(b) Determination of heat generation rate

Power loss = $\sum Q\Delta p$

= Pipe 1 loss + pump loss+ pipe 2 loss + pipe 3 loss + pipe 4 loss

= 0.00243 × 5.84 + (18.7−16.8) + 0.00243 × 16.3 +0.00972 × 80.3 + 0.00729 × 58.4

= 0.014+1.9+0.04+0.78+0.43

Power loss = Heat generation rate = 3.16 kW

(c) Cylinder speed for each position of DCV

$$Q_{\text{pump}} = Q_1 = Q_2 = 0.00243\ \mathrm{m^3/s}$$

Upper position of DCV

$$v_{\text{ext}} = \frac{Q_{\text{pump}}\,(\mathrm{m^3/s})}{A_{\text{p}}\,(\mathrm{m^2})} = \frac{0.00243}{\dfrac{\pi(0.203^2)}{4}} = 0.0751\ \mathrm{m/s}$$

Spring-centered position of DCV

$$v_{\text{ext}} = \frac{Q_{\text{pump}}\,(\mathrm{m^3/s})}{A_{\text{rod}}\,(\mathrm{m^2})} = \frac{0.00243}{\dfrac{\pi(0.102^2)}{4}} = 0.297\,\mathrm{m/s}$$

Lower position of DCV

$$v_{\text{ret}} = \frac{Q_{\text{pump}}\,(\mathrm{m^3/s})}{A_{\text{annulus}}\,(\mathrm{m^2})} = \frac{0.00243}{\dfrac{\pi(0.203^2)}{4} - \dfrac{\pi(0.102^2)}{4}} = 0.100\,\mathrm{m/s}$$

Example 1.12
For the meter-in flow control valve system of Fig. 1.27, the following data are given:

Desired cylinder speed	0.254 m/s
Cylinder piston diameter	0.508 m
Cylinder load	13340 N
Specific gravity of oil	0.9
Pressure-relief valve setting	6895 kPa

Determine the required capacity coefficient of flow control valve.

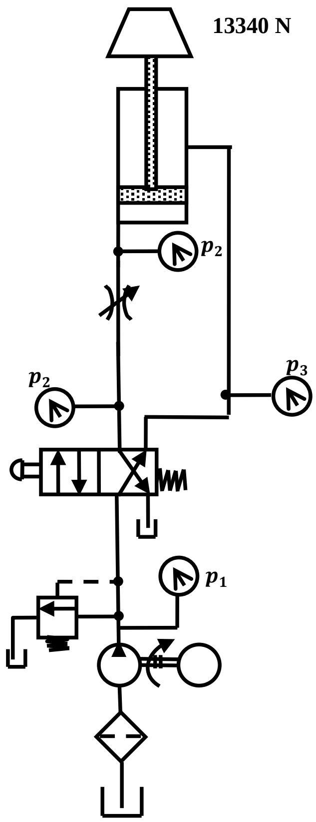


Figure 1.27

Solution: We have

$$C_V = \frac{V_{cyl} \times A_{piston}}{\sqrt{\dfrac{p_{PRV} - (F_{load} / A_{piston})}{SG}}} \tag{1.7}$$

C_V are $\mathrm{LPM}/\sqrt{\mathrm{kpa}}$ .Therefore, we have the following units for the terms in the above equation:

$$Q = v_{cyl} A_p = \mathrm{LPM},\ p_{PRV} = \mathrm{kPa},\ (F_{load})/(A_{piston}) = \mathrm{Pressure} = \mathrm{kPa}$$

The flow rate is given by

$$\begin{aligned}
Q &= v_{\text{cyl}} A_{\text{p}} \\
&= 0.254 \frac{\text{m}}{\text{s}} \times 0.00203\,\text{m}^2 \times \frac{1\text{L}}{0.001\,\text{m}^3} \frac{60\,\text{s}}{1\ \text{min}} \\
&= 30.9\,\text{LPM}
\end{aligned}$$

Now it is given that $p_{\text{PRV}} = 6895\,\text{kPa}$ and

$$\frac{F_{\text{load}}}{A_{\text{piston}}} = \frac{13340\,\text{N}}{0.00203\,\text{m}^2} \times \frac{1\text{kPa}}{1000\,\text{N/m}^2} = 6570\,\text{kPa}$$

Substituting values in Eq. (1.7), we get

$$\begin{aligned}
C_{\text{V}} &= \frac{v_{\text{cyl}} \times A_{\text{piston}}}{\sqrt{\dfrac{p_{\text{PRV}} - (F_{\text{load}} / A_{\text{piston}})}{\text{SG}}}} \\
&= \frac{30.9}{\sqrt{\dfrac{6895 - 6570}{0.9}}} \\
&= \frac{30.9}{19.0} = 1.63\ \frac{\text{LPM}}{\sqrt{\text{kpa}}}
\end{aligned}$$

Lecture 26

HYDRAULIC CIRCUIT DESIGN AND ANALYSIS

Example 1.14

Design a car crushing system. The crushing force required is such that a 15 cm diameter cylinder is required at a working pressure of 126.5 kg/cm^2. Time for crushing is about 10 s and the stroke required to flatten the car is 254 cm. Compare the power required by the circuit without and with accumulator.

Accumulator details:

It is a gas-loaded accumulator

Time taken for charging =5 mm

Initial pressure of charging (pre-charged),p_1 = 85 kg/cm^2

Charged pressure of accumulator,p_2= 210 kg/cm^2

Minimum pressure for crushing, p_3= 126.5 kg/cm^2

Solution: Given

Diameter of piston = 15 cm

Distance traveled in 10 s=254cm

Distance traveled in 1 s= 25.4 cm

Therefore, the stroke velocity, v = 25.4 cm/s

Pressure required to crush= 126.5 kg/cm^2

Circuit requirements without accumulator

$$\text{Area of piston}=(\pi/4)D^2= (\pi/4)15^2= 176.71\ \text{cm}^2= 176.71 \times 10^2\text{mm}^2$$

Volume of flow,

$$V=\text{Area} \times\text{Stroke}=176.71 \times 10^2\times 2540$$
$$=44884340\text{mm}^3 = 4.488 \times 10^{-2}\text{m}^3$$

Flow rate

$$Q = V/10 = 4.488 \times10^2/ 10 = 4.488 \times10^{-3}\text{m}^3/\text{s}$$

Power required

$$P= Q\times p = 4.488 \times10^{-3}\times 126.5\times10\times10^4$$
$$= 56773.2\ \text{W} = 56.77\ \text{kW}$$

If accumulator is not used, the flow rate of 4.488 ×10 m^{-3}/s is required from a pump with capacity of 56.77 kW.

Circuit requirements with accumulator:

Time taken for charging the accumulator=5 mm

Initial pressure of charging (pre-charged),p_1=85 kg/cm^2

Charged pressure of accumulator,p_2 =210 kg/cm^2

Minimum pressure for crushing,p_3=126.5 kg/cm^2

This can be shown by the diagram shown in Fig. 1.29.

The pressure relation in the accumulator can be given as

$$p_1v_1 = p_2v_2 = p_3v_3 \text{ (at a constant temperature)}$$

Let us first equate

$$p_2v_2 = p_3v_3$$

Rearranging we get

$$v_3 = \frac{p_2 v_2}{p_3}$$

Substituting the valves of p_2 and p_3, we get

$$v_3 = \frac{210 \times v_2}{126.5} = 1.66 v_2$$

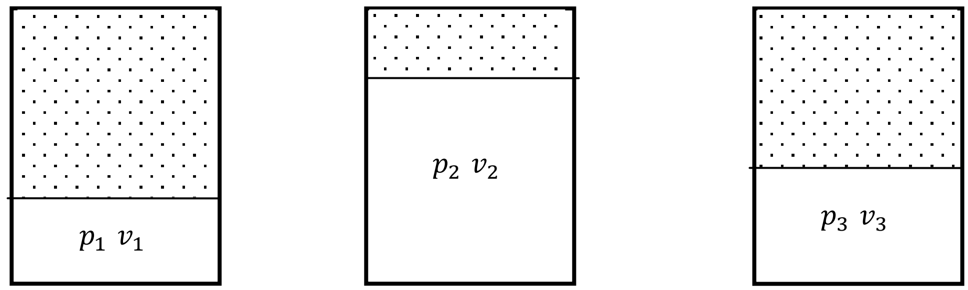


Figure 1.29

We know that the oil supplied by the accumulator after charging is$v_3 - v_2$.
This is used for the cylinder displacement for crushing.
The amount of oil required for crushing with the given constructional details as calculated above is 4488 $\times 10^{-2}$ m^3.
Therefore, equating the above volume required, we get

$$v_3 - v_2 = 4.488 \times 10^{-2}\ m^3$$
$$\Rightarrow 1.66\ v_2 - v_2 = 4.488 \times 10^{-2}\ m^3$$

Solving the above relation, we get

$$v_2 = 0.068\ m^3$$

and $\quad v_3 = 1.66 \times 0.068 = 0.1129\ m^3$

Now

$$p_1 v_1 = p_2 v_2$$
$$\Rightarrow v_1 = p_2 v_2 / p_1$$

Substituting known values, we get

$$v_1 = (210 \times l0 \times l0^4 \times 0.068)/(85 \times 10 \times 10^4) = 0 = 168\ m^2$$

While charging the accumulator, the oil pumped is

$$v = v_1 - v_2 = 0.168 - 0.068 = 0.1\ m^3$$

Time taken for charging the accumulator (given) = 5 min = 300 s.
Therefore, the flow rate is

$$Q = \frac{v}{t} = \frac{0.1}{300} = 3.33 \times 10^{-4}\ m^3/s$$

Charged pressure (given) p= 210 $\times 10^7$ N/m^2
Power required

$$P = Q \times p = 3.33 \times 10^{-4} \times 210 \times 10^7 = 7 kW$$

Power required without accumulator = 56.77 kW
Power required with accumulator = 7 kW
The circuit to do the crushing can be seen in Fig. 1.30.

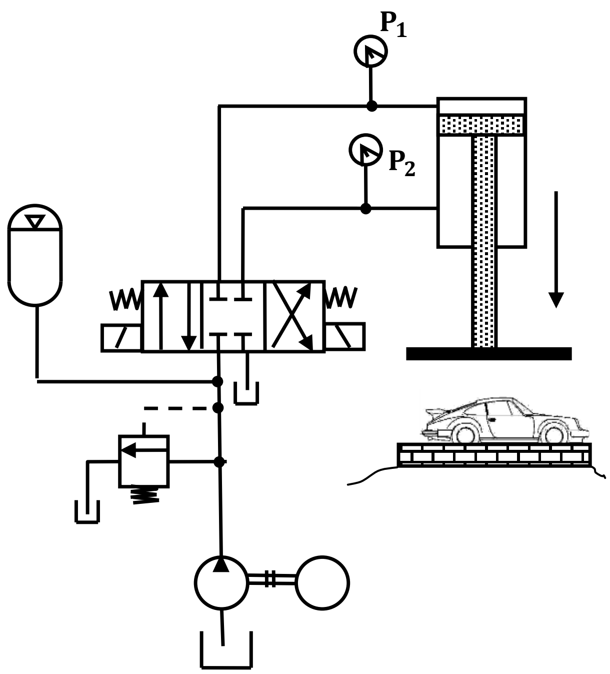


Figure 1.30

Example 1.15

A pump delivers oil at a rate of 18.2 gallons/min into the blank end of a cylinder of diameter 75 mm (Fig. 1.31). The piston contains a 25 mm diameter cushion plunger that is 25 mm; therefore, the piston decelerates over a distance of 25 mm at the end of the suction stroke. The cylinder drives a load of 7500 N that slides on a horizontal bed with coefficient of friction = 0.1. The pump relief valve pressure setting is 5.5 N/mm^2. What is the pressure acting due to cushioning deceleration?

Solution: Pump flow rate is

$$Q = 18.2 \text{ gallons/min(GPM)} \qquad = \frac{18.2}{3.785\times10^{-3}\times 60} = 80.15\,\text{m}^3/\text{s}$$

Note: 1 gallon = 3.785 L,1 L = 1000 cc = 10^{-3} m^3

Piston diameter = 75 mm

Plunger diameter = 25 mm

Plunger length = 25 mm

Load acting = 7500 N

Friction coefficient (μ) = 0.1

Relief pressure p_1= 5.5 N/mm^2

This can be visualized in Fig. 1.31.

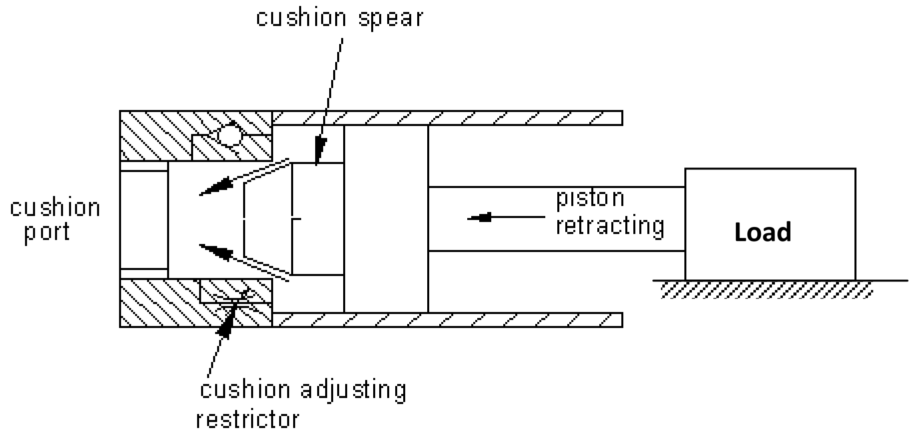


Figure 1.31

Velocity of extension until cushioning=v_1
Final velocity at the end of extension stroke=v_2 = zero (if smoothly it stops)
The velocity of extension until cushioning is given by

$$v_1 = \frac{Q}{A}$$

Area of the piston blank is

$$A= \left(\frac{\pi}{4}\right)\left(D^2\right) = \left(\frac{\pi}{4}\right)\left(\frac{75}{100}\right)^2$$

So

$$v_1 = \frac{80.15\,\text{m}^3/\text{s}}{\left(\frac{\pi}{4}\right)\left(\frac{75}{100}\right)^2} = 0.2598\,\text{m/s}$$

The final velocity v_2= 0. Writing the equation of motion

$$v_1^2 = 2 \times a \times s$$

the deceleration due to the cushioning effect can be given as

$$a = \frac{v_1^2}{2 \times s} = \frac{0.2598^2}{2 \times 25 \times 10^{-3}} = 1.349\ \frac{\text{m}}{\text{s}^2}$$

Now equating the forces on the piston we get

Blank end force−Rod end force−Frictional force + Inertial force = 0 (1.8)

We have
Blank end force=$p_1 \times (\pi/4)D^2$
Rod force=$p_2 \times (\pi/4)(D^2 - d^2)$
Frictional force $=\mu \times w$
Inertial force= Mass × Acceleration
Substituting all the values in Eq. (1.8), we get

$$[5.5 \times 10^6 \times (\pi/4)(0.075)^2] - [p_2 \times (\pi/4)\{(0.075)^2 - (0.025)^2\}] - (0.12 \times 7500) - (7500/9.81) \times 1.349 = 0$$

This gives

$$p_2 = 6216183.35\ \text{N/m}^2 = 6.22\ \text{N/mm}^2 = 6.22\ \text{MPa}$$

This is the pressure required during deceleration due to cushion effect.

Example 1.16
A double-acting cylinder is used in a regenerative circuit as shown in Fig. 1.32. The relief valves is set at 7.5 N/mm^2, the piston area is 150 cm^2, the rod area is 40 cm^2 and the flow is 20 gallons/min. Find the cylinder speed and load-carrying capacities for various DCV.

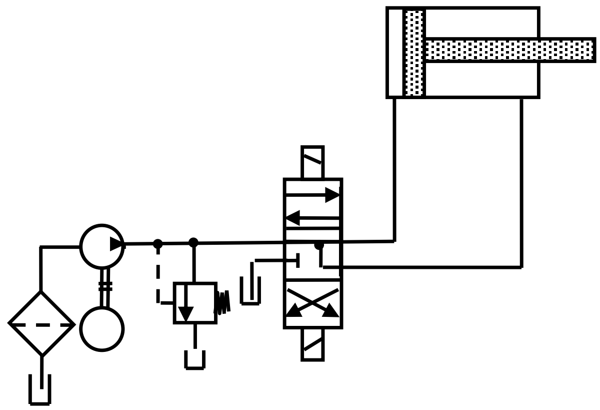


Figure 1.32

Solution: Given data
Pump flow Q_p= 20 gallons/min= [(20 ×3.785 ×10^{-3})/60] m^3/s
Piston area A_p =150 ×10^{-4} m^2 = 150 ×10^2 mm^2
Rod area A_r= 40 ×10^{-4} m^2 = 40 × 10^2 mm^2
Relief pressure p = 7.5 N/mm^2

Now
Center position of the DCV = Center position of the valve −Regenerative forward stroke
Velocity = Pump flow/area of the rod

$$\text{Velocity in the forward stroke} = \frac{Q_p}{A_r}$$

$$= \frac{20 \times 3.785 \times 10^{-5}}{60 \times 40 \times 10^{-4}}$$

$$= 0.315 \text{ m/s}$$

(Recollect from the regenerative circuit that because pressure acts on both sides, the load carrying capacity is less.) The load-carrying capacity, that is, the force that can be applied is

$$F_f = p \times [A_p - (A_p - A_r)]$$
$$= p \times A_r$$
$$= 7.5 \times 40 \times 10^2$$
$$= 30000 \text{ N}$$
$$= 30\text{kN}$$

We shall consider the left envelop of the DCV.In this position, the cylinder extension is similar to the regular double-acting cylinder where the pump flow is divertedtothe blank side (without regeneration).

$$\text{Velocity in the forward stroke} = \frac{Q_p}{A_p}$$

$$= \frac{20 \times 3.785 \times 10^{3}}{60 \times 150 \times 10^{4}}$$

$$= 0.08411 \text{ m/s}$$

The force that can be carried in this position is

$$F_{f(normal)} = p \times (A_p)$$
$$= 7.5 \times 150 \times 10^2$$
$$= 112500 \text{ N}$$
$$= 112.5 \text{ kN}$$

Now we consider the right envelop of the DCV, when the cylinder retracts as a regular double-acting cylinder.

$$\text{Velocity during the return stroke} = \frac{Q_p}{A_p - A_r}$$
$$= \frac{20 \times 3.785 \times 10^{-3}}{60 \times (150 \times 10^4 - 40 \times 10^{-4})}$$
$$= 0.1147 \text{ m/s}$$

The force that can be applied would be less than the forward stroke because the rod reduces the actual piston area to do work:

$$F_{r(normal)} = p \times (A_p - A_r)$$
$$= 7.5 \times (150 \times 10^2 - 40 \times 10^2)$$
$$= 82500 \text{ N}$$
$$= 82.5 \text{ kN}$$

Example 1.17

Two double-acting cylinders are to be synchronized connecting them in series as shown in Fig. 1.33.The load acting on each cylinder is 4000 N. Cylinder 1 has the piston diameter 50 mm and rod diameter 20 mm. If the cylinder extends 200 turns in 0.05 s, find the following:

(a) The diameter of the second cylinder.
(b) The pressure requirement of the pump.
(c) The capacity of the pump assuming the efficiency of the pump to be 85% and overall efficiency of the system as 90%.

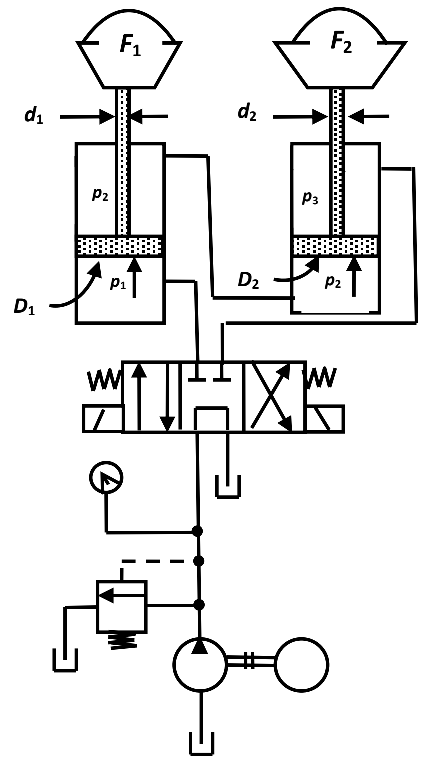


Figure 1.33

Solution:
Force on both cylinders$p$ = 4000 N
Diameter of the piston of cylinder lD_1 = 50 mm
Diameter of the rod of cylinder 1d_1 = 20 mm

(a) Diameter of second cylinder

Pressure on the rod side of cylinder l = Pressure of fluid leaving cylinder 1
= Force/(Area of piston 1 −Area of rod 1)
$=p/(A_{p1} - A_{r1})$
$= 4000 \times 4/[\pi \times (D_l^2 - d_1^2)]$
$= 4000 \times 4/ [\pi \times (50^2 - 20^2)]$
$= 2.425\ \text{N/mm}^2$
Also, for serial synchronizing circuits,
Pressure of fluid leaving cylinder 1 = Pressure of fluid coming into cylinder 2
Pressure of fluid coming into cylinder 2 = 2.425 N/mm^2
= Force/area of the piston of cylinder 2
So

$$1.425 = 4000 \times 41\ (\pi \times D_2^2)$$
$$\Rightarrow D_2 = 45.825\ \text{mm}$$

(b) The pressure requirement of the pump

Let us now determine the pressure required at the piston side of cylinder 1:

$$F_1 = 4000 \text{ N} = [(\pi/4) \times D_1^2 \times p_1] - [(\pi/4) \times (D_1^2 - d_1^2) \times p_2]$$

Substituting the known values, we get

$$4000 \text{ N} = [(\pi/4) \times 50^2 \times p_1] - [(\pi/4) \times (50^2 - 20^2) \times 2.425]$$

We get $p_1 = 4.074 \text{N/mm}^2$.

Length of the cylinder $L = 200$ mm
Time taken to extend 200mm $t = 0.05$ s
Efficiency of the pump = 0.9
Efficiency of the system =0.85
The forward stroke velocity is

$$v_f = L/t = 200/0.05 = 4000 \text{ mm/s}$$

The flow rate required from the pump is

$$\begin{aligned} Q &= \text{Area of the piston of cylinder } 1 \times v_f \\ &= (\pi/4) \times D_1^2 \times v_f \\ &= (\pi/4) \times 50^2 \times 4000 \\ &= 7.85 \times 10^{-3} \text{m}^3/\text{s} \end{aligned}$$

(c) Capacity of the pump

Let us now calculate the pump capacity in kW:
Input capacity in kW

$$\frac{p_1 Q}{\eta_{\text{pump}} \eta_o} = \frac{4.074 \times 10^6 \times 7.85 \times 10^{-3}}{0.9 \times 0.85} = 41.805 \text{kW}$$

Output capacity of the pump = $Q \times p$

$$\begin{aligned} &= 7.85 \times 10^{-3} \times 4.074 \times 10^6 \\ &= 31.981 \text{ kW} \end{aligned}$$

Example 1.18

A high–low circuit uses a low-pressure pump of 1.4 N/mm^2 and a high-pressure pump of 12.6 N/mm^2. The press contains eight cylinders and the total load of the press is 5600 kN. The length of cylinder is 200 mm whereas the punching stroke is only for 6 mm. The time for lowering is 0.05 s and the time for pressure build-up and pressing is 0.03 s. Determine the following:

(a) The piston diameter of cylinder.
(b) Pump flow rates.
(c) Total motor capacity.
(d) If a single pump of 12.6 N/mm^2 is used, find the kW capacity required.

Solution:
(a) Piston diameter of cylinder

Number of cylinders = 8

Total force = 5600 kN
High-flow pump pressure= 1.4 N/mm^2 = p_{high} = 1.4 ×10^6 N/m
Low-flow pump pressure= 12.6 N/mm^2 = p_{low} = 12.6 ×10^6 N/m
Stroke length, L= 6 mm
Force acting in one cylinder= Total load/Number of cylinders
= 5600000 / 8
= 700000 N
Pressure during punching = 126 N/mm^2
Piston diameter of the cylinder,

$$D^2 = 700000 \times 4\ (\pi \times 12.6)$$

$$\Rightarrow D = 266 \text{ mm} = 0.266 \text{ m}$$

(b) Pump flow rate

The flow requirements of the low-pressure pump:
Stroke velocity = Stroke length/time taken
= 6/0.05 = 120 mm/s = 0.120m/s
Flow from the low-pressure pump= Area × Stroke velocity
= $(\pi/4) \times 0.266^2 \times 0.120$
= 0.00667 m^3/s
Power output is

$$P_{low} = Q_{low} \times p_{low}$$
$$= 1.4 \times 10^6 \times 0.00667 \text{ W} = 9.338 \text{ kW}$$

When the pressure of oil is increased in a compartment, the volume changes. Usually,

Volume change = $\frac{1}{2}$% of initial volume for 70 bar

Compressed volume = $\frac{1}{2}$% of $(\pi/4) \times D^2 \times L$
$= 0.005 \times (\pi/4) \times (0.266)^2 \times 0.006$
$= 1.667 \times 10^{-6}$ m^3
This volume is compressed because of the difference in the pressure level of the high- and low-pressure pumps:
Difference in pressure = 12.6 − 1.4 N/mm^2= 11.2 ×10^6 N/m^2
Therefore, the volume change for 11.2 ×10^6 N/m^2 is

$$11.2 \times 10^6 \times 1.667 \times 10^{-6} / 70 \times 10^6 = 2.6558 \times 10^{-6} \text{ m}^3$$

Therefore, the flow from the high-pressure pump

$$Q_{high} = 2.6558 \times 10^6 / 0.03 \text{ (volume/time taken)}$$
$$= 8.85 \times 10^{-5} \text{ m}^3/\text{s}$$

(c) Motor capacity

Hence, the power output is

$$P_{high} = Q_{high} \times p_{high}$$
$$= (8.85 \times 10^{-5}) \times (12.6 \times 10^6)$$
$$= 1.115 \text{ kW}$$

The total kilowatt capacity required is

$$P_{high} + P_{low} = 9.338 + 1.115 = 10.453 \text{ kW}$$

(d) If a single pump is used, capacity requirement

Instead, if a single pump with high-pressure capacity is required, then

Flow required ($Q_{combined}$) = Flow from the low-pressure pump + flow from the high-pressure pump

$$= 0.00667 + 8.85 \times 10^{-5} m^3/s$$
$$= 0.00676 \text{ m}^3/s$$

The power capacity required for the above pump is

$$\text{Power capacity} = Q_{combined} \times p_{high}$$
$$= 0.00676 \times 12.6 \times 10^6$$
$$= 85157.1 \text{ W}$$
$$= 85.157 \text{ kW}$$

From the above calculations, it is evident that when a high–low circuit is used instead of a single high-pressure pump, the power requirement is reduced considerably.

Example 1.19

A high–low circuit with an unloading valve is employed for press application. This operation requires a flow rate of 200 LPM for high-speed opening and closing of the dies at the maximum pressure of 30 bar. The work stroke needs a maximum pressure of 400 bar, but a flow rate between 12 and 20 LPM is acceptable. Determine the suitable delivery for each pump.

Solution: A high–low circuit uses a high-pressure, low-volume pump and a low-pressure, high-volume pump.During closing or opening, both the pumps supply fluids. During work stroke, the high-pressure pump alone supplies fluid. Power requirement is the same for both processes.

Theoretical power required to open or close the dies is

$$P = \frac{200 \times 10^{-3} \times 30 \times 10^5}{60} = 10000 \text{ W}$$

To utilize this power for the pressing process, the flow required is calculated. We know that

$$\text{Power} = \text{Flow} \times \text{Pressure}$$
$$10000 = Q \times 400 \times 10^5$$

Solving we get

$$Q = 2.5 \times 10^{-4} \text{ m}^3/s$$
$$= 2.5 \times 10^{-4} \times 60 \text{ m}^3/\text{min}$$
$$= 15 \times 10^{-3} \text{ m}^3/\text{min}$$
$$= 15 \text{ LPM}$$

This is acceptable. Therefore, the delivery of the high-pressure, low-volume pump = 15 LPM.

The delivery of the low-pressure, high-volume pump = 200 – 15 =185 LPM.

An equivalent single fixed displacement pump having a flow rate of 200 LPM and working at a pressure of 400 bar requires a theoretical input power of 133.3kW.

Example 1.20

A press with the platen weighing 5 kN is used for forming. The force required for pressing is 100 kN and a counterbalance valve is used to counteract the weight of the tools. The cylinder with a piston diameter 80 mm and a rod diameter 60 mm is used. Calculate the pressure to achieve 100 kN pressing force.

Solution:

Weight of the tools = 5 kN = 5 $\times 10^3$ N

Full bore area= $\frac{\pi}{4} \times 0.08^2$ = 0.005 m^2

Annulus area,

$$A_p - A_r = \frac{\pi}{4} \times (0.08^2 - 0.06^2) = 0.0022 \text{ m}^2$$

Pressure required at rod side to balance tools is

$$\frac{5 \times 10^3}{0.0022} \times 10^{-5} = 22.7 \text{ bar}$$

Suggested counterbalance valve setting =1.3 $\times$ 22.7 = 29.5 bar

Pressure at full bore side to overcome counterbalance = $29.5 \times \frac{0.0022}{0.005}$ = 13 bar

Pressure to achieve 100 kN pressing force at full bore side = $\frac{100 \times 10^3 \times 10^{-5}}{0.005}$ + 13 = 213 bar

Example 1.21

In a meter-in circuit, a cylinder with 100 mm bore diameter and 70 mm diameter is used to exert a forward thrust of 100 kN, with a velocity of 0.5 m/min. Neglect the pressure drop through the piping valves. If the pump flow is 20 LPM, find the following:

(a) Pressure required at the pump on extend.
(b) Flow through the flow-control valve.
(c) Relief-valve setting.
(d) Flow out of the pressure-relief valve.
(e) System efficiency during extend.

Solution:Both Q and q are being used. Kindly check for correctness

(a)Force needed during extend,F = 100 kN. Therefore, thepressure required at pump on extend p' is

$$p' = \frac{100 \times 10^3}{\frac{\pi}{4} \times 0.1^2} = 127 \text{ bar}$$

(b) Velocity during extend,v = 0.5 m/min.

Flow through the flow control valve is

$$q = v \times A_p$$

$$= 0.5 \times \frac{\pi}{4} \times 0.1^2$$

$$= 3.9 \times 10^{-3} \text{ m}^3\text{/min}$$

$$= 3.9 \text{ LPM}$$

(c) Relief-valve setting, p = 127 + 10 % (127) = 140 bar
(d) Flow out of the pressure-relief valve is

$$q' = Q - q = 20 - 3.9 = 16.99 \text{ LPM}$$

(e) System efficiency is

$$\frac{p'q}{pQ} \times 100 = \frac{127 \times 3.9}{140 \times 20} \times 100 = 17.6\%$$

Example 1.22
A hydraulic intensifier is meant to enhance the fluid pressure from 50 to 200 bar. Its small-cylinder capacity is 23 L and has a stroke of 1.5 m. Find the diameter of the larger cylinder to be used for this intensifier.

Solution:
The capacity of small cylinder Q = Area of small cylinder × Stroke
Diameter of the small cylinder is

$$d = \left(\frac{Q \times 4}{\pi \times s}\right)^{1/2} = \left(\frac{10^{-3} \times 23 \times 4}{\pi \times 1.5}\right)^{1/2} = 0.140\text{m} = 140\text{mm}$$

Let p_s be the supply pressure and p_i be the intensifier pressure. Then intensification ratio is

$$\frac{A_i}{A_s} = \frac{p_s}{p_i} = \frac{D^2}{d^2} = \frac{200}{50}$$

$$\Rightarrow \frac{D}{d} = 2$$

Diameter of larger cylinder is

$$D = 2 \times d = 2 \times 140 = 280 \text{ mm}$$

Example 1.23
A punch press circuit with five stations operated by five parallel cylinders is connected to an intensifier. The cylinders are single-acting cylinders with spring return and the piston diameter of the cylinder is 140 mm. The cylinders are used for punching 10 mm diameter holes on sheet metal of 1.5 mm thickness. The ultimate shear strength of sheet material is 300 MN/m^2. The punching stroke requires 10 mm travel. If the intensification ratio is 20 and the stroke of the intensifier is 1.3 m, determine the following:

(a) Pressure of oil from the pump.
(b) Diameter of small and large cylinders of intensifier.

Solution:
(a) Pressure of oil from the pump
The force required to punch the hole is

F = Shear area × Shear strength
= π × Diameter of hole × Thickness of sheet × Shear strength
= $\pi \times 10 \times 10^{-3} \times 1.5 \times 10^{-3} \times 300 \times 10^{6}$
= 14137 N

Pressure developed at the load cylinder is

$$\frac{F}{A} = \frac{14137}{\frac{\pi}{4} \times (0.14)^2} = 9.2\,\text{bar}$$

This is the pressure developed by the intensifier. The pressure from the pump is the pressure exerted on the large cylinder of intensifier.

Pump pressure

$$p_i = p_s \times \left(\frac{A_s}{A_i}\right) = 9.2 \times \frac{1}{20} = 0.46\,\text{bar}$$

(b) Diameter of small and large diameter

Volume of oil required in the cylinders during punching strokeis

V_{oil} = 5 × Area of cylinder × Punch stroke

$$= 5 \times \frac{\pi}{4} \times 0.14^2 \times 10 \times 10^{-3} = 7.7 \times 10^{-4}\ \text{m}^3$$

This is supplied by the intensifier. Now

Area of small cylinder × Stroke of intensifier = $7.7 \times 10^{-4}\ \text{m}^3$

So

Area of small cylinder $A_s = \dfrac{7.7 \times 10^{-4}}{1.3}$

Also

Area of larger cylinder A_L = Intensification ratio ×A_s
= 20 ×5.9 ×10^{-4}
= 0.118 m^2

Diameter of the small cylinder is

$$d = \left(\frac{5.9 \times 10^{-4} \times 4}{\pi}\right)^{1/2} = 0.027\text{m}$$

Diameter of the larger cylinder

$$D = \left(\frac{0.0118 \times 4}{\pi}\right)^{1/2} = 0.122\text{m}$$

Example 1.24

A double-acting cylinder is hooked up in a regenerative circuit for drilling application. The relief valve is set at 75 bar. The piston diameter is 140 mm and the rod diameter is 100 mm. If the pump flow is 80 LPM, find the cylinder speed and load-carrying capacity for various positions of direction control valve.

Solution:

Center position of the valve: Regenerative extension stroke

$$\text{Cylinder speed} = \frac{Q_p}{A_r} = \frac{\left(\frac{80\times10^{-3}}{60}\right)}{\left(\frac{\pi}{4}\times0.1^2\right)} = 0.169\,\text{m/s}$$

Load-carrying capacity is

$$p\times A_r = 75\times10^5\times\frac{\pi}{4}\times(0.1)^2 = 58905\text{ N}$$

Left position of the valve: Extension stroke without regeneration:

$$\text{Cylinder speed} = \frac{Q_p}{A_p} = \frac{\left(\frac{80\times10^{-3}}{60}\right)}{\left(\frac{\pi}{4}\times0.14^2\right)} = 0.86\,\text{m/s}$$

Load-carrying capacity is

$$p\times A_p = 75\times10^5\times\frac{\pi}{4}\times(0.14)^2 = 115453\text{N}$$

Right position of the valve: Retraction stroke

$$\text{Cylinder speed} = \frac{Q_p}{A_p - A_r} = \frac{\left(\frac{80\times10^{-3}}{60}\right)}{\left(\frac{\pi}{4}\times0.14^2-0.1^2\right)} = 0.177\,\text{m/s}$$

Load-carrying capacity is

$$p(A_p - A_r) = 75\times10^5\times(0.14^2-0.1^2) = 56548\,\text{N}$$

Example 1.25

Two double-acting cylinders are to be synchronized by connecting them in series. The load acting on each cylinder is 4000 N. If one of the cylinders has the piston diameter 50 mm and rod diameter 28 mm, find the following:

(a) The diameter of the second cylinder.
(b) Pressure requirement of the pump.
(c) Power of the pump in kW if the cylinder velocity is 4 m/s.

Solution: The area of second cylinder is

$$A_{p2} = A_{p1} - A_{r1} = \frac{\pi}{4}(0.05^2-0.028^2) = 1.35\times10^{-3}\text{ m}^2$$

Diameter of the second cylinder is

$$D_{p2} = \sqrt{\left(\frac{A_{p2}\times4}{\pi}\right)} = \sqrt{\left(\frac{1.35\times10^{-3}\times4}{\pi}\right)} = 0.041\,\text{m}$$

The pump supplies oil to the first cylinder, so the pressure requirement of the pump is

$$p_1 = \frac{F_1 + F_2}{A_{p1}} = \frac{4000 + 4000}{\frac{\pi}{4}(0.05)^2} = 40.7\,\text{bar}$$

Cylinder velocity = 4m/s
Flow requirement of pump

$$Q = \frac{\pi}{4} \times 0.05^2 \times 4 = 7.85\times10^{-3}\ \text{m}^3/\text{s}$$

Power of pump in kW

$$\frac{p_1 \times Q}{1000} = \frac{40.7\times10^5 \times 7.85\times10^{-3}}{1000} = 32\,\text{kW}$$

Example 1.26

A pump delivers 60 L/min, the system maximum working pressure is 250 bar and the return line maximum pressure is 80 bar. Select suitable tubes.Use the data provided in Table 1.1: Cold-drawn seamless CS tubes (DIN 2391/C)(DIN is a German Standard).

Table 1.1 Cold-drawn seamless CS tubes (DIN 2391/C)

Cold-Drawn Seamless Carbon Steel Tubes for Pressures Purposes to DIN 2391/C				
Outer Diameter × Wall Thickness	**Approximate Weight**	**Maximum Working Pressures (bar)**		
		Safety Factor	**Safety Factor**	**Safety Factor**
6 × 1.5	0.166	703	586	441
6 × 1.0	0.123	428	359	269
8 × 1.5	0.240	496	414	310
8 × 1.0	0.173	310	255	193
10 × 3.5	0.561	1089	903	676
10 × 2	0.395	531	441	331
10 × 1.5	0.314	386	317	241
10 × 1.0	0.222	241	200	152
12 × 2.5	0.586	552	462	345
12 × 1.5	0.388	310	262	193
14 × 2.5	0.709	324	282	217
14 × 1.5	0.462	262	221	166
15 × 2.5	0.771	428	352	262
15 × 1.5	0.499	241	207	152
16 × 3.0	0.962	490	407	303
16 × 2.5	0.832	393	331	248

16×2.0	0.691	310	255	193
16×1.5	0.536	228	186	145
18×1.5	0.610	200	166	124
20×4.0	1.58	531	441	331
20×3.0	1.26	379	317	234
20×2.5	1.08	303	255	193
20×2.0	0.888	241	200	152
20×1.5	0.684	179	152	110
22×3.0	1.41	338	283	214
22×2.0	1.07	221	179	138
22×1.5	0.758	159	138	103
25×4.0	2.072	394	263	210
25×3	1.63	297	248	186
25×2	1.13	193	159	117
28×4	2.37	359	297	221
28×3.5	2.11	310	255	193
28×2.5	1.57	214	179	131
28×2.0	1.28	166	138	103
28×1.0	0.6666	83	69	52
30×4.0	2.56	332	276	207
30×3.5	2.00	241	200	152
30×3.0	2.367	242	161	121
38×4.0	4.07	324	269	228
38×3.5	3.35	255	214	159
38×3.0	2.59	186	159	117
40×6.0	5.03	379	317	234
40×5.0	4.32	310	255	193
42×3.0	2.885	201	133	101
48×5.0	6.21	310	255	193
65×8.0	11.24	303	255	186

Solution:
Return line: Select a velocity of 1.0 m/s. Now

Flow area = $\frac{\text{Discharge}}{\text{Velocity}} = \frac{60 \times 10^{-3}}{60} \times \frac{1}{1.0} = 0.001 \text{ m}^2$

Now

Inner diameter of tube $=\left(\dfrac{0.001\times 4}{\pi}\right)^{1/2}$ = 0.035m= 35 mm

Select a standard tube size of inner diameter 36mm, outer diameter 42 mm and wall thickness 3mm.This has a safe working pressure of 101 bar, so it is suitable for the return lines.

Delivery lines: Select a velocity of 3.5m/s. Now

$$\text{Flow area} = \left(\frac{60}{60}\times 10^{-3}\right)\times\frac{1}{3.5} = 2.857\times 10^{-4}\ \text{m}^2$$

$$\text{Inner diameter} = \left(\frac{2.857\times 10^{-4}\times 4}{\pi}\right) = 0.019\,\text{m} = 19\,\text{mm}$$

Select a standard tube size of inner diameter 19mm, outer diameter 25 mm and wall thickness 3mm. This has a safe working pressure of 297 bar and is suitable for the delivery lines.

Example 1.27

A flow control valve has a controlled flow C_v of 10.5 LPM/ $\sqrt{\text{bar}}$ and a free flow C_v of 32.4 LPM/ $\sqrt{\text{bar}}$. Determine the pressure drop across the valve in both the controlled flow and free flow directions. The system has a flow rate of 19 L/min and uses standard hydraulic oil of specific gravity 0.9.

Solution: Pressure drop in the controlled flow direction

$$\Delta p = \frac{Q^2}{C_V{}^2}\times \text{SG} = \frac{19^2}{10.5^2}\times 0.9 = 2.94\,\text{bar}$$

Pressure drop in the free flow direction

$$\Delta p = \frac{Q^2}{C_V{}^2}\times \text{SG} = \frac{19^2}{32.4^2}\times 0.9 = 0.309\,\text{bar}$$

Example 1.28

A cylinder has to exert a forward thrust of 150 kN and a reversible thrust of 15 kN (Fig. 1.34). The retract speed should be approximately 5 m/min utilizing full pump flow. Assume that the maximum pump pressure is 150 bar. Pressure drops over the following components and their associated pipe work are as follows:

Filter = 3 bar
Direction control valve (each flow path) = 2 bar
Determine the following:

(a) Suitable cylinder (assume 2:1 ratio; piston area to rod area).
(b) Pump capacity.
(c) Relief-valve setting pressure.

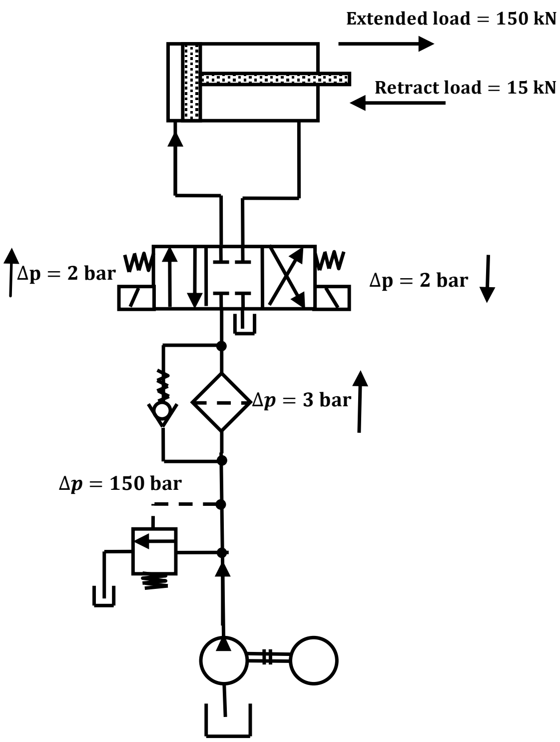


Figure 1.34

Solution:

(a) Back pressure at the annulus side of cylinder is 2 bar. This is equivalent to 1 bar at the full bore end because of the 2:1 area ratio.Therefore,
Maximum available pressure at the full bore end of cylinder = Maximum pump pressure – (Pressure drops + Back pressure)

$= 150 - 3 - 2 - 1$

$= 144$ bar

Now

$$\text{Full bore area} = \frac{\text{Load}}{\text{Pressure}} = \frac{150 \times 10^3}{144 \times 10^5} = 0.0104 \text{ m}^2$$

$$\text{Piston diameter} = \left(\frac{4}{\pi} \times 0.0104\right)^{1/2} = 0.115\text{m} = 115\text{mm}$$

Now select a cylinder with 125mm bore ×90 mm rod diameter. So

Bore area = 12.26×10^{-3} m^2

Rod area = 6.3×10^{-3} m^2

This is approximately in the 2:1 ratio.

(b) Now
Flow rate required for a retract speed of 5 m/min (full pump flow)= (Bore area – Rod area) ×Retract velocity

$= (12.26 \times 10^{-3} - 6.35 \times 10^{-3}) \times 5$

= 0.02955m^3/min
= 29.55 L/min

(c) We have

Pressure to overcome the load while extending= $\frac{150\times10^3}{12.26\times10^6}$ = 12.2 ×10^6 N/m^2= 122 bar

Pressure drop over the direction control valve P to A = 2 bar

Pressure drop over the direction control valve B to T(because of 2:1 area ratio 2 bar × $\frac{1}{2}$)= 1 bar

Pressure drop over the filter = 3 bar
Therefore,
Pressure required at the pump during the extend stroke= 122 +2+1+3 =128 bar

Pressure to overcome load during retraction= $\frac{15\times10^3}{(12.26\times10^{-3}-6.35\times10^{-3})}$ = 2.5 ×10^6 N/m^2=25 bar

Pressure drop over the direction control valve P to B = 2 bar
Pressure drop over the direction control valve A to T (because of the 2:1 area ratio, 2 bar ×2) = 4 bar
Pressure drop over the filter =3 bar
Therefore,
Pressure required at the pump during the retract stroke= 25 + 2+ 4+ 3=34 bar
Relief-valve setting = Maximum pressure + 10%=128 +(0.1 ×128)=141 bar

Example 1.29

A press cylinder having a bore diameter of 140 and a 100 mm diameter rod is to have an initial approach speed of 5 m/min and a final pressing speed of 0.5 m/min. The system pressure for a rapid approach is 40 bar and for final pressing is 350 bar. A two-pump, high–low system is to be used. Both pumps may be assumed to have the volumetric and overall efficiencies of 0.95 and 0.85, respectively. Determine the following:

(a) The flow to the cylinder for the rapid approach and final pressing.
(b) Suitable deliveries for each pump.
(c) The displacement of each pump if the drive speed is 1720 RPM.
(d) The pump motor power required during the rapid approach and final pressing.
(e) Retract speed if the pressure required for retraction is 25 bar maximum.

Solution: A high–low circuit uses a high-pressure, low-volume pump and a low-pressure, high-volume pump.

(a) The flow to the cylinder rod for the rapid approach is

Q_{rapid} = Bore diameter ×Velocity of initial approach

$$= \frac{\pi}{4}\times0.140^2\times5 = 0.077\,\frac{\text{m}^3}{\text{min}} = 77\,\text{LPM}$$

The flow to the cylinder for final pressing is

$Q_{\text{high pressure}}$ = Bore diameter ×Velocity of final pressing

$$= \frac{\pi}{4}\times 0.140^2\times 0.5 = 0.0077\,\frac{\mathrm{m}^3}{\mathrm{min}} = 7.7\,\mathrm{LPM}$$

(b) During final pressing, only a high-pressure, low-volume pump supplies fluid.
Pump delivery = 7.7 LPM
During the rapid approach, both pumps supply fluid. The high-volume pump delivery is

$$Q_{\text{rapid}} - Q_{\text{high pressure}} = 77 - 7.7 = 69.3\ \text{LPM}$$

(c) The displacement of a low-volume pump

$$D_{\text{P-high pressure}} = \frac{Q_{\text{high pressure}}}{N_{\mathrm{p}}\times\eta_{\mathrm{V}}} = \frac{7.7}{1720\times 0.95} = 4.7\times 10^{-3} = 4.7\,\mathrm{mL}$$

The displacement of a high-volume (low pressure) pump

$$D_{\text{P-low pressure}} = \frac{Q_{\text{low pressure}}}{N_{\mathrm{p}}\times\eta_{\mathrm{V}}} = \frac{69.3}{1720\times 0.95} = 42.3\times 10^{-3}\,\mathrm{L} = 42.4\,\mathrm{mL}$$

(d) Pump motor power required during the rapid approach:

$$\frac{P_{\text{low pressure}}\times Q}{\eta_{\mathrm{o}}} = \frac{40\times 10^5\times 0.077}{60\times 1000\times 0.85} = 6.04\ \mathrm{kW}$$

Pump motor power required during final pressing:

$$\frac{P_{\text{high pressure}}\times Q_{\text{high pressure (low volume)}}}{\eta_{\mathrm{o}}} = \frac{350\times 10^5\times 0.0077}{60\times 1000\times 0.85} = 5.3\,\mathrm{kW}$$

(e)We have

$$\text{Retract speed} = \frac{Q}{\frac{\pi}{4}(D^2-d^2)} = \frac{4\times 0.077}{\frac{\pi}{4}(0.140^2-0.100^2)} = 10.2\,\mathrm{m/s}$$

Objective-Type Questions
Fill in the Blanks

1. In a regenerative circuit, the speed of extension is greater than that for a regular double-acting cylinder because the flow from the ______ regenerates with the pump.
2. For two cylinders to be synchronized, the piston area of cylinder 2 must be equal to ______ between the areas of piston and rod for cylinder 1.
3. Meter ______ systems are used primarily when the external load opposes the direction of motion of the hydraulic cylinder.

4. One drawback of a meter ______ system is the excessive pressure build-up in the rod end of the cylinder while it is extending.
5. Fail–safe circuits are those designed to prevent injury to operator or damage to ______.

State True or False

1. In a regenerative circuit, when the piston area equals two-and–a-half times the rod area, the extension and retraction speeds are equal.
2. The load-carrying capacity for a regenerative cylinder during extension equals pressure times the piston rod area.
3. When two cylinders are identical, the loads on the cylinder are not identical, and then extension and retraction can be synchronized.
4. When a load pulls downward due to gravity, in such a situation a meter-in system is preferred.
5. A machine intended for high-volume production uses rapid traverse and feed circuits.

Review Questions

1. List three important considerations to be taken into account while designing a hydraulic circuit.
2. What are the advantages of a regenerative circuit?
3. Explain the regenerative circuit for a drilling machine.
4. With the help of a neat sketch, explain the pump-unloading circuit.
5. With the help of a circuit diagram, explain a double-pump hydraulic system (Hi–Lo circuit).
6. Explain the application of a counterbalance valve.
7. Explain the application of a pilot check valve for locking a double-acting cylinder.
8. Explain the speed control circuit for a hydraulic motor.
9. What are the conditions for the two cylinders to be synchronized?
10. What is a fail-safe circuit?

Answers
Fill in the Blanks

1.Rod end
2.Difference
3.In
4.Out
5.Equipment/machine

State True or False

1.False
2.True
3.False
4.False
5.True

Lecture 27

SERVO VALVES

Learning Objectives

Upon completion of this chapter, the student should be able to:

- Define servo valve.
- Compare servo valves with proportional valves.
- Appreciate the history of servo valves.
- Describe the working of a servo torque motor.
- Describe the working of single-stage spool-type servo valves.
- Describe the working of jet-type servo valves.
- Analyze the valve performance.
- Define dead band and hysteresis.
- Analyze mathematically the simple servo systems.

1.1 Introduction

Servo valves were developed to facilitate the adjustment of fluid flow based on changes in the load motion. Simply put, it is a programmable orifice.In machine motion control, servo systems involve continuous monitoring, feedback and correction. Also they are used to improve efficiency, accuracy and repeatability. The most common applications of servo valves are in aerospace vehicles, particularly in primary flight controls. In aircraft, control surfaces such as ailerons, elevators and rudders are positioned by servo units. In space vehicles, control during launch is provided by movable thrust nozzles that are positioned by servo units. Even the drill bits for angle drilling of oil wells are servo controlled.

A servomechanism is defined as an automatic device for controlling a large amount of power by means of a very small amount of power and automatically correcting the performance of a mechanism. The automatic and continuous correction requires return of information from the mechanism– feedback, in other words. Therefore, a servo valve is operated without feedback and it is not a true servomechanism. In Chapter 17 we have studied about proportional valves and Table 1.1 gives comparison between servo valve and electrohydraulic proportional control valves (EHPV).

Table 1.1 Comparison of servo valves and electrohydraulic proportional valves

Feature	**Servo Valve**	**EHPV**
Electrical operator	Torque motor	Proportional solenoid
Manufacturing precision	Extremely high	Moderately high
Feedback circuitry	Main system as well as valve	Valve (depending on type), main system (seldom)
Cost (compared with a solenoid valve)	Very expensive	Moderately expensive

1.2 History of Electrohydraulic Servomechanisms

The earliest recognized servomechanism is the water clock invented around 250 BC(Fig. 1.1) by the Alexandrian inventor Ktesbios. In this device, time was recorded by the level of water in a graduated vessel. Water flows into the vessel at a controlled and constant rate from a water reservoir above it. The control of the flow rate from the reservoir involves a mechanism.Velocity flow rate from the outlet of a reservoir is determined by the equation

$$v = \sqrt{2gh}$$

where V is the velocity, g is the acceleration due to gravity and h is the height of water above the outlet.The volume flow rate through the outlet depends upon the size of the outlet and the fluid velocity. Thus,

$$Q = \text{Velocity} \times \text{Area}$$

From these equations, we can see that as the water level in Ktesbios's reservoir goes down, the flow rate from the reservoir decreases. Ktesbios's solution to the problem was to use a second reservoir mounted above the first. He used a float to modulate an orifice through which water was fed into the primary reservoir. This kept the water level constant, resulting in hours of constant length.

Numerous servomechanisms were invented during Industrial Revolution in the mid-1700s and afterward. Many were associated with steam boiler technology where they were used to control water level, water and steam flow, steam pressure and the speed and position of steam-operated mechanisms.

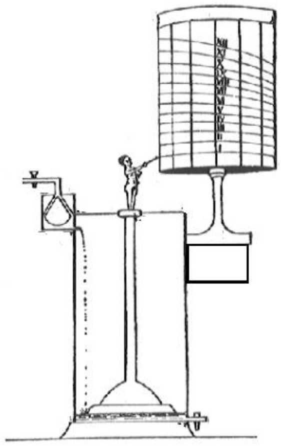


Figure 1.1 Water clock by Ktesbios.

1.3 Electrohydraulic Servomechanism Concepts

Figure 1.2 represents a typical fluid power system that uses a proportional valve to control the speed of hydraulic motor. The EHPV is set to provide the necessary flow to drive the motor at the required speed. As long as there are no disturbances, the speed remains constant. If there is any change in the operating parameters such as load, fluid temperature, viscosity and wear then the motor speed is likely to change. There is nothing designed in the system to detect the change and present the information to the valve controller that can automatically correct the change and return the speed to the required level. Speed correction is the responsibility of the operator who must make the required control adjustments.

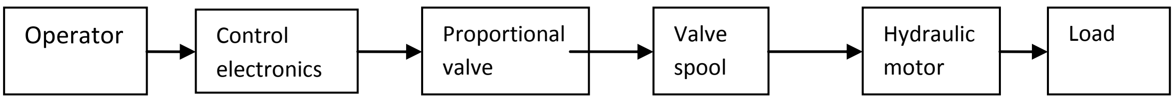


Figure 1.2 Proportional valve block diagram.

Although this type of circuit is perfectly satisfactory for a very large number of applications, some require automatic and continuous corrections. These circuits require servomechanisms. These mechanisms can be simply referred to as servo valves.

Figure 1.3 shows the circuit that has the same purpose as that of Fig.1.2, but in this circuit the operator has been relieved from the responsibility of speed corrections. Instead, a tachometer generator has been installed that senses the load speed. This information is automatically and continuously fed back to the control electronics (usually a printed circuit board) where it is compared with the operator command signal input.

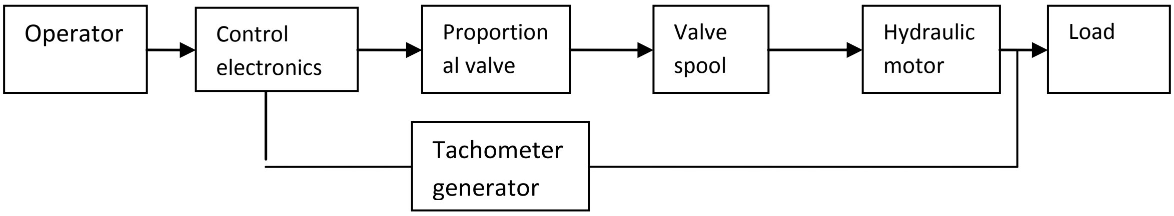


Figure 1.3 Servo control provides automatic and continuous corrections for any changes in motor rpm.

If any difference is found between these signals, the electronic circuitry automatically generates a correction signal proportional to the difference. This signal repositions the valve to correct the flow rate as required. This “sense and correct” function is continuous, so any and every change in load speed is automatically corrected. The system required to perform this function includes three major segments: the servo valve, the command electronics and the feedback transducer.

1.4 Servo Valves

Servo valves can be used in virtually any aspect of fluid power system operations, including the following:

1. Positioning of cylinders and rotary actuators.
2. Speed of cylinders and motors.
3. Cylinder force and motor torque.
4. Acceleration and deceleration.
5. System pressure.
6. Flow rate.

The most common applications are for cylinder positioning and motor speed control. The valves for these functions incorporate both direction and flow control in sliding spool arrangement that is positioned by a torque motor.

1.4.1 Torque Motor

A torque motor is illustrated in Fig. 1.4. It is a simple electromagnetic device consisting of one or two permanent magnets, two pole pieces, a ferromagnetic armature and two coils. The permanent magnet polarizes the upper and lower pole pieces, so that they present equal and opposite magnetic fields. Torque motors are very low-power devices operated on low-voltage DC power.

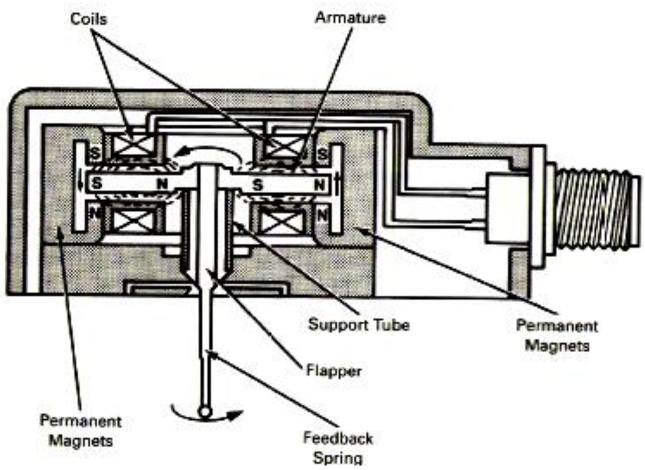


Figure 1.4 Servo valve torque motor.

The armature is mounted at its mid-point so that it is free to rotate through a very limited are either clockwise or counter-clockwise. The ends of the armature are extended into the gaps between the pole pieces. The magnetic field holds the armature in a neutral position. The two coils surround the arms of the armature to form two small electromagnets. When a current is passed through the coils, a magnetic field is generated. The polarity of the field depends on the direction of the current flow. In Fig. 1.5, the current flow causes the left end to become the South Pole and right end to become the North Pole, resulting in counter-clockwise rotation of the armature.

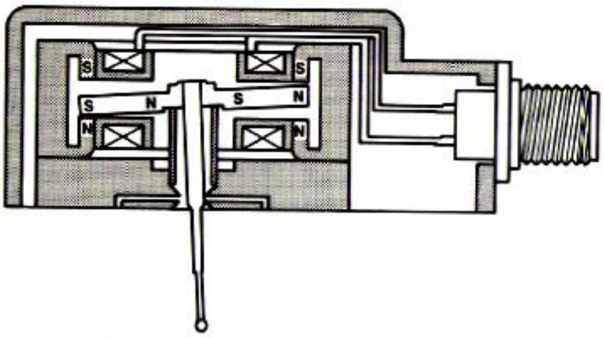
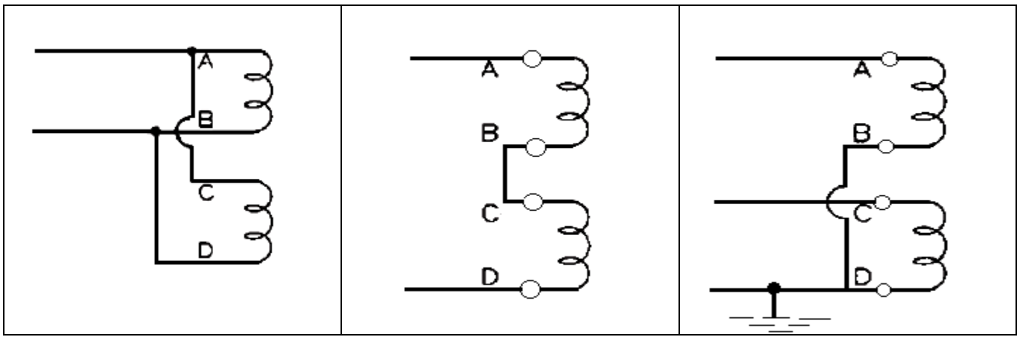


Figure 1.5 Servo torque motor operation.

The two coils of a torque motor may be connected in three different configurations: parallel, series and the so-called push–pull arrangement. These options are illustrated in Fig. 1.6. The push–pull arrangement is the most common. In this arrangement, both the leads B and D are connected to ground through the control circuit amplifier. Leads A and C are connected to separate output terminals on the command amplifier. When the voltage input to both coils equals, the armature is centered. Increasing the voltage input to one coil while simultaneously reducing the input to the other coil by the same amount causes the armature to rotate. The voltage can be varied from zero to its maximum value for each coil, but the polarity is never reversed. This means that the position of the armature is determined by the differential torque. When the voltage is the same for both coils, the torque is equal and the armature is centered. Any change in the voltage to either coil results in the rotation of the armature.

This push–pull method of connecting the coil is preferred for at least three reasons:

1.Any change in the current as a result of voltage fluctuations, temperature changes or other causes is canceled by the equal and opposite effects on the coil.
2. There is more stability in armature positioning because of the opposing torque.
3. The power consumption is lower than in the other two configurations for the other two circuits.



(a) (b) (c)

Figure 1.6 Torque motors can be connected in several combinations. (a) Parallel (aiding) , (b) series (aiding), (c) push–pull (opposing).

The input to this arrangement is expressed as a differential current ΔI, where ΔI is the difference between the coil currents. The control power is calculated from the equation

$$P = (\Delta I)^2 R$$

where P represents the control power, ΔI represents the differential current $= I_{AD} - I_{BC}$ and R represents the resistance of one coil.In parallel connection, the direction of rotation depends on the polarity of the input signal. The coils in the parallel circuitassist each other rather than oppose (as in a push–pull circuit). That is, both of them attempt to move either clockwise or counter-clockwise. Reversing the polarity of coils reverses the polarity of the rotation. The control power is then found from

$$P = (I_p)^2 \left(\frac{R}{2}\right)$$

where I_p is the total current through the circuit and R is the resistance of the coil.In a series circuit, the coils assist, rather than oppose, the armature rotation. As with the parallel circuit, a polarity change is required to change the direction of rotation. The control power of a series circuit is given by

$$P = (I_S)^2 2R$$

where I_S is the current in the series circuit and R is the resistance of each coil.Notice that the series and parallel circuits have the same maximum power requirement, which is exactly twice that of the push–pull circuit.

It is interesting to note that these two low-power torque motors can control a two- or three-stage valve that may be flowing 300–400 LPM or more at 140–300 bar. Taking in-between values, we see that the power output of the valve approaches 90000 W. If we define the power gain of the valve as the output power divided by the control power, we have

$$\text{Power gain} = \frac{90000 \text{ W}}{1.6 \text{ W}} = 5.625 \times 10^4$$

1.4.2 Valve Spools

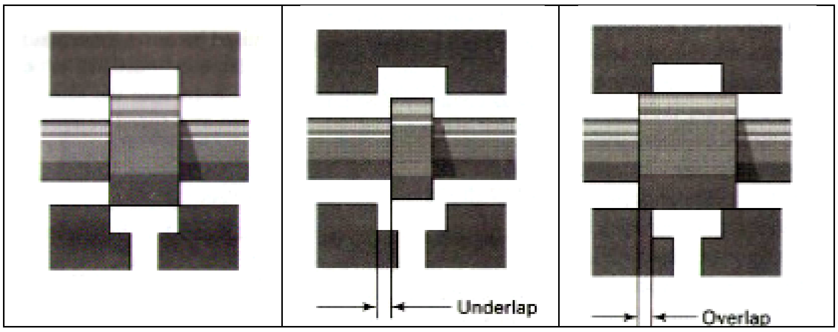
At first glance, the hardware of a servo valve looks very similar to that of any spool-type directional control valve, that is, a sliding spool that operates in the bore of the valve body to open and close the flow path between ports. The actual differences are found more in the manufacturing process and clearance specifications than in the basic design.

The servo valve spool and the bore (in which spool moves) are very high precision components. Typically, spool and bore straightness and diametrical tolerances are held to ±1 μm. The radial clearance between the spool and the bore is typically 3–5 μm. In most valves, the radial expansion that results from holding the spool in hand for a few minutes would prevent its insertion into the bore. To achieve this precision, a great deal of hand finishing is involved in the manufacturing process. The spool and the body are quite often a matched set, and the parts are not interchangeable.

Special spool surface finishes are often employed. Nitriding is often used to provide extra surface hardness and a glass-like smooth finish. This reduces the friction and improves the wear characteristics. Servo valves may be either three- or four-way. The spool may have two, three or four lands, depending on the function and on the manufacturer's preference. It has been shown that four-land spools can have slightly large clearances without incurring unacceptable leakage. This means that they have improved wear characteristics and are somewhat more tolerant of contaminants in the fluid. The two outer lands also assist in keeping the spool precisely centered.

As with most spool-type valves, circumferential grooves are machined into the spool lands. The purpose of the grooves is to reduce the side loading by equalizing the pressure around the spool and holding it centered into the bore. A spool with three grooves can have as little as 6% of the side force as found in the ungrooved spools.

Spool "lap" defines the width of the lands relative to the width of the ports in the valve bore. There are three possible lap configurations: Overlap, underlap and line-to-line. These are shown in Fig. 1.7.



(a) (b) (c)

Figure 1.7 (a) Zero overlap. (b) Underlapped. (c) Overlapped

By far, the most common condition is the line-to-line (or zero-overlap) spool. Here, the bandwidth exactly matches the port width. Thus, when the spool is centered, there is no flow. Any movement of the spool, regardless of how little, results in flow through the valve. This valve is suitable for closed-loop position, speed, and force control applications because of its precise metering characteristics about the null (neutral) position. Unfortunately, even a small amount of wear on either the land or port edge results in leakage in the null position.

Overlapped spools have lands that are 0.5–5% wider than the ports. These spools have the advantage of providing lower leakage flow in the null position than the line-to-line configuration.However, the overlap means that the precision achievable about the null position is compromised because of the relatively large dead band. For instance, when used as a position controller, a cylinder that is being extended stops at a different position when being retracted even with the same command input. An overlapped valve can be satisfactorily employed as a speed controller as long as it is operated well away from its null position.

In many servo valve control circuits, dither is used to reduce the effects of static friction (termed stiction). Dither is a very low amplitude command signal superimposed over the normal command signal that results in a continuous, very short stroke, lateral oscillation of the spool. In such systems, a slight overlap may be used to prevent unexpected leakage in the null position.

An underlapped spool has lands that are 0.5–1.5% narrower than the ports. This design is often referred to as “open center” although there really are no open-center servo valves. The underlap is far too small to be a true open center. This type of valve provides very rapid response to commands about the null position, but it has the disadvantage of having non-linear flow characteristics near null. This compromises control to some extent.

1.4.3 Valve Configurations

Servo valves may be single-stage (also called direct-acting), two-stage or three-stage, depending primarily on the flow requirements of the system.Single-stage valves may be used when the flow requirements are low (usually less than 20 LPM, depending on the valve design). These valves commonly utilize a sliding spool mechanically connected to the torque motor armature. The flow capacity is dictated by the low force available from the torque motor and the limited stroke of the spool.

1.4.4 Single-StageSpool-Type Servo Valve

Figure 1.8 shows a single-stage servo valve. The mechanical connection between the torque motor armature and the spool is a stiff wire. When there is no command input to the torque motor, the armature is in the neutral (nulled) position, which, in turn, causes the spool to be in the nulled position, and there is no flow through the valve. A clockwise deflection of the armature pushes the spool to the left, opening up flow path from P to B and A to T. A counter-clockwise deflection opens P to A and B to T.

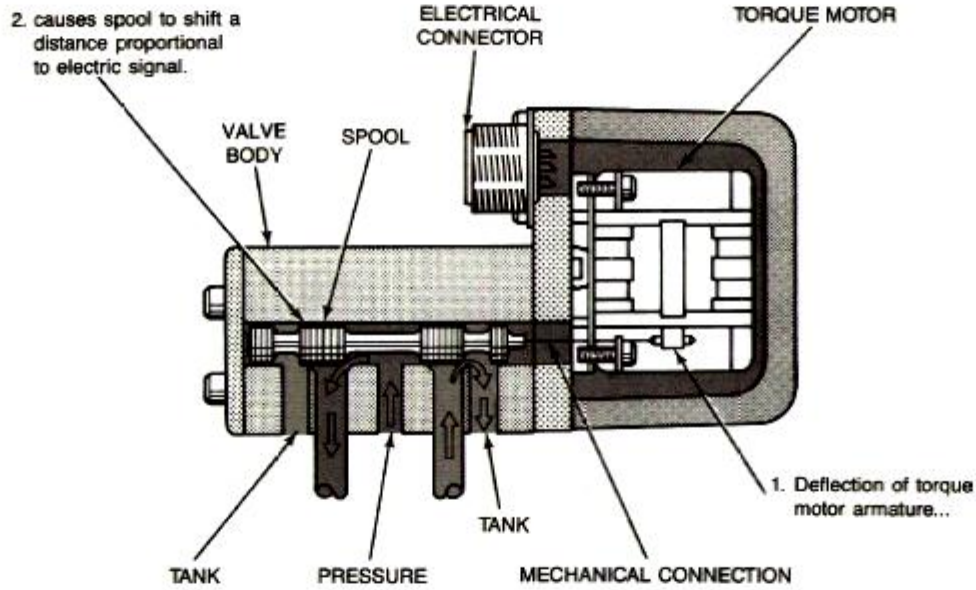


Figure 1.8 Single-stage spool-type servo valve.

For higher flow rates, two- or even three-stage valves must be used. In these valves, second and third stages are always sliding spools that are pilot operated from the previous stages. The first stage may use the sliding spool, but there are other designs also.

1.5.5Two-Stage Servo Valve

Figure 1.9 shows a two-stage servo valve, pilot operated. This valve can be used to control the direction and speed of a hydraulic motor. The pilot spool is positioned by the torque motor. The pilot stage is sleeve. This sleeve is associated with the internal feedback mechanism of the valve. There are two pressure sources: p_c is the control pressure for piloting the main spool and p_s is the supply pressure for operating the system.

In neutral position, as shown in Fig. 1.9, the middle spool land blocks off the pilot part to the left (large) end of the main (second-stage) spool. The pressure on the large end of the spool is half the control pressure. The pressure at the small end of the spool is always equal to the control pressure. In order to be balanced, the large end must have twice the area of the small end. The result is that the main spool is static. Because no input has been made to the pilot spool, the main spool is static. Because no input has been made to the pilot spool, the main spool is in neutral position, and so there is no flow to the hydraulic motor.

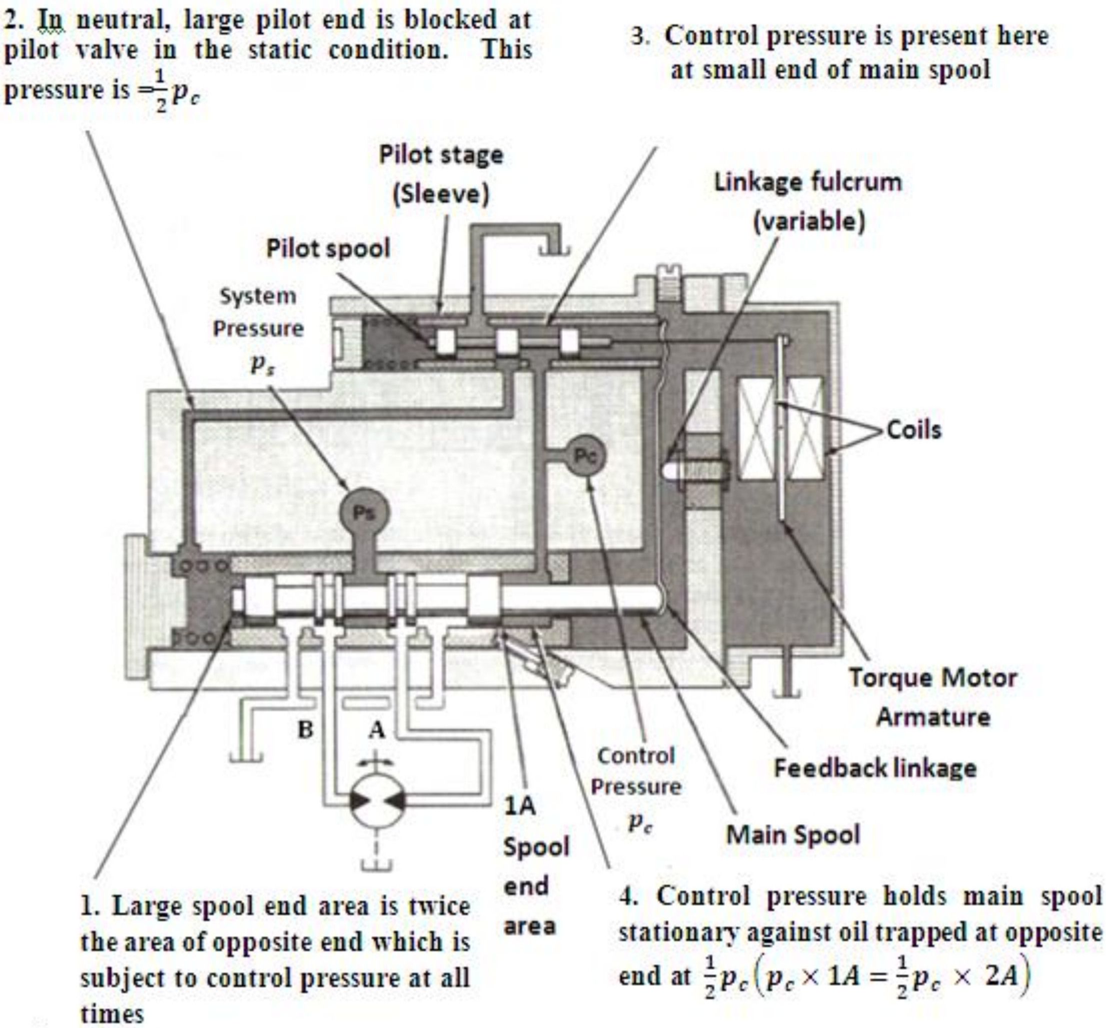


Figure 1.9 Two-stage spool-type servo valve.

A counter-clockwise movement of torque motor armature pushes the pilot spool to the left. This opens up the port (through the pilot-stage sleeve) to the large end of the main spool. The pressure in that end increases, causing a force imbalance that shifts the main spool to the right and opens flow paths P to A and B to T.

This movement of the main spool also initiates the internal feedback action. As the main spool moves to the right, it pushes the feedback linkage. This linkage transmits the main spool movement to the pilot-stage sleeve, causing the sleeve to slide to the left. When the sleeve port moves to the point where it is blocked by the pilot spool land, flow to the large end of the main spool stops. Consequently, the main spool stops. The exact stopping point is predetermined and is controlled by the deflection of the torque motor. Because the opening created by the movement of the main spool constitutes a control orifice, both the direction and the speed of rotation of the hydraulic motor are set.

De-energizing the torque motor returns the pilot spool to its neutral position, which opens up a flow path through the pilot-stage sleeve and ports the large end of the main spool to the tank. The resulting force imbalance shifts the spool to the left. This allows the pilot-stage spring to push the spool back to the right. When the sleeve port realigns with the pilot spool land, all pilot flow stops, the main spool stops (in its neutral position), and the hydraulic motor stops. A similar but opposite sequence takes place when the torque motor armature is deflected clockwise.

1.5.6Double-Flapper Nozzle Pilot Stage

Figure 1.10 shows a double-flapper nozzle pilot stage. The nozzle is a physical part of the torque motor armature and extends into a fluid cavity inside the valve. Control pressure for piloting the main spool enters the pilot section through the ports on opposite sides of the flapper. The pressure is then channeled to nozzles located in the fluid cavity into which the flapper extends. These channels are also connected to the pilot chambers at the ends of the main spool.

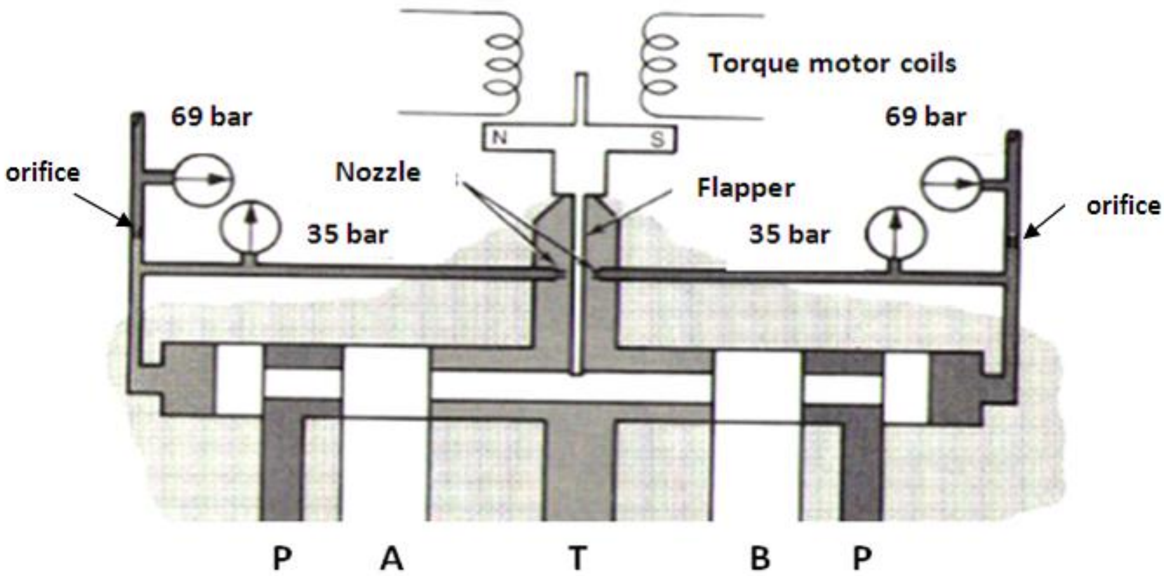


Figure 1.10 Two-stage spool-type servo valve.

When the torque motor is not energized, the armature is centered, which positions the flapper exactly in the middle of the cavity. Because the distances between the nozzle lands and the flapper are very small, this, in effect, forms small orifices that restrict the flow and generate a pilot pressure. With the flapper exactly centered, both orifices are of the same size, resulting in the same pressure in both pilot chambers. This causes a force balance on the main spool and holds it in its neutral position.

If the armature is deflected counter-clockwise, the flapper moves toward the right-hand nozzle. This movement reduces the orifice associated with that nozzle while increasing the orifice for the opposite nozzle. This increases the pressure in the right-hand pilot chamber and decreases the opposite pressure. The result is a force imbalance that shifts the main spool to the left and opens the related flow paths.

The internal feedback mechanism in the flapper-nozzle valve is flexible metal rod (usually termed a spring) that is attached to the end of the flapper and inserted into a ball joint in the main spool. As the main spool shifts to the left, the feedback spring exerts a force on the flapper (and on the armature) that tends to return it to its neutral position. The force is proportional to the distance moved by the spool. So as the spool shifts to the left, the restoring force in the flapper increases. When the force is sufficient to center the flapper, the control orifices are again equal, as are the pressures in the pilot chambers, and so the main spool stops. It remains in that position until a subsequent command signal causes the armature to move again.

1.5.7Jet Pipe Servo Valve

Figure 1.11 shows another pilot stage – jet pipe stage. In this device, the jet nozzle is attached to and moves with the torque motor armature. The pilot fluid is directed by the jet nozzle into two receiver ports that connect to the pilot chambers at the ends of the main spool. In neutral position, the fluid is evenly divided, which results in equal pressures in the pilot chambers. As a result, the main spool remains stationary. A deflection of the torque motor directs the jet more directly into one of the receiver ports, causing the force imbalance and main spool movement. Internal feedback is provided by a feedback spring in a manner similar to that in the flapper nozzle system.

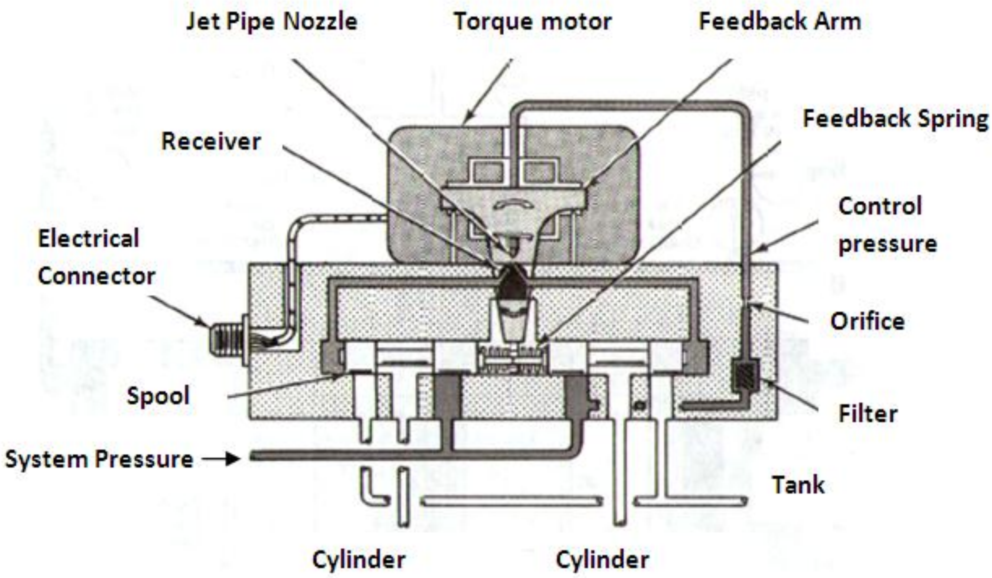


Figure 1.11 Two-stage spool-type servo valve.

A variation of the jet pipe concept is the deflector jet valves shown in Fig. 1.12. In this valve, the jet pipe concept does not change. Rather, deflectors are moved into the fluid jet, which direct it into the appropriate receiver ports.

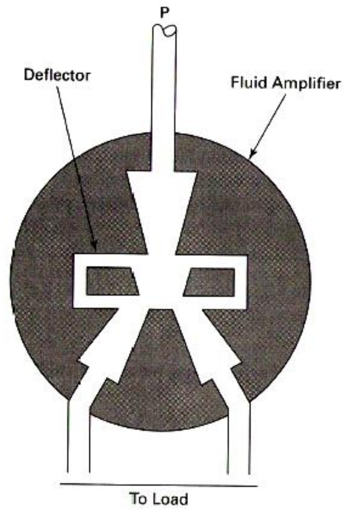


Figure 1.12 Deflector jet valve.

The standard symbols for servo valves are shown in Fig. 1.13. The symbols do not hint the number of stages or type of pilot stage used. They show only function. The directional flow control valves are always three-position, infinitely positionable units and usually have closed centers. The actuator symbols represent the summing junction of the electronic amplifier, which receives two input signals (reference and feedback) and produces a single output signal.

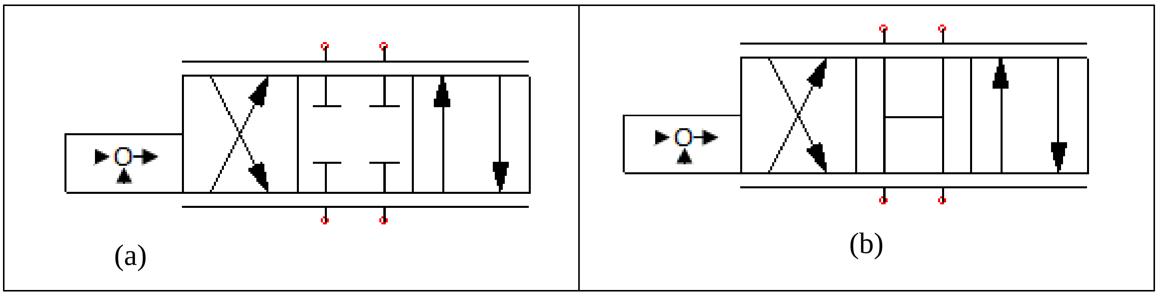


Figure 1.13 Servo symbols: (a) Line-to-line and overlapped spool; (b) underlapped spool**.**

1.5.8Pressure Flow Characteristics

Servo valves are generally considered to be “high-pressure” devices. Usually the pressure rated is 200 bar, although most are capable of operating higher than 300 bar. Servo valves are typically flow rated at 60–70 bar differential; that is, the flow rate stated for the valve is the flow that occurs at 60–70 bar pressure drop across a fully opened valve. There is a very deliberate logic in the choice of 60–70 bar. The majority of servo systems use a working pressure of 200 bar. It can be mathematically shown that the maximum transmission of power from the pump to the cylinder or motor occurs when the pressure drop through the valve is one-third of the system working pressure. This parameter is related to the basic orifice equation, which describes the flow rate through orifice. Thus, the valves are flow rated at the maximum power transfer pressure drop based on the popular 200 bar. It has also been determined that valve control is optimized at the 65 bar pressure drop.

If the pressure drop across the valve is different from the optimum, the flow rate changes proportionately. The non-optimum flow rate can be calculated using the equation

$$q = q_r \sqrt{\frac{\Delta p}{\Delta p_r}} \qquad (1.1)$$

where q is the adjusted flow rate, $q_r$is the rated flow, Δp is the actual pressure drop and Δp_r is the rated pressure drop (usually 65 bar).

1.5.9Valve Performance

The performance of a valve can be described by numerous parameters. They are generally divided into two categories: dynamic response and static response. The dynamic response characteristics are the same as those for proportional valves, frequency response and amplitude ratio. Figure 1.14 is a Bode diagram showing these characteristics(frequency response and amplitude ratio) for a specific valve.

During the normal operation of a valve, the valve is likely to experience the same current input (set point) frequently. Sometimes, this specific current input is approached from a lower current setting. At other times, it is approached from higher settings. All valves have the characteristic that this current set point results in different spool positions, depending on whether it is approached from a lower or higher current.

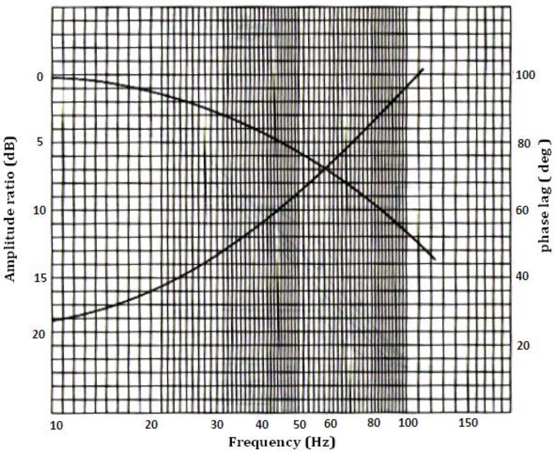


Figure 1.14 Valve performance.

This characteristic is termed hysteresis. Typical hysteresis curves for servo valves and proportional control valves are shown in Fig. 1.15. Hysteresis is expressed as the percent difference in the rated current required to give the same output when approached from higher and lower inputs. For servo valves, it is typically 1–2%. To overcome the problem of hysteresis, some controllers are designed so that the set point is always approached from the lower side. This requires a deliberate undershoot when approaching from the high side.

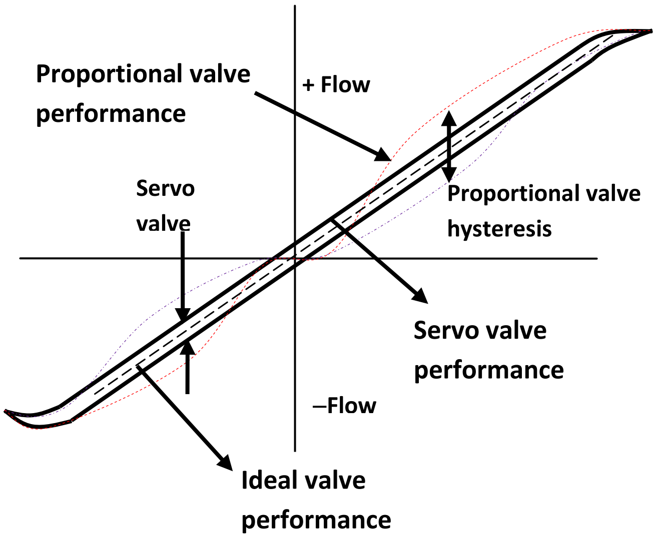


Figure 1.15 Hysteresis for a servo valve and a typical proportional valve.

A second important valve characteristic is the valve dead band. Dead band occurs only at the null position, as shown in Fig. 1.16. It is defined as the current required to move the spool from the exact centered position to the position where the first flow output is seen. It is usually expressed in milliamps or percent-rated current. Dead band is the result of the spool inertia, overlap, static friction and any other forces that might impede the initial motion.

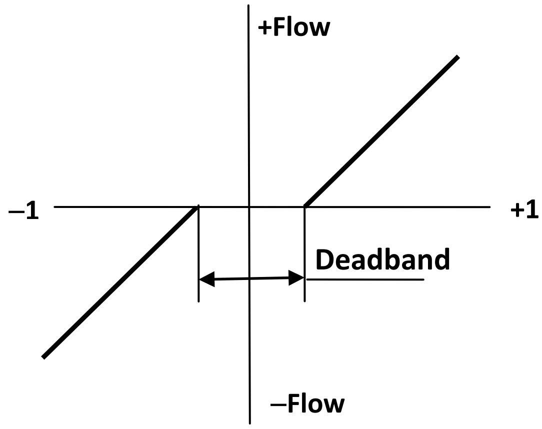


Figure 1.16 Dead band of a valve.

A similar phenomenon is threshold. Threshold current is the smallest input current required to overcome spool inertia and other impending forces to cause the spool to move. The primary difference between threshold and dead band is that threshold occurs throughout the spool stroke, whereas dead band occurs only at the null position. Threshold contributes to the dead band rating.

Information concerning hysteresis, dead band and other valve performance characteristics is available from the valve manufacturers. These characteristics can be significant in evaluating the suitability of a valve for a specific application.Threshold is the current that must be applied before a response is detected. Good-quality two-stage valves have a threshold less than 0.5% of rated current.

1.4.10Gain and Feedback

One of the most important concepts of servo system is its ability to constantly monitor its output and automatically make corrections to ensure that the output remains at the commanded level. This is accomplished through the use of some type of feedback from a transducer that monitors the output parameter. Gain is defined as the ratio of output by input. Two gains are defined for servo valves: flow gain and pressure gain. Flow gain is the ratio of flow to input current. Flow gain is determined by measuring control flow versus input current. Flow gain is the slope of the graph of control flow versus input current. Pressure gain is defined as the ratio of pressure and input current. Pressure transducers are mounted on the output ports and pressure difference is measured as a function of input current. The range of input current to produce a load pressure change from −40% to +40% of supply pressure is determined. Pressure gain is difficult to measure.

1.4.10.1 The Control Ratio Equation

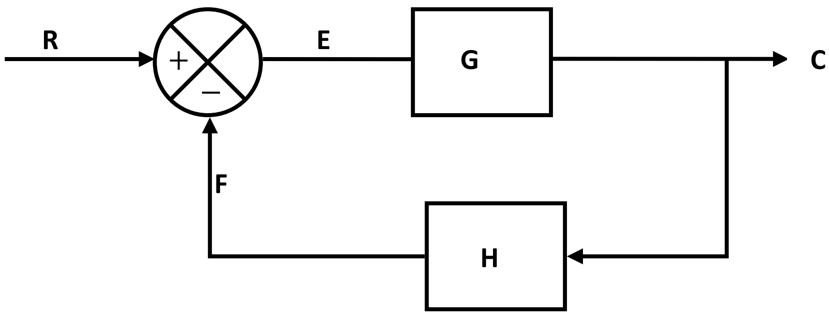


Figure 1.17 A basic feedback – closed loop.

One of the most important aspects of system analysis is an understanding of system gain.Figure 1.17 is a block diagram of a generic system with feedback. The command input signal is given a designator R for reference. The gain of the loop leg is designated G and termed the forward loop gain. The gain of the feedback loop is denoted by H and the controlled output is shown as C.
As the system operates, a feedback signal F is continuously generated. Its value is based on the value of C and the feedback gain. Thus,

$$F = CH$$

This feedback signal is fed to the summing junction (in the op-amp), where it is compared with the reference signal. The result is an error signal E whose value is

$$E = R - F = R - CH \qquad (1.2)$$

This results in a change in the controlled variable C, which then becomes

$$C = EG \qquad (1.3)$$

and the process repeats itself.Equating Eqs. (1.2) and (1.3), we get

$$C = (R - CH)G = RG - CGH$$

so that

$$RG = C + CGH = C(1 + GH)$$

and, eventually,

$$\frac{C}{R} = \frac{G}{1 + GH} \qquad (1.4)$$

This equation is known as the control ratio, and closed-loop gain or the closed-loop transfer function of the system. The right-hand side of Eq. (1.4) defines the system gain and is commonly referred to as the closed-loop gain of the system.

Example 1.1

A torque motor is connected in a push–pull circuit. Each coil has a resistance of 20 Ω and is rated at 200 mA.Find

(a) The voltage of each coil when the armature is centered.

(b) The maximum value of Δ*l*.

(c) The maximum control power for the torque motor.

Solution

(a) The maximum voltage for the coil is

$$E = IR = 200 \text{ mA} \times 20\ \Omega = 4 \text{ V}$$

The armature is centered when

$$E = \frac{E_{\max}}{2} = 2\,\text{Volts}$$

(b) The differential current is

$$I = I_{AD} - I_{BC}$$

The maximum value will occur when the maximum voltage is applied to one coil, so that zero voltage is applied to the other. In this case,

$$\Delta I = I_{AD} - I_{BC} = \frac{E_{\max}}{R} - 0 = \frac{4\,\text{V}}{20\,\Omega} = 200\,\text{mA}$$

(c) The maximum control power is then

$$P = (\Delta I^2)R = (200 \text{ mA})^2(20\ \Omega) = 0.8 \text{ W}$$

Example 1.2

A torque motor is connected in a parallel circuit. Each coil has a resistance of 20 Ω and is rated at 200 mA**.**Find

(a)The voltage of each coil when the armature is centered.

(b) The maximum value of ΔI.
(c) The maximum control power for the torque motor

Solution: (a)The voltage to each coil will remain the same (4 V).
(b) The current through the circuit will increase because of lower resistance. For a parallel circuit made up of two equal resistors, the equivalence resistance is $R/2$; in this case, 10 Ω.The value of I is the total current, which we find from

$$I_P = \frac{E}{R} = \frac{4\,\mathrm{V}}{10\Omega} = 0.4\,\mathrm{A} = 400\,\mathrm{mA}$$

(c) Control power is given by

$$P = (I_P)^2(R/2) = (400\ \mathrm{mA})^2(20\ \Omega/2) = 1.6\ \mathrm{W}$$

Example 1.3
A torque motor is connected in a series circuit. Each coil has a resistance of 20 Ω and is rated at 200 mA. Find
(a)The voltage of each coil when the armature is centered.
(b) The maximum value of ΔI.
(c) The maximum control power for the torque motor

Solution: (a) A torque motor is connected in a series circuit. Therefore, total resistance is $2R$ or 40 Ω. The maximum current will be 200 mA.
(b) A torque motor is connected in a series circuit thereforethe maximum voltage is

$$\text{Voltage } (E) = IR = (200\ \mathrm{mA})(40\ \Omega) = 8\ \mathrm{V}$$

(c)The control power is

$$P = (I_s)^2\,(2R) = (200\ \mathrm{mA})^2(2)(20\ \Omega) = 1.6\ \mathrm{W}$$

Example 1.4
A servo valve is flow rated at 56 LPM at 65 bar differential (Fig. 1.5). It is to be operated in a 130 bar system. What will its adjusted flow rate be at the optimum power transfer Δp?

Solution: For optimum power transfer, the pressure drop across the valve should be

$$\frac{130}{3} = 43.33\ \text{bar}$$

From Eq. (1.1), we have

$$q = q_r\sqrt{\frac{\Delta p}{\Delta p_r}} = 56\sqrt{\frac{43.3}{65}} = 45\ \text{LPM}$$

example>
Example 1.5
Figure 1.18 shows a servo valve which is rated at 65 bar pressure drop across the valve in a 195 bar system.
(a) Find the pressure drop when motor is fully loaded.
(b) Find the pressure drop when motor is not fully loaded.

(c) Find the pressure drop when motor is overloaded. Will there be a flow?
(d) Derive an expression for fluid temperature rise across the valve.

Note that the term “6 bar pressure drop” imply that the drop across one flow path through the valve.

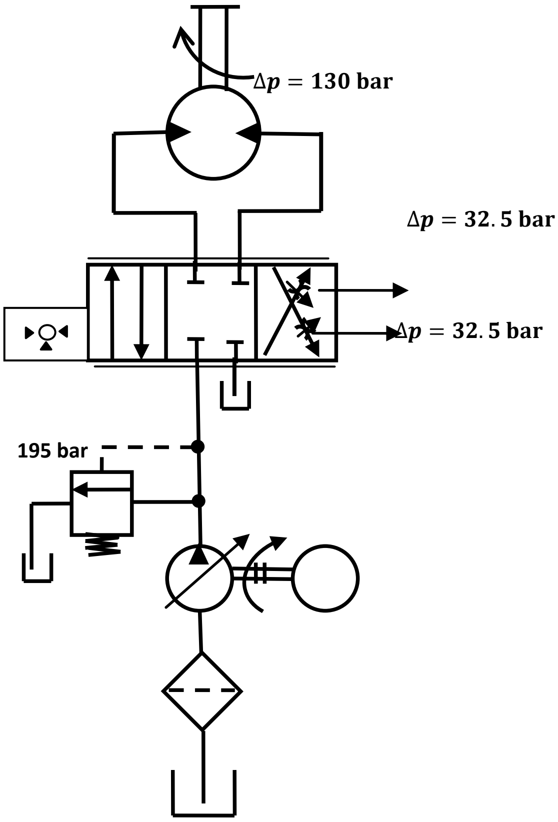


Figure 1.18

Solution:

(a) When motor is at full load: From Fig. 1.18, with the pump compensator set at 195 bar, the pressure is reduced to 195 – 32.5 = 162.5 bar across the P to A path through the valve. If we assume the motor to be operating at full load, 130 bar is dropped across the motor. The final 32.5bar is then dropped through the B to T side of the valve.

(b) When motor is not at full load: If the motor is not fully loaded, then the system pressure drops below 195 bar. In this case, the 65 bar drop across the valve still occurs, but the power transfer diminishes. (c) When motor is overloaded: If the motor is overloaded so that more than 130 bar is required to rotate it, then the 65 bar drop across the valve is not possible. The result is a lower flow rate [Eq.(1.1) applies], so the motor will turn more slowly. The extreme of this condition is a stalled motor. In this case, the full 195 bar will be dropped across the motor. Obviously, there is no flow in this case.
(d) Heat generation and fluid temperature rise: Because the servo valve represents a major pressure drop, it will also generate a considerable amount of heat. Let

HGR = Heat generation rate = $W \times C_p \times \Delta T$

P = Power loss across the valve = $\Delta p \times Q$

The fluid temperature rise across the valve can be found using the equation

$$\Delta T = \frac{\mathrm{HGR}}{C_p W} \qquad (1.5)$$

where ΔT = temperature rise ($^\circ$C), C_p = specific heat of the oil = 1.8 kJ/kg K, W (kg/s) = weight flow rate = γQ, and γ = fluid-specific weight.

Example 1.5

For the servo valve of Example 1.5, determine the power loss, heat generation rate and the temperature rise across the valve. The fluid has a specific weight of 8620 N/mm^3.

Solution: The power loss is

$$P = \Delta p \times q = 43.3\times10^5 \times (45/1000) \times (1/60)$$
$$= 3.25 \text{ kW}$$

The resultant heat generation rate is found from Eq.(1.5). The increase in the fluid temperature is

$$\Delta T = \text{HGR} / C_p W$$

The weight flow rate is

$$W = \gamma Q$$

$$\text{Oil flow rate in kg/s} = \gamma Q \text{ (oil flow rate in m}^3\text{/s)}$$
$$= 895 \times 45 \times \frac{10^{-3}}{60}$$
$$= 0.67125 \text{ kg / s}$$

Therefore,

$$\Delta T = \text{HGR} / C_p W$$
$$= \frac{4.97}{1.8 \times 0.889} = 2.689\,°\text{C}$$

This is the temperature rise experienced by every drop of oil that flows through the valve. The rise occurs in the time required for the fluid to flow through the portion of the valve where the pressure drop occurs.

Although the weight flow rate appears in the equation, the temperature rise is actually independent of flow rate. So the temperature rise for any fluid is a function of pressure drop only.

Example 1.7

In the given servo system (Fig. 1.19), suppose the forward loop contains a hydraulic motor. The forward loop gain is 300 rpm/V and the feedback loop consists of a tachogenerator that has a gain of 0.2 V/rpm. What will be the speed of the hydraulic motor for an input of 3.5 V?

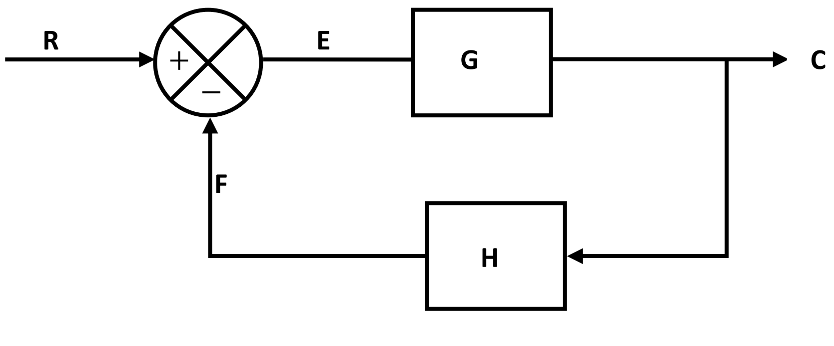


Figure 1.19

Solution: Given

Reference input, R = 3.5V

Forward loop gain, G = 300 rpm/V

Feedback loop gain, H = 0.2 V/rpm

The equation of closed-loop transfer is given by

$$\frac{C}{R} = \frac{G}{1+GH} \qquad (1.6)$$

This gives

$$C = \left(\frac{G}{1+GH}\right)R$$
$$= \left(\frac{300}{1+300\times 0.2}\right)(3.5)$$
$$= 17.2 \text{ rpm}$$

Normally, if the feedback system is not included, the 3.5 V input should give 1050 rpm. Because there is feedback, the output is 17.6 rpm.

Example 1.8
Figure 1.20 shows a closed-loop feedback system with multiple elements in the forward loop. Derive an expression for closed-loop gain.

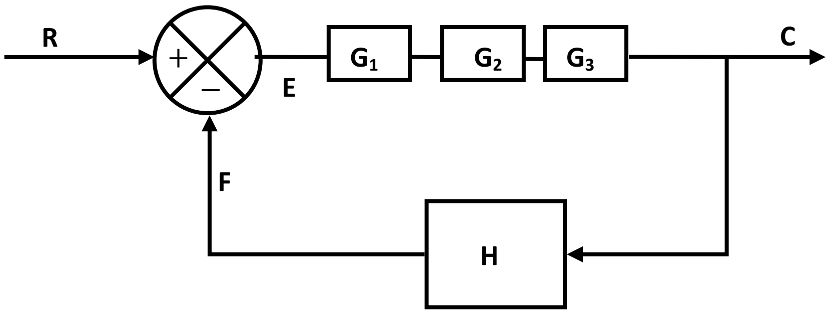


Figure 1.20

Solution:Let us have a system that has more elements in the forward loop. The gain can still be found by using the Eq.(1.6). Let us first resolve all the forward loop gains into a single value *G*. The forward loop gain *G* is the product of all the individual element gains between the summing junction and the output. Thus,

$$G = G_1 G_2 G_3 \qquad (1.7)$$

So

$$\frac{C}{R} = \frac{G}{1+GH} = \frac{G_1 G_2 G_3}{1+G_1 G_2 G_3 H} \qquad (1.8)$$

Example 1.9
Determine the speed of the hydraulic motor in the circuit shown in Fig. 1.21.

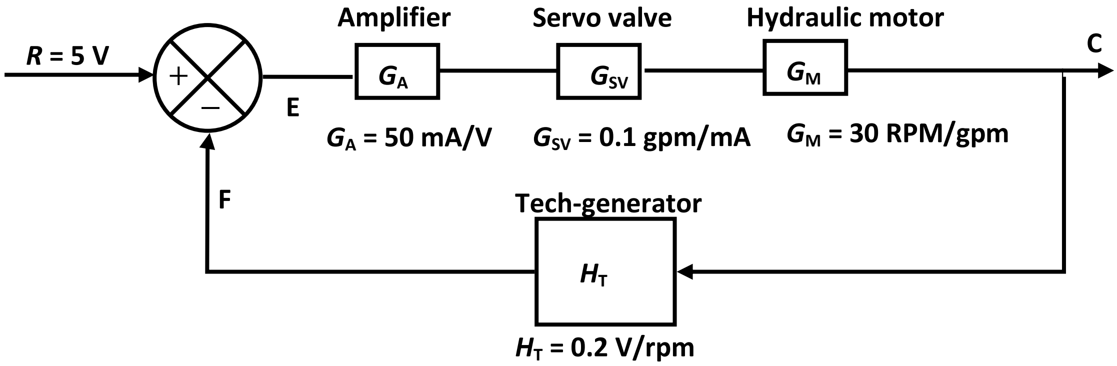


Figure 1.21

Solution: Using Eq.(1.7) we can write

$$\begin{aligned} G &= G_A G_{SV} G_M \\ &= (50\,\text{mA/V})(0.1\,\text{gpm/mA})(30\,\text{rpm/gpm}) \\ &= 150\ \text{rpm/V} \end{aligned}$$

Using the closed-loop transfer equation, we have

$$\frac{C}{R} = \frac{G}{1+GH}$$

$$\Rightarrow C = \left(\frac{G}{1+GH}\right)R$$

$$\Rightarrow C = \left(\frac{150}{1+150\times 0.2}\right)(5) = 24.2\ \text{rpm}$$

Objective-Type Questions
Fill in the Blanks

1. A servomechanism is defined as an automatic device for controlling a large amount of power by means of a very small amount of _______.
2. Servo valves are operated by _______ motors.
3. Typically, spool and bore straightness and diametrical tolerances are held to _______.
4. Overlapped spools have lands that are 0.5% to _______% wider than the ports.
5. In many servo valve control circuits, _______ is used to reduce the effects of static friction (termed stiction).
6. ______ valves may be used where the flow requirements usually less than 20 LPM_______.

State True or False

1. Servo valves are less expensive than proportional valves.
2. There is no dead zone for a servo valve.
3. A servo valve uses always zero or underlap spool.
4. The maximum operating frequency of a servo valve is 10 Hz.
5. Servo that uses feedback electronics is more accurate.

Review Questions

1. Define a servo valve.
2. How do servo valves differ from proportional control valves?
3. Explain the operation of a torque motor.
4. Define underlap, overlap and line to line in the context of servo valve spools.
5. Define dead band.
6. Define threshold.
7. Define hysteresis.
8. List and define the types of hydraulic amplifiers.
9. At what pressure drop are servo valves usually rated?
10. Define gain.
11. Define flow gain.
12. Define pressure gain.
13. What are the uses of servo valves?
14. What is a torque motor?
15. What is a spool lap?
16. What are the three types of servo valves?

Answers
Fill in the Blanks

1. Power
2. Torque
3. $\pm 1\ \mu m$
4. 5%
5. Dither
6. single stage

State True or False

1. False
2. False
3. False
4. True
5. True

Lecture 28

ACCUMULATORS

Learning Objectives
Upon completion of this chapter, the student should be able to:

- Define an accumulator.
- Explain various types of accumulators.
- Differentiate between separator and non-separator types of accumulators.
- Size an accumulator for various applications.
- Describe various applications of accumulators.
- Analyze the performance of hydraulic systems using accumulators.

1.1 Introduction

A hydraulic accumulator is a device that stores the potential energy of an incompressible fluid held under pressure by an external source against some dynamic force. This dynamic force can come from different sources. The stored potential energy in the accumulator is a quick secondary source of fluid power capable of doing useful work.
There are three basic types of accumulators:

1. Weight-loaded or gravity accumulator: Schematic diagram of weight loaded accumulator is shown in Fig. 1.1.It is a vertically mounted cylinder with a large weight. When the hydraulic fluid is pumped into it, the weight is raised. The weight applies a force on the piston that generates a pressure on the fluid side of piston. The advantage of this type of accumulator over other types is that it applies a constant pressure on the fluid throughout its range of motion. The main disadvantage is its extremely large size and heavy weight. This makes it unsuitable for mobile application.

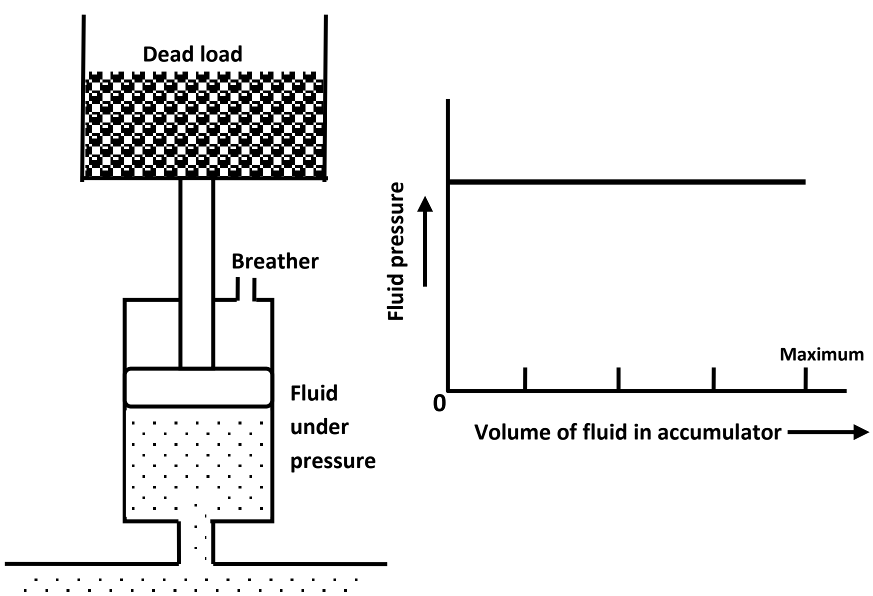


Figure 1.1 Dead weight accumulator**.**

2. **Spring-loaded accumulator:** A spring-loaded accumulator stores energy in the form of a compressed spring. A hydraulic fluid is pumped into the accumulator, causing the piston to move up and compress the

spring as shown in Fig. 1.2. The compressed spring then applies a force on the piston that exerts a pressure on the hydraulic fluid.
This type of accumulator delivers only a small volume of oil at relatively low pressure. Furthermore, the pressure exerted on the oil is not constant as in the dead-weight-type accumulator. As the springs are compressed, the accumulator pressure reaches its peak, and as the springs approach their free lengths, the accumulator pressure drops to a minimum.

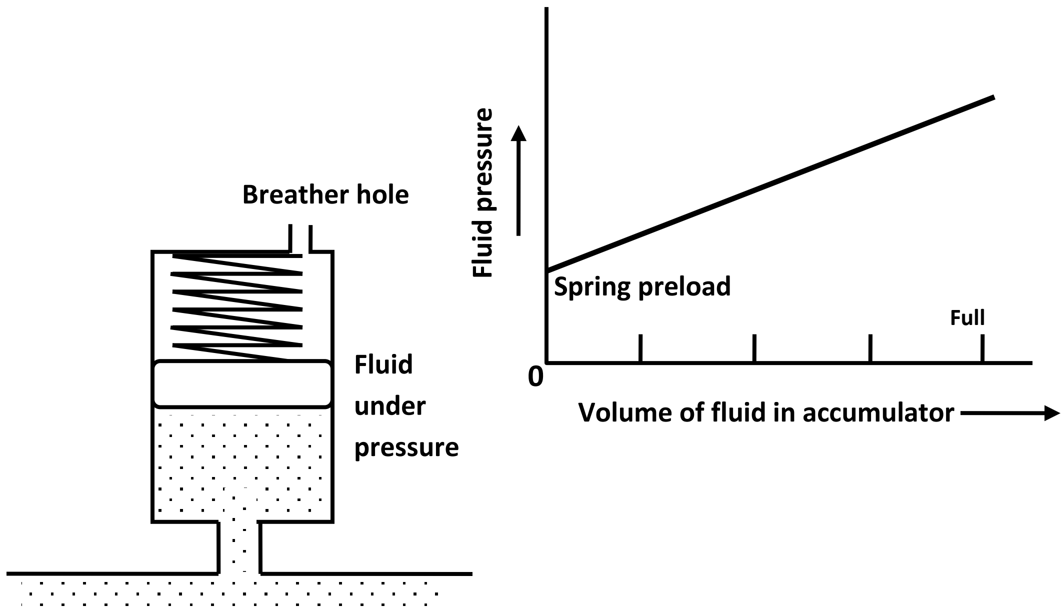


Figure 1.2 Spring-loaded accumulator.

3. Gas-loaded accumulator: A gas-loaded accumulator is popularly used in industries. Here the force is applied to the oil using compressed air. Schematic diagram of a gas loaded accumulator is shown in Fig. 1.3.A gas accumulator can be very large and is often used with water or high water-based fluids using air as a gas charge. Typical application is on water turbines to absorb pressure surges owing to valve closure and on ram pumps to smooth out the delivery flow. The exact shape of the accumulator characteristic curve depends on pressure–volume relations:

- **Isothermal (constant temperature):** This occurs when the expansion or compression of the gas is very slow. The relationship between absolute pressure p and volume V of the gas is constant:

$$pV = \text{constant} \qquad (1.1)$$

- **Isentropic (adiabatic processes):** This is where there is no flow of energy into or out of the fluid. The law that the gas obeys is given by pV^{γ} = constant, where γ is ratio of specific heat and is approximately equal to 1.4.
- **Polytropic:** This is somewhere between isothermal and isentropic. This gas change is governed by the law pV^{n} = constant, where n is somewhere between 1 and 1.4 and is known as the polytropic coefficient.

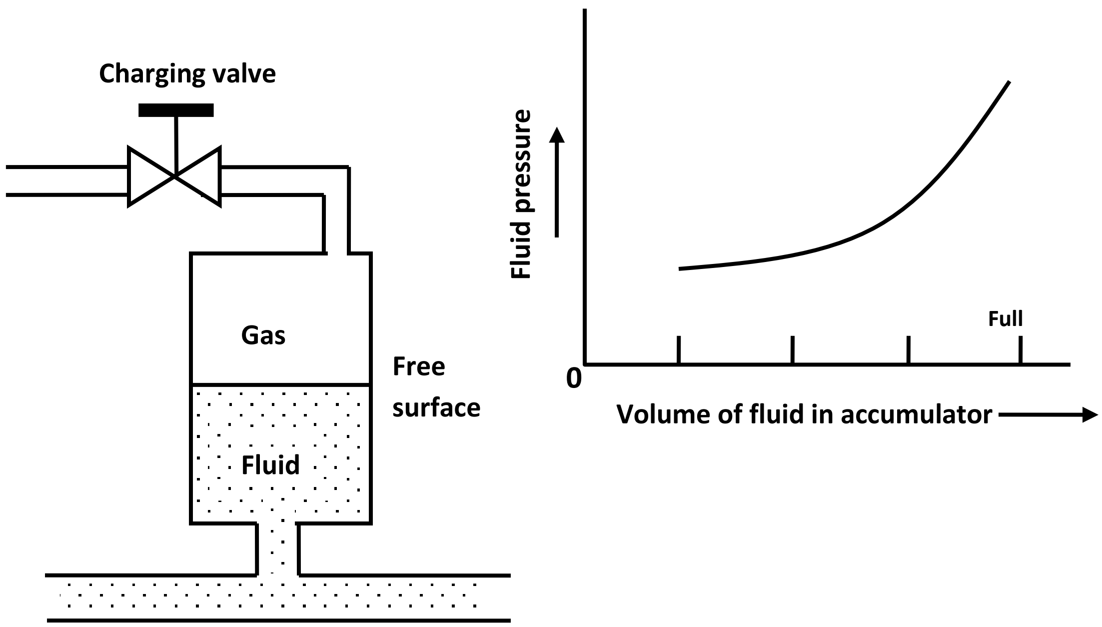


Figure 1.3 Gas-loaded accumulator.

There are two types of gas-loaded accumulators:

- **Non-separator-type accumulator:** Here the oil and gas are not separated. Hence, they are always placed vertically.
- **Separator-type accumulator:** Here the oil and gas are separated by an element. Based on the type of element used to separate the oil and gas, they are classified as follows:

(a) *Piston-type accumulator:*Schematic diagram of a piston type accumulator is shown in Fig. 1.4.It consists of a cylinder with a freely floating piston with proper seals. Its operation begins by charging the gas chamber with a gas (nitrogen) under a pre-determined pressure. This causes the free sliding piston to move down. Once the accumulator is pre-charged, a hydraulic fluid can be pumped into the hydraulic fluid port. As the fluid enters the accumulator, it causes the piston to slide up, thereby compressing the gas that increases its pressure and this pressure is then applied to the hydraulic fluid through the piston. Because the piston is free sliding, the pressure on the gas and that on the hydraulic fluid are always equal.

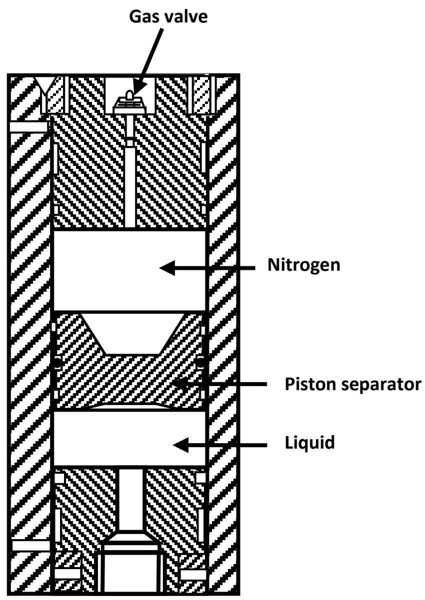


Figure 1.4 Piston-type accumulator.

(b) *Diaphragm accumulator:*In this type, the hydraulic fluid and nitrogen gas are separated by a synthetic rubber diaphragm. Schematic diagram of diaphragm accumulator is shown in Fig. 1.5. The advantage of a diaphragm accumulator over a piston accumulator is that it has no sliding surface that requires lubrication and can therefore be used with fluids having poor lubricating qualities. It is less sensitive to contamination due to lack of any close-fitting components.

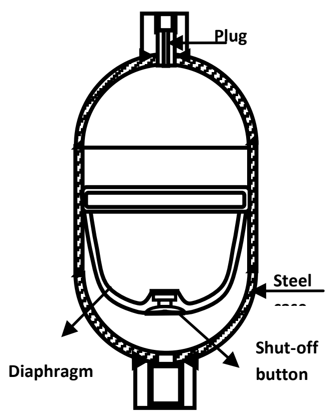


Figure 1.5 Diaphragm-type accumulator.

(c) *Bladder accumulator:* It functions in the same way as the other two accumulators. Schematic diagram of bladder accumulator is shown in Fig. 1.6. Here the gas and the hydraulic fluid are separated by a synthetic rubber bladder. The bladder is filled with nitrogen until the designed pre-charge pressure is achieved. The hydraulic fluid is then pumped into the accumulator, thereby compressing the gas and increasing the pressure in the accumulator. The port cover is a small piece of metal that protects the bladder from damage as it expands and contacts the fluid port.

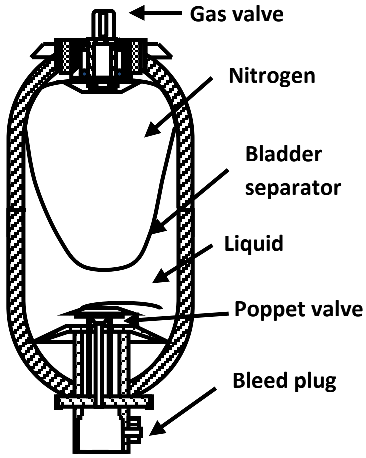


Figure 1.6 Bladder-type accumulator.

In an accumulator, at any point of time, we either compress a pre-charged gas or allow it to expand. This compression or expansion brings about a status change in the gas, which is governed by the perfect gas equation:

$$pV = mRT \qquad (1.2)$$

where $p$is the absolute pressure in bar, V is the gas volume in m^3, m is the mass in kg and R is the universal gas constant. (The most common gas used in industry is nitrogen.) For the particular gas and the accumulator, the value of mR is constant and the gas equation is written as

$$\frac{pV}{T} = \text{constant}$$

or

$$\frac{p_0V_0}{T_0} = \frac{p_1V_1}{T_1} \qquad (1.3)$$

When the change takes place over a long period of time, the temperature of the gas remains constant and such a change is called isothermal, resulting in the equation

$$p_0V_0 = p_1V_1 = p_2V_2 \qquad (1.4)$$

When the change occurs instantaneously, there is no time for heat transfer from the work to the environment.Such a change is called isentropic or reversible adiabatic and is given by

$$p_0V_0^n = p_1V_1^n = p_2V_2^n \qquad (1.5)$$

All changes between isothermal and isentropic are called polytropic. The pressure–volume diagram shown in Fig. 1.7 helps us to understand how the volume variation as a function of pressure depends on the value of the polytropic exponent n that for nitrogen is contained within the limits $1 \le n \le 1.4$. The value of n is taken to be equal to 1 if the compression and expansion process takes place under the isothermal process. For adiabatic conditions, the value of n is taken equal to 1.4.

Isothermal conditions can be considered to exist if the accumulator is used as a volume compensator, leakage compensator and pressure compensator or as a lubrication compensator. In all other cases, such as energy accumulation, pulsation damping, emergency power source, dynamic pressure compensator, shock absorber, hydraulic spring, etc., expansion and compression process may be considered to take place under "adiabatic" conditions. Generally, the adiabatic condition is considered to exist if the compression or expansion period is less than 3 min.

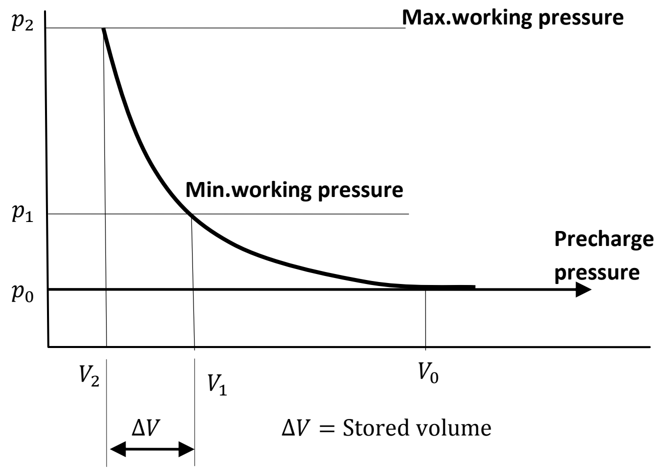


Figure 1.7 Pressure–volume diagram.

1.2 Accumulator Selection

After ascertaining the type of accumulator that is appropriate for the purpose envisaged, what remains is determining the volume of the form of "high pressure fluid." Accumulators are manufactured to a variety of pressure ratings and the one chosen should be rated for a pressure more than the maximum system working pressure p_2.

However, the values of the following basic parameters should be established before proceeding further: Working pressures p_1 and p_2. The value of p_2 is found from the ratio $p_2 / p_0 \leq 4$. The maximum gas pre-charge pressure is found from the relationship $p_0 \leq 0.9 p_1$ or $p_0 \geq 0.25 p_1$. The gas pre-charge pressure must be as close as possible to the minimum working pressure p_1 to obtain maximum storage. Special values for p_0 are used in pulsation damping and shock absorber applications ($p_0 = 0.8 p_1$). Other parameters to be determined are the volume of fluid ΔV that needs to be stored ($\Delta V = 0.75 V_0$), the maximum required flow rate and the operating temperature.

1.2.1 Sizing Accumulators for Isothermal Condition

For isothermal condition, the Boyle–Mariotte law can be rewritten in terms of V_1 and V_2 as

$$V_1 = V_0 \left(\frac{p_0}{p_1} \right) \tag{1.6}$$

and

$$V_2 = V_0 \left(\frac{p_0}{p_2} \right) \tag{1.7}$$

The difference between V_1 at the minimum operating pressure p_1 and V_2 at the maximum operating pressure p_2 gives the amount of the stored fluid ΔV. Thus,

$$\Delta V = V_1 - V_2 = V_0 \left(\frac{p_0}{p_1} \right) - V_0 \left(\frac{p_0}{p_2} \right)$$

$$\Rightarrow \Delta V = V_0 \left(\frac{p_0}{p_1} - \frac{p_0}{p_2} \right)$$

$$\Rightarrow V_0 = \frac{\Delta V}{p_0 / p_1 - p_0 / p_2} \tag{1.8}$$

In the above equations, V_1 and V_2 are nitrogen volumes at the pressures p_1 and p_2, and V_0 is the nitrogen pre-charge volume at the pressure p_0 in liters. It is the maximum volume of gas that can be stored in the accumulator. The size of the accumulator, while conforming to the standard available sizes, should be at least 5–10% more than this volume V_0. Any increase in the value of ΔV results in a corresponding increase in the size of the accumulator. Likewise any decrease in the value of p_0 or the value of $(p_1 - p_2)$ requires an accumulator of higher volume.

1.2.2 Sizing Accumulators for Adiabatic Condition

Starting from the basic formula $p_0V_0^n = p_1V_1^n = p_2V_2^n$, it can be shown that for adiabatic conditions, the values of maximum nitrogen volume V_0 at the pre-charge pressure p_0 and the stored volume of oil are given by the following equations:

$$\Delta V = V_0\left[\left(\frac{p_0}{p_1}\right)^{0.7143} - \left(\frac{p_0}{p_2}\right)^{0.7143}\right]$$

$$\Rightarrow V_0 = \frac{\Delta V}{[(p_0/p_1)^{0.7143} - (p_0/p_2)^{0.7143}]} \qquad (1.9)$$

Here, the value of the polytropic exponent n is taken equal to 1.4 and therefore $1/n$ becomes 0.7143. Here again intermediate values can be used for more accurate results.

1.2.3 Sizing Accumulators for Emergency Reserve

This is a typical application where both isothermal and adiabatic conditions prevail due to slow storage and quick discharge. For this condition, the accumulator volume is given by

$$V_0 = \frac{\Delta V(p_2/p_0)}{[(p_2/p_1)^{0.7143} - 1]}$$

$$\Rightarrow \Delta V = V_0 p_0 \frac{\left[\left(\frac{p_2}{p_1}\right)^{0.7143} - 1\right]}{p_2} \qquad (1.10)$$

1.2.4 Sizing Accumulators for Pulsation Damping

Pulsation damping is typically an adiabatic condition because both storage and discharge have to be accomplished in a very short time. Because pressure pulsation is a phenomenon associated with piston pumps, the stored volume ΔV is a product of the pump displacement q in liters and a constant $\boldsymbol{k}$ that depends on whether the pump is single-acting or double-acting and the number of pistons involved. Pressure pulsation is highest in a single-acting single-piston pump delivering large flows at high pressures. Here the k factor is about 0.69. A single-piston double-acting pump has the same k factor as that of a double-piston single-acting pump whose k factor is equal to 0.29. A three- or four-piston single-acting pump has a k factor of 0.12. For any other pump configuration, an average k factor of 0.05 can be taken with reasonable accuracy.

The stored volume is given by

$\Delta V = kq$ where q is the pump flow rate in LPM/RPM × number of piston. Now

$$V_0 = \frac{\Delta V}{[(p_0/p_1)^{0.7143} - (p_0/p_2)^{0.7143}]} \qquad (1.11)$$

where$p_1 = (p - x)$ and $p_2 = (p + x)$. Here, p is the average working pressure in bar and x is given by $(a \times p)/100$ bar , where a is percentage of pulsation.

1.2.5 Sizing Accumulators for Hydraulic Line Shock Damping

A suitable accumulator can neutralize water hammering in pipes due to shock waves caused by sudden closure of valves. Typical applications can be found in water, fuel and oil distribution circuits. The volume of the accumulator required to absorb the shock waves is given by

$$V_0 = \frac{4Qp_2(0.0164L\,2t)}{1000\ \ (p_2\,2\,p_1)}$$

where Q is the flow rate in LPM;p_2 is the maximum pressure in bar;L is the length of the pipe in meters;t is the acceleration, deceleration or the valve shut-off time in s and p_1 is the operating pressure with free flow in bar. In some applications such as hydraulic lift trucks, accumulators may be used to absorb hydraulic shocks when the valve shifts or to absorb load-induced pressure surges when the truck runs over uneven ground.

1.2.5.1 Influence of Variation in Temperature on Accumulator Volume

The nitrogen pre-charge pressure in an accumulator is based on the expected maximum rise in the circulating hydraulic oil temperature. This temperature can drop due to changes in environmental factors resulting in a comparable drop in the pre-charge pressure. According to Gay Lussac's law, this variation in pressure affects the volume resulting in lower accumulator capacity. It is therefore necessary to have an accumulator of higher volume so that the useful volume ΔV remains unaffected.

1.2.5.2 Influence of Pressure on Accumulator Volume

The fact that n, the value of adiabatic index, lies between 1 and 1.4 is true for perfect gases. But nitrogen used in accumulators does not behave like a perfect gas when pressure increases and this affects the value of n and consequently the accumulator volume V_0. So for pressures between 200 bar and 350 bar, the value of adiabatic index n may be assumed to lie in the range 1.5–1.6.

1.2.6 Sizing of Additional Gas Bottles

Sometimes the size V_0 as determined by Eqs. (1.6)–(1.12) may be more than the available accumulator sizes or the size so determined is too big for accommodation within the frame of the machine. Also sometimes it may become necessary to maintain a small difference between p_1 and p_2, which results in a higher stored volume ΔV and a much larger accumulator volume V_0. In such cases, it is convenient to get required volume by additional bottles.

For example, if an application requires 43 L of fluid to be discharged adiabatically between 70 and 55 bar, the total volume required would be close to 434 L. The volume requirement is high because the pressure difference is small. If a 22.5 cm bore piston accumulator can hold 125 L, then auxiliary gas bottles can be installed some distance away to meet the balance of volume requirement.

In applications such as energy reserve volume compensation and hydraulic line shock damping, it is recommended that a higher pre-charge pressure $p_0 = 0.97p_1$ be maintained. After obtaining the value of V_0 in the normal way, it must be split into two portions: The minimum indispensable portion that is contained in the accumulator and the portion that is contained in additional gas bottles.

Lecture 29

ACCUMULATORS [CONTINUED]

1.3 Applications of Accumulators

There are five basic applications where accumulators are used in a hydraulic system:

1. **Accumulator as an auxiliary power source:** The purpose of accumulator in this application is to store the oil delivered by the pump during a portion of the work cycle. The accumulator then releases the stored oil on demand to complete the cycle, thereby serving as a secondary power source.

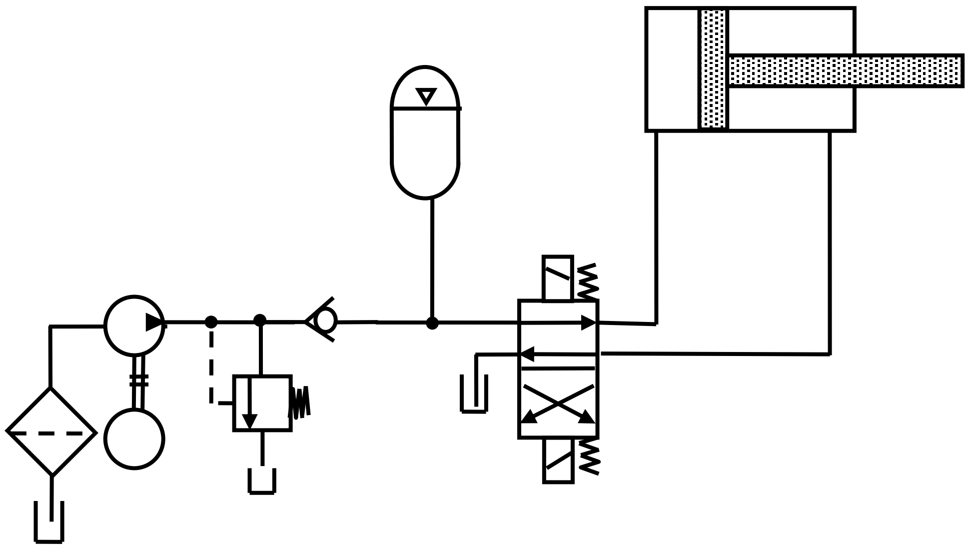


Figure 1.8 Accumulator as an auxiliary power source.

The schematic diagram is shown in Fig. 1.8. When the four-way valve is manually activated, oil flows from the accumulator to the blank end of the cylinder. This extends the piston until it reaches the end of the stroke. When the cylinder is in its fully extended position, the accumulator is being charged. The four-way valve is then deactivated for retraction of the cylinder oil flows from both the pump and accumulator to retract the cylinder rapidly.

2.Accumulator as a leakage compensator: An accumulator can be used as a compensator for internal and external leakage during an extended period in which the system is pressurized but not in operation. The pump charges the accumulator and the system until the maximum pressure sets the pressure switch ON. The schematic diagram is shown in Fig. 1.9. The contacts on the pressure switch then open to automatically stop the electric motor that drives the pump. The accumulator then supplies leakage oil to the system during a long period. Finally, when the system pressure drops to the minimum pressure setting of the pressure switch, it closes the electrical circuit of the motor until the system gets recharged. The check valve is placed between the pump and accumulator so that the pump does not reverse when the motor is stopped and does not permit all the accumulator charge to drain back into the power unit. With this circuit, the only time the power unit operates is when the pressure drops to an unsafe operating level. This saves electric power and reduces the heat in the system.

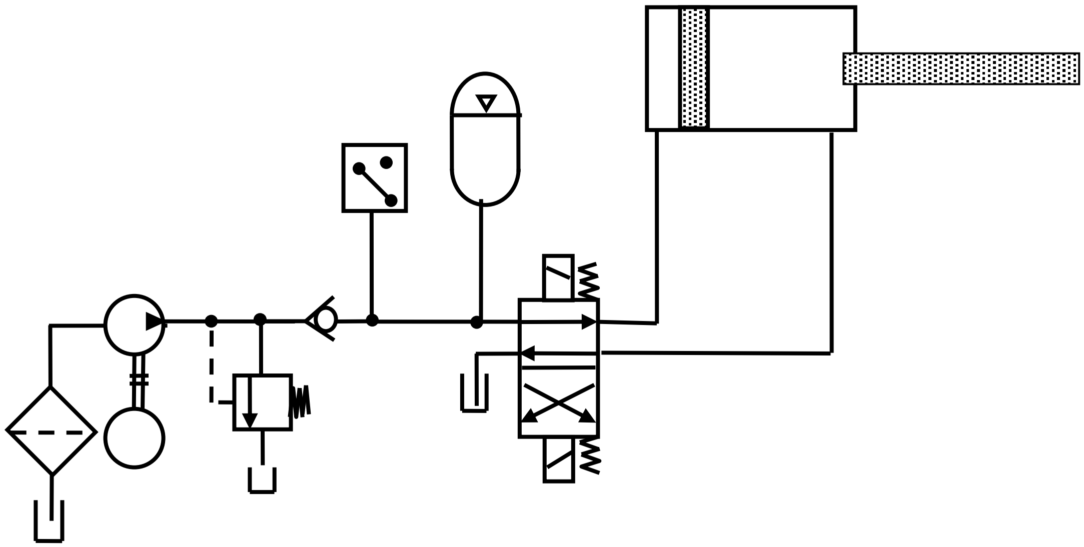


Figure 1.9 Accumulator as a leakage compensator**.**

3.Accumulator as an emergency power source: In some hydraulic systems, safety dictates that a cylinder be retracted even though the normal supply of oil pressure is lost due to a pump or electrical power failures. The schematic diagram is shown in Fig. 1.10. In it, a solenoid activated three-way valve is used along with the accumulator. When the three-way valve is energized, oil flows to the blank end of the cylinder and also through the check valve into the accumulator and the rod end of the cylinder. The accumulator charges as the cylinder extends.

If the pump fails due to an electric failure, the solenoid de-energizes, shifting the valve to its spring offset mode. Then the oil stored under pressure is forced from the accumulator to the end of the cylinder. This retracts the cylinder to its starting position.

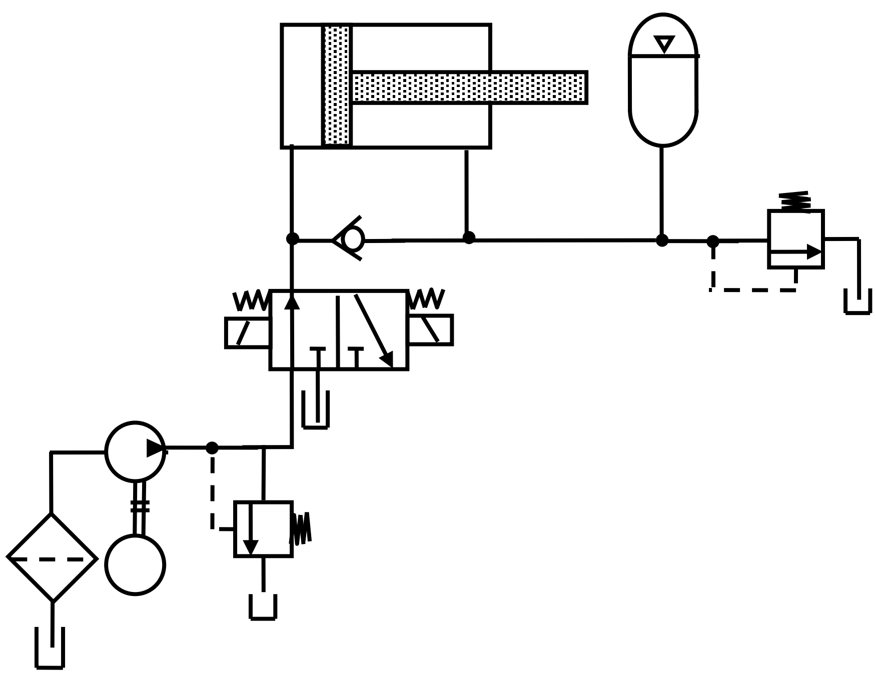


Figure 1.10 Accumulator as an emergency power source**.**

4. **Accumulator as a hydraulic shock absorber:** One of the important applications of accumulators is the elimination of hydraulic shock.The schematic diagram is shown in Fig. 1.11. Hydraulic shock is

caused by the sudden stoppage or declaration of a hydraulic fluid flowing at relatively high velocity in a pipe line. Rapidly closing a valve creates a compression wave. This compression wave travels at the speed of sound upstream to the end of the pipe and back again to the closed valve, which causes an increase in pressure.
The resulting rapid pressure pulsations or high-pressure surges may cause damage to the hydraulic system components. If an accumulation is installed near the rapidly closing valve, the pressure pulsations or high-pressure surges are suppressed.

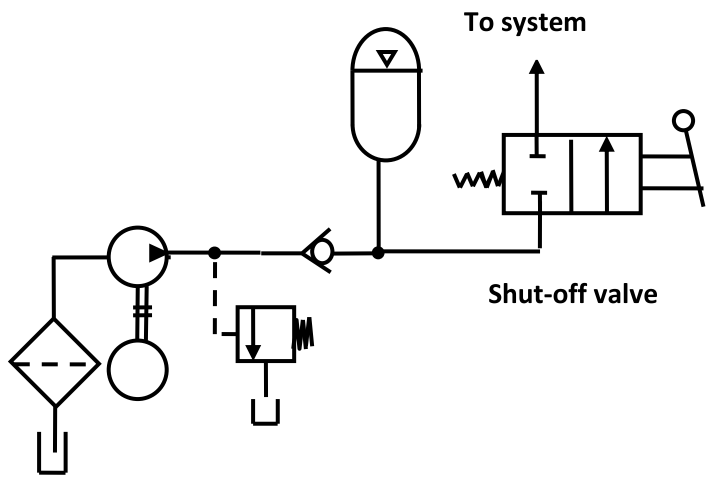


Figure 1.11 Accumulator as a hydraulic shock absorber**.**

5. **Accumulator as a thermal expansion compensator:** When closed-loop hydraulic systems are subjected to heat conditions, both the pipe lines and the hydraulic fluid expand volumetrically. Because the coefficient of cubical expansion of most fluids is higher than that for pipe material, this expanded liquid volume increases the entire system pressure. This condition may cause pressures to exceed the limits of safety and may damage the system components. Under these conditions, an accumulator of proper capacity pre-charged to the normal system working pressure is installed. It takes up any increase in the system fluid volume, thus reducing the system pressure to its safe limits. The accumulator also feeds the required volume into the system as thermal contraction takes place.The schematic diagram of such an arrangement is shown in Fig. 1.12.

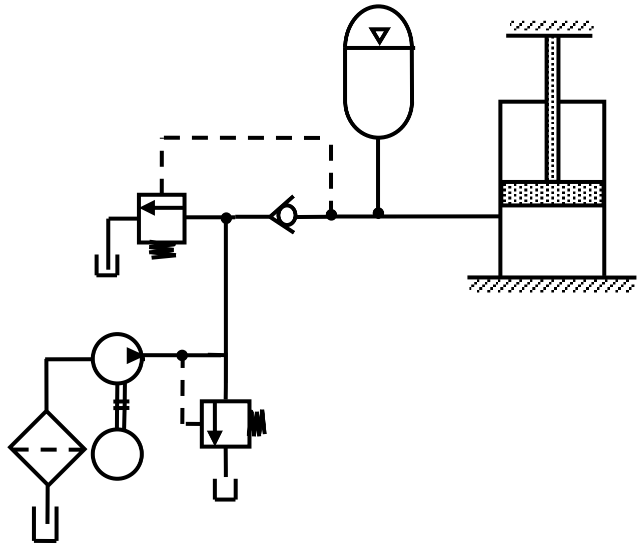


Figure 1.12 Accumulator as a thermal expansion compensator.

Example 1.1

A hydraulic cylinder has to move a certain load through a certain distance in 1 s at a pressure of 140 bar. An accumulator is integrated into the circuit to provide peak power. The accumulator is charged for the first 20 s and discharged in 2 s. The delivery expected from the accumulator is 0.6 L in 2 s as the pressure falls from 250 to 140 bar. Calculate the accumulator volume. The operating temperature is +25–70ºC. Also calculate the reduction in input power due to the accumulator.

Solution: This is a case of isothermal compression and adiabatic expansion. The equation considered here is

$$V_0 = \frac{\Delta V(p_2 / p_0)}{[(p_2 / p_1)^{0.7143} - 1]}$$

Because the maximum system pressure is above 200 bar, we consider a higher value of 1.6 for the adiabatic index *n*. Therefore, 1/1.6 = 0.625. Inserting this value into the above equation, we have $p_0 = 0.8 p_1 = 0.9 \times 140 = 126$ bar (gauge) =127 bar (absolute). So

$$V_0 = \frac{\Delta V(p_2 / p_0)}{[(p_2 / p_1)^{0.7143} - 1]}$$

$$= \frac{0.6(251/127)}{[(251/141)^{0.7143} - 1]}$$

$$= 2.74 \text{ L}$$

If we apply the correction for temperature change because the pre-charge pressure p_0 was based on the maximum temperature indicated, we have the new corrected volume

$$V_{0T} = 2.74\ (343/298) = 3.15 \text{ L}$$

A 3.5 L accumulator would effectively serve the purpose. The delivery from the accumulator is 0.6 L in 2 s, 0.3 L/s or 18 L/min.

In the absence of accumulator, the pump has to supply all of this delivery at a pressure of 140 bar. The HP required would be

$$(18 \times 140)/600 = 4.2 \text{ kW}$$

Here the efficiency of the pump is not considered. If the accumulator is included in the circuit, the pump has to deliver 0.6/20 = 0.03 L/s or 1.8 L/min sufficient to charge the accumulator to a pressure of 250 kgf/cm^2 within the time interval of 20 s. Here again the flow required to retract the cylinder is not considered.

The power requirement in this case would be (1.8 × 250)/600 = 0.75 kW.

The power saved is (4.2 – 0.75) = 3.45 kW.

Example 1.2

A hydraulic molding press is kept closed at a maximum system pressure of 200 kgf/cm^2 for a duration of 60 min during the curing period. The maximum leakage permitted during this period is 2 cm^3/minute and minimum fall in pressure permitted is 198 kgf/cm^2. Calculate the accumulator volume.

Solution: This is an application where the accumulator is used as a leakage compensator under isothermal conditions. Therefore, the equation used for this is

$$V_0 = \frac{\Delta V}{[p_0 / p_1 - p_0 / p_2]}$$

Here ΔV = (2×60)/1000 = 0.12 L, p_0 = (0.9×198) = 178 kgf/cm^2.Inserting these values in the above equation, we have

$$V_0 = \frac{0.12}{[179/199 - 179/201]} = 13.3\text{L}$$

A standard 15 L accumulator would meet the requirement.

Example 1.3

A three-piston single-acting pump of flow rate Q = 133 LPM is operating at 20 bar and at 148 RPM. The working temperature is 400°C. Calculate the accumulator volume needed to limit the remaining pulsation to –2.5%.

Solution: This is a typical condition of pulsation damping in adiabatic phase due to high-speed compression and expansion. For this condition,

$$V_0 = \frac{\Delta V}{[(p_0 / p_1)^{0.7143} - (p_0 / p_2)^{0.7143}]}$$

Here –V = kq where k for a three-piston single-acting arrangement can be taken as 0.12. The pump displacement q is given by

$$q = 133/3 \times 148 = 0.3 \text{ L}$$

Now

$$a = \frac{2.5}{100} \times 100 = 0.5$$

Now $p_1 = (p - a) = (20 - 0.5) = 1.5$ bar, $p_2 = (p + a) = (20 + 0.5) = 20.5$ bar, $p_0 = (0.7 \times 20) = 14$ bar. Substituting these values into the above equation, we have

$$V_0 = \frac{(0.12 \times 0.3)}{[(15/19.5)^{0.7143} - (15/20.5)^{0.7143}]} = 1.5\,\mathrm{L}$$

A 2 L accumulator would be adequate. Because all calculations are based on the absolute temperature and pressure, the temperatures must be expressed in degree Kelvin (°K) that is obtained by adding 273 to the operating temperatures recorded in °C. Similarly to obtain absolute pressure, 1 bar is added to the values of *P* given.

Example 1.4

A dead-load accumulator has a cylinder bore of 500 mm and is to operate at a system pressure of 200 bar. What is the dead load required?

Solution: The dead load required will be the product of piston area times the pressure.

$$\text{Load} = \pi/4 \times (500/1000)^2 \times 200 \times 10^5\ \mathrm{N}$$
$$= 3.93 \times 10^6\ \mathrm{N}\ \text{(approx. 400 Ton)}$$

Example 1.5

Calculate the increase in pressure if a cylinder 300 mm length × 500 mm stroke is locked in the extended condition and then subjected to a 20°C rise in temperature. What is the volume of fluid to be stored in an accumulator fitted to compensate for thermal expansion? (Take the bulk modulus to be 15000 bar).

Solution: Total volume of fluid contained in the cylinder is

$$\pi/4 \times (0.3)^2 \times 0.5\ \mathrm{m}^3 = 35.3\ \mathrm{L}$$

Change in volume is

$$35.3 \times 0.0007 \times 20 = 0.49\ \mathrm{L}$$

which is the additional volume to be stored. Without an accumulator, the change in pressure in the closed system is Δp, that is,

$$\Delta p = \beta(\Delta V/V)$$
$$= 15000 \times 0.49/35.3$$
$$= 210\ \text{bar}$$

Example 1.7

What size of accumulator is necessary to supply 500 cc of fluid in a hydraulic system of maximum pressure of 200 bar that drops to 100 bar minimum? Assuming N_2 gas pre-charged of 66 bar, find adiabatic and isothermal solution.

Solution: Stages of pre-cl , charging and delivery are shown in Fig. 1.13.

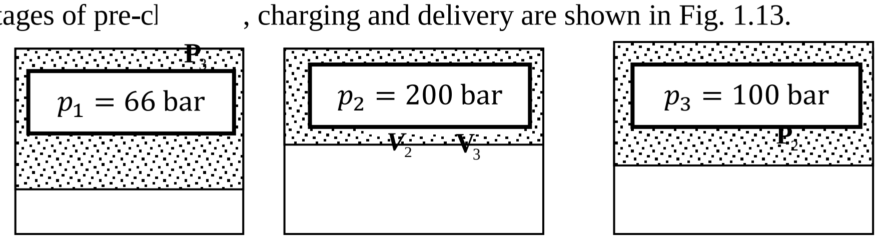


Figure 1.13

We shall assume the expansion and compressions as isothermal and adiabatic.

(a) Isothermal condition:

$$V_3 - V_2 = \text{volume of oil that can be delivered} = 0.005\text{m}^3$$

$$V_3 = V_2 + 0.005$$

Using $p_2V_2 = p_3V_3$ we get

$$V_2 = \frac{p_3}{p_2} \times V_3 = \left[\frac{100+1.013}{200+1.013}\right] \times [V_2 + 0.005]$$

$$= 0.5025\ V_2 + 0.002512$$

$$= 0.00505\ \text{m}^3$$

Using $p_1V_1 = p_2V_2$ we get

$$V_1 = \frac{p_2}{p_1} \times V_2 = \left[\frac{200+1.013}{60+1.013}\right] \times 0.00505 = 0.01663\ \text{m}^3$$

(b) Adiabatic condition:

$$p_2V_2^{1.4} = p_3V_3^{1.4}$$

$$\Rightarrow V_2 = V_3 \times \left(\frac{p_1}{p_2}\right)^{1/1.4}$$

$$= (V_2 + 0.005)\left(\frac{101.013}{201.013}\right)^{0.714}$$

$$= 0.6118V_2 + 0.003059$$

$$= 0.00788\ \text{m}^3$$

Using $p_1V_1^{1.4} = p_2V_2^{1.4}$ we get

$$V_1 = 0.00788\left(\frac{201.013}{67.013}\right)^{0.714} = 0.01726\ \text{m}^3$$

Example 1.8

What size of accumulator is necessary to supply 4917 cm^3 of fluid into a hydraulic system of maximum operating pressure of 207 bar that drops to minimum 103.5 bar? Assuming a nitrogen gas pre-charge of accumulator to be 67 bar, obtain both isothermal and adiabatic solutions.

Solution: Stages of pre-charging, charging and delivery are shown in Fig. 1.14.

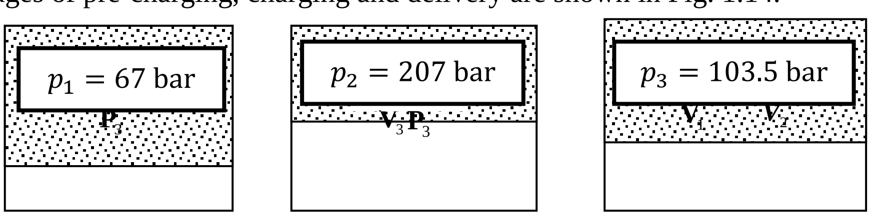


Figure 1.14

We shall assume the expansion and compressions to be isothermal and adiabatic.

(a) Isothermal condition:

$V_3 - V_2$ = volume of oil that can be delivered = 0.004917 m^3

$V_3 = V_2 + 0.004917$

Using $p_2V_2 = \quad p_3V_3$ we get

$$V_2 = \frac{p_3}{p_2}\times V_3 = \left[\frac{103.5+1.013}{207+1.013}\right]\times\left[V_2+0.004917\right]$$

$= 0.5025\,V_2 + 0.00247$

$= 0.00496$ m^3

Using $p_1V_1 = p_2V_2$ we get

$$V_1 = \frac{p_2}{p_1}\times V_2 = \left[\frac{207+1.013}{67+1.013}\right]\times 0.00496 = 0.01518 \text{ m}^3$$

(b) Adiabatic condition: Using $p_2V_2^{1.4} = p_3V_3^{1.4}$ we get

$$V_2 = V_3\times\left(\frac{p_1}{p_2}\right)^{1/1.4}$$

$$= (V_2+0.004917)\left(\frac{104.013}{208.013}\right)^{0.714}$$

$= 0.0117V_2 + 0.003007$

$= 0.007740$ m^3

Using $p_1V_1^{1.4} = p_2V_2^{1.4}$ we get

$$V_1 = 0.007746\left(\frac{208.013}{68.013}\right)^{0.714}$$

$= 0.0172$ m^3

Example 1.9

A gas-charged accumulator supplies energy to a system with 15 L of oil within the pressure range of 125–175 bar. If the accumulator has pre-charged pressure of 90 bar, size the accumulator for (a) isothermal and (b) adiabatic pressures.

Solution: Stages of pre-charging, charging and (are sh Fig. 1.15.

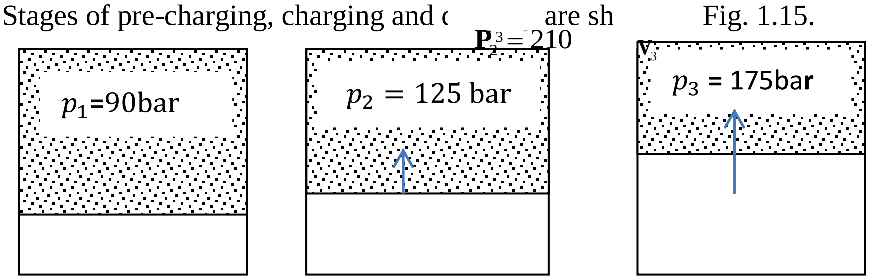


Figure 1.15

Let the pre-charging pressures be p_1 and V_1. Gas is compressed by incoming oil from pressure 90–175 bar. When the bladder is compressed to 175 bar, the volume of oil inside the accumulator is 15 L. Therefore, we can write

$$V_3 - V_1 = 15$$

(a) Isothermal condition:

$$p_1 V_1 = p_3 V_3$$

$$\Rightarrow 90 \times V_1 = 175 \times (V_1 - 15)$$

$$\Rightarrow 90\ V_1 = 175 V_1 - 2625$$

$$\Rightarrow V_1 = \frac{2625}{86} = 30.88\ \text{ L}$$

(b) Adiabatic condition:

$$p_1 V_1^{\gamma} = p_3 V_3^{\gamma}$$

$$\Rightarrow 90(V_1)^{\gamma} = 175(V_1 - 15)^{\gamma}$$

since there will be some heat loss and perfect adiabatic condition is not possible.Take $\gamma = 1.3$. We get

$$90(V_1)^{1.3} = 175\ (V_1 - 15)^{1.3}$$

$$\Rightarrow \frac{90}{175} = \left(\frac{V_1 - 15}{V_1}\right)^{1.3}$$

$$\Rightarrow \frac{V_1 - 15}{V_1} = \left(\frac{90}{175}\right)^{1/1.3}$$

$$\Rightarrow V_1 - 15 = 0.6\ V_1$$

$$\Rightarrow V_1 = 37.5\ \text{L}$$

Example 1.10

An accumulator has a ram diameter of 0.4 m and lift of 10 m. It is loaded with 1000 kN of total weight. The packing friction is 5% of the load on the ram. Find the power delivered to the main if the RAM descends steadily through its full stroke in 5 min while the pump delivers 50 LPS through the accumulator.

Solution: The pressure developed in an accumulator due to weight is

$$\text{Pressure} = \frac{\text{net load}}{\text{Area}}$$

$$= \frac{1000 \times 0.95}{\pi / 4 \times 0.4^2} = 7560\,\text{kPa}$$

Pressure energy (head) supplied by the accumulator to water is

$$\frac{p}{\rho \times g} = \frac{7560}{9810} = 770\,\text{m of water}$$

Energy supplied by the accumulator (during ascend) is

$$Q \times g \times h = 50 \times 9.81 \times 770$$

$$= 377685\ \text{W}$$

Energy supplied by the accumulator (during descend)

$$\frac{\text{Net load} \times \text{ Stroke}}{\text{Time of descent}} = \frac{1000 \times 1000 \times 0.95 \times 10}{300}$$

$$= 31667\ \text{W}$$

Total power delivered is

$$377685 + 31667 = 409352\ \text{W} = 409.4\ \text{kW}$$

Example 1.11

A gas-charged accumulator supplies energy to a system with 6.7 L of oil within the pressure range of 150–110 bar. The accumulator has the pre-charge pressure of 85 bar. What should be the size of the accumulator if the oil is to be supplied (a) in about 5 s and (b) in about 5 min?

Solution: Stages of pre-charging, charging and delivery are shown in Fig. 1.16.

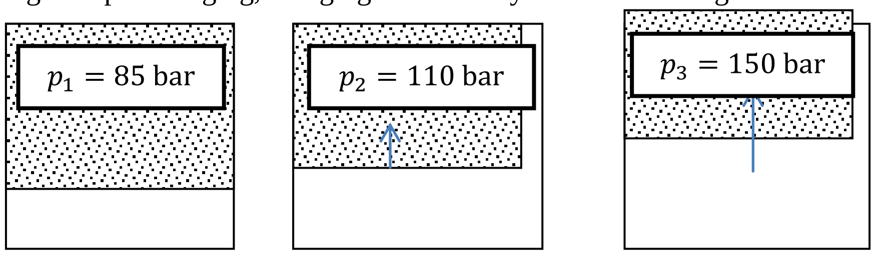


Figure 1.16

Let the pre-charging pressure be p_1 (85 bar). Gas is compressed by incoming oil from pressure 85 to 150 bar. When the bladder is compressed to 150 bar, the volume of oil inside the accumulator is 6.7 L. Therefore, we can write

$$V_3 - V_1 = 6.7$$

Consider the adiabatic condition with $\gamma = 1.25$.

(a) In about 5 s:

$$p_1 V_1^{\gamma} = p_2 V_2^{\gamma}$$

$$\Rightarrow 85(V_1)^{\gamma} = 150 \times \left(V_1 - \frac{6.7}{5}\right)^{\gamma}$$

$$\Rightarrow 85(V_1)^{1.25} = 150 \times (V_1 - 1.34)^{1.25}$$

$$\Rightarrow \frac{85}{150} = \left(\frac{V_1 - 1.34}{V_1}\right)^{1.25}$$

$$\Rightarrow \left(\frac{V_1 - 1.34}{V_1}\right) = \left(\frac{85}{150}\right)^{1/1.25}$$

$$\Rightarrow V_1 = 3.144 \text{ LPS}$$

Capacity of accumulator = 3.144 × 5 = 15.72 L

(b) In about 5 min:

Capacity of accumulator = 3.144 × 5 × 60 = 943.2 L

Example 1.12

A circuit has been designed to crush a car body into bale using a 152 mm diameter hydraulic cylinder. The hydraulic is to extend 2.54 m during a period of 10 s. The time between crushing strokes is 5 min. The following accumulator gas absolute pressures are given: p_1(gas pre-charge pressure) = 84 bar (abs), p_2(gas charge pressure when the pump is turned ON) = 210 bar (abs) = pressure relief value setting, p_3(minimum pressure required to actuate load) = 126 bar (abs).

(a) Calculate the required size of the accumulator.

(b) What are the pump hydraulic kW power and flow requirements with and without accumulator?

Solution: Stages of pre-charging, charging and delivery are shown in Fig. 1.17.

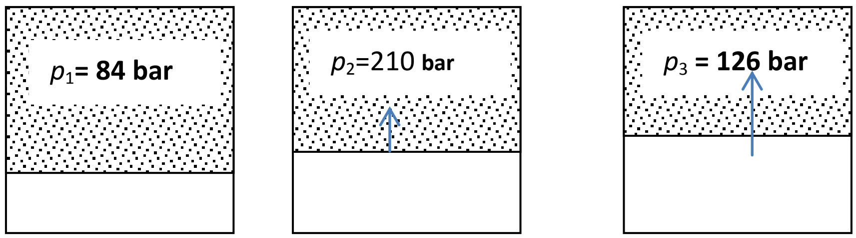


Figure 1.17

Let the pre-charging pressure be p_1 (85 bar). Gas is compressed by incoming oil from pressure 84 to 210 bar and accumulator is discharged till the pressure reaches 126 bar.

(a) Without accumulator: Let the compression and expansion of gas follow isothermal law:

$$p_1V_1 = p_2V_2 = p_3V_3$$

Here V_c is the volume of hydraulic cylinder. It can accommodate ($V_3 - V_2$) amount of oil

$$V_c = (V_3 - V_2)$$

$$\Rightarrow p_3V_3 = p_2V_2$$

$$\Rightarrow V_3 = \frac{p_2V_2}{p_2} = \frac{210 \times V_2}{126} = 1.67V_2 \quad (1.13)$$

$$\Rightarrow V_c = \frac{\pi}{4}d^2I = \frac{\pi}{4}(0.152)^2 \times 2.54 = 0.0461\text{m}^3 = (V_3 - V_2) \; (1.14)$$

Using Eq. (1.13) in Eq. (1.14) and solving, we get

$$V_2 = 0.0688 \text{ m}^3$$

$$V_3 = 0.155 \text{ m}^3$$

$$V_1 = \frac{p_2V_2}{p_1} = \frac{210 \times 0.0688}{840} = 0.172\,\text{m}^3 = 172\,\text{L}$$

(b) With accumulator: The pump charges accumulator in every 2.5 min. In other words, two times in five minutes.

Flow supplied by the pump

$$Q_p = \frac{2(V_3 - V_2)}{30} = \frac{2(46.1)}{300} = 0.307\,\text{LPS}$$

Neglecting all losses, power supplied to the pumpis

$$p_{\text{pump}} = p_2 \times Q_{\text{pump}}$$

$$= \frac{(210 \times 10^5)(0.307 \times 10^{-3})}{1000} = 6.45 \text{ kW}$$

Without accumulator: The pump extends cylinder in 10 s. Flow supplied by the pump is

$$Q_p = \frac{46.1}{10} = 0.461 \text{ LPS}$$

Neglecting all losses, power supplied to the pump is

$$p_{\text{pump}} = p_2 \times Q_{\text{pump}}$$

$$= \frac{(126 \times 10^5)(461 \times 10^{-5})}{1000} = 58.1 \text{ kW}$$

It can be seen that flow and power requirement by the pump is more without accumulator.

Example 1.13

What size of accumulator is necessary to supply 10000 cm^3 of fluid is a hydraulic system of maximum pressure of 200 bar to 100 bar minimum. Assuming N_2 gas per-charged pressure of 80 bar. Find adiabatic and isothermal solution.

Solution: Let

V_1 = Volume of accumulator (cm^3)
V_2 = Volume of gas at high pressure (cm^3)
p_2 = Maximum pressure, bar = 200 bar
p = Minimum pressure, bar = 100 bar
p_1 = Per-charged pressure, bar = 80 bar

Let V_1be the volume of gas in the accumulator at pre-charged 80 bar and V_2be the volume of gas in the accumulator at 200 bar. Now

$$V_1 = V_2 + 10000 \text{ cm}^3$$
$$\Rightarrow V_2 = V_1 - 10000$$

(a) **Adiabatic process:**We have $p_1 V_1^{\gamma} = p_2 V_2^{\gamma}$. Now $\gamma = 1.25$. So

$$(80)V_1^{\gamma} = (200)(V_1 - 10000)^{\gamma}$$

$$\Rightarrow \frac{80}{200} = \left(\frac{V_1 - 10000}{V_1}\right)^{\gamma}$$

$$\Rightarrow 0.4 = \left(\frac{V_1 - 10000}{V_1}\right)^{1.25}$$

$$\Rightarrow \frac{V_1 - 10000}{V_1} = (0.4)^{1/1.25} = (0.4)^{0.8} = 0.4804$$

$$\Rightarrow V_1 - 10000 = 0.4804 V_1$$

$$\Rightarrow V_1 - 0.4804\ V_1 = 10000$$

$$\Rightarrow 0.5195\ V_1 = 10000$$

$$\Rightarrow V_1 = 19249.3 \text{ cm}^3$$

Size of accumulator = 19249.3 cm^3

(b) Isothermal process: We have

$$p_1V_1 = p_2V_2$$
$$\Rightarrow 80 \times V_1 = 200 \times (V_1 - 10000)$$
$$\Rightarrow \frac{80}{200} = \frac{V_1 - 10000}{V_1}$$
$$\Rightarrow 0.4 = \frac{V_1 - 10000}{V_1}$$
$$\Rightarrow V_1 - 10000 = 0.4V_1$$
$$\Rightarrow V_1 - 0.4V_1 = 10000$$
$$\Rightarrow 0.6\ V_1 = 10000$$
$$\Rightarrow V_1 = 16666.7\ \text{cm}^3$$

Size of accumulator $=16666.7\ \text{cm}^3$

Objective-Type Questions
Fill in the Blanks

1. A hydraulic accumulator is a device that stores the potential energy of an ______ held under pressure by an external source against some dynamic force.
2. The main disadvantage of a weight-loaded or gravity accumulator is its extremely ______ size and ______ weight.
3. A spring-loaded accumulator stores energy in the form of a ______ spring.
4. Pulsation damping is typically an ______ condition because both storage and discharge have to be accomplished in a very short time.
5. The nitrogen pre-charge pressure in an accumulator is based on the expected maximum rise in the circulating hydraulic oil ______.
6. An accumulator permits _____ to be absorbed and stored in a hydraulic system.
7. Air or _____ should never be used in gas-charged accumulators.

State True or False

1. A gas-loaded accumulator is popularly used in industries.
2. In a bladder accumulator, the bladder is filled with oxygen until the designed pre-charge pressure is achieved.
3. An accumulator can be used as a compensator for both internal and external leakages.
4. One of the important applications of accumulators is the elimination of hydraulic shock.
5. An accumulator can be used as a fail–safe device.

Review Questions

1. Define an accumulator and explain its function.
2. What are the different types of accumulators?
3. Mention some of the industrial applications of an accumulator. Explain any one of them with an example.
4. Why are accumulators used?
5. Define and derive an expression for the volumetric capacity of bladder-type accumulators.
6. Explain the construction and operation of piston-type accumulators.
7. Explain the construction and operation of bladder-type accumulators.
8. Explain the construction and operation of diaphragm-type accumulators.
9. Discuss in detail the application of hydraulic accumulators as energy storage elements. Draw a hydraulic circuit for this application.
10. Discuss in detail the application of hydraulic accumulators for protection against shocks.
11. Discuss in detail the application of hydraulic accumulators in protecting against thermal expansion.
12. Discuss in detail the application of hydraulic accumulators for internal leakage compensation and the application of constant pressure.
13 What is the difference between separator and non-separator types of accumulators.
14 Name three different types of separator-type accumulators.
15. What are the advantages of bladder accumulators over piston accumulators?

Answers
Fill in the Blanks

1. Incompressible fluid
2. Large, heavy
3. Compressed
4. Adiabatic
5. Temperature
6. Energy
7. Pure oxygen

State True or False

1. True
2. False
3. True
4. True
5. True

Lecture 30

ACCESSORIES USED IN FLUID POWER SYSTEMS

Learning Objectives

Upon completion of this chapter, the student should be able to:

- Differentiate between internal leakage and external leakage.
- Explain various types of seals used in fluid power.
- Explain the working of a durometer hardness tester.
- List various functions of a reservoir.
- Describe various types of reservoirs used in fluid power.
- Explain the design considerations for reservoirs used in fluid power.
- Carry out the heat transfer calculation of reservoirs.
- Differentiate between a strainer and a filter.
- Describe various types of filters used in fluid power applications.
- Define the beta ratio efficiency of a filter.
- Explain the importance of a heat exchanger.
- Size a heat exchanger.

1.1 Introduction

Any hydraulic system is associated with a major problem, that is, leakage. This reduces efficiency and increases the power losses. Hence, sealing devices play a vital role in a hydraulic system by increasing the efficiency and decreasing the power losses. Leakage can be overcome by proper maintenance of the system and proper selection of seals and sealing at the design stage.

Leakage in a hydraulic system can be classified as follows:

1. Internal leakage: This occurs in hydraulic components built with operating clearances. Moving parts need to be lubricated and leakage path may be designed solely for this purpose. Internal leakage does not cause loss of fluid because the fluid returns to the reservoir. This leakage increases the clearances between mating parts due to wear. If the entire system leakage becomes large enough, the actuators do not operate properly.

2. External leakage: External leakage represents loss of fluid from the system. It also represents a safety hazard. Improper assembly of pipe fittings is the most common cause of external leakage. Over-tightened fittings may become damages or vibrations can cause properly tightened fittings to become loose. Failure to connect drain lines, excessive operating pressure and contamination might cause the fluid to externally leak.

1.2 Functions of Seals

Seals are used in hydraulic systems to prevent excessive internal and external leakage and to keep out contamination.Various functions of seals include the following:

1. They prevent leakage – both internal and external.
2. They prevent dust and other particles from entering into the system.
3. They maintain pressure.
4. They enhance the service life and reliability of the hydraulic system.

1.2.1 Classification of Hydraulic Seals
Hydraulic seals can be classified as follows:

1. **According to the method of sealing:**

    - *Positive sealing:* A positive seal prevents even a minute amount of oil from getting past. A positive seal does not allow any leakage whatsoever (external or internal).
    - *Non-positive sealing:*A non-positive seal allows a small amount of internal leakage, such as the clearance of the piston to provide a lubrication film.

2. **According to the relative motion existing between the seals and other parts:**

    - **Static seals:** These are used between mating parts that do not move relative toone another. Typical examples are flange gaskets and seals, o-rings, etc. These are relatively simple. They are essentially non-wearing and usually trouble-free if assembled properly.
    - **Dynamic seals:** These are assembled between mating parts that move relative to each other. Hence, dynamic seals are subject to wear because one of the mating parts rubs against the seal.

3. **According to geometrical cross-section:**

- **O-ring seal:**O-ring is the most widely used seal for hydraulic systems. It is a molded synthetic rubber seal that has a round cross-section in its free state (Fig. 1.1). O-ring can be used for the most static and dynamic conditions. It gives effective sealing through a wide range of pressures, temperatures and movements with the added advantages of sealing pressure in both directions and providing low running friction on moving parts.

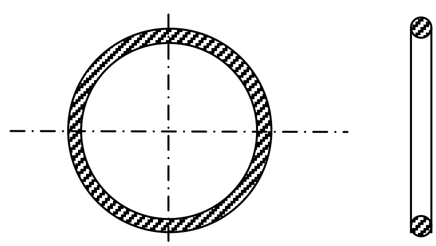


Figure 1.1 O-ring.

The action of the O-ring packing in the rectangular groove, in the V-groove and in the rectangular groove with backup washers at various pressures are shown in Fig. 1.2. It can readily be seen that without backup washers, as the pressure increases, the ring is forced or extruded into the clearance space. When the backup ring washers are used, however, this is prevented and packing life is thus extended.

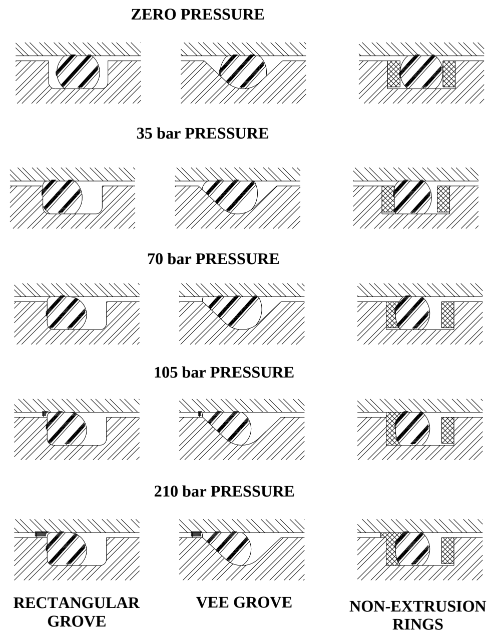


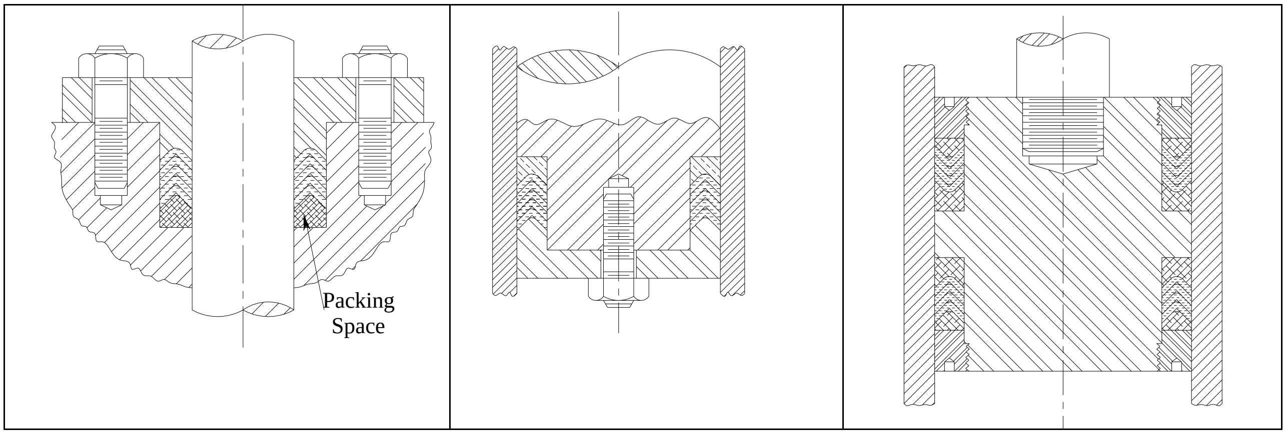
Figure 1.2 Relative position of O-ring packings in different grooves at increasing pressure.

- **V-ring seal and U-ring seal:** V- and U-ring seals are compression-type seals used in virtually in all types of reciprocating motion applications (Fig. 1.3). These include piston rods and piston seals in pneumatic and hydraulic cylinder, press rank, jacks and seals on plungers and piston in reciprocating pumps. They are also readily suited to certain slow rotary applications such as valve stems. These packings (which can be molded into U-shapes as well as V-shapes) are frequently installed in multiple quantities for more effective sealing.

(a) (b)

Figure 1.3(a)V-ring seal and (b) U-ring seal.

There are three important designs of V packings. They are shown in Fig. 1.4.Figure 1.4(a) shows an outward packed installation where the female support is an integral part of the nose on the gland ring. Figure 1.4(b) illustrates an installation of V packing on end of ram where a male ring is a part of the retaining gland ring. For inside-packed, double-acting cylinder, Fig. 1.4(c) illustrates a typical design. Here the gland rings are threaded and the male and female rings are separate units.



(a) (b) (c)

Figure 1.4 V packings:(a) An outside packed V-ring installation;(b) installation of V packing on the end of the ram;(c) V packing for double-acting cylinder.

- **T-ring seal:**T-ring seal isa dynamic seal that is extensively used to seal cylinder-pistons, piston rods and other reciprocating parts (Fig. 1.5). It is made of synthetic rubber molded in the shape of the cross-section Tandreinforced by backup rings on either side as shown in Fig. 1.5. The sealing edge is rounded and seals very much like an O-ring.

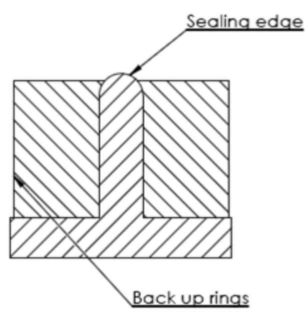


Figure 1.5T-ring seal.

- **Piston cup packings:** Piston cup packings are designed specifically for pistons in reciprocating pumps and pneumatic and hydraulic cylinders. They offer the best service life for this type of application, require a minimum recess space and minimum recess machining, and can be installed easily and quickly. In fastening the fabricated cup packing on the end of the piston, it is recommended that some provision be made to prevent the full pressure load from being taken on the bottom of the cup. The packing shown in Fig. 1.6 is single acting, while that shown in Fig. 1.7 is double acting. Piston shoulder K should be 0.127mm less than the nominal thickness of the packing. The tolerance on K is +0.00 to –0.0.762mm. Figure 1.8 shows U cup packings installed in a piston assembly. Note other seals in cylinder.

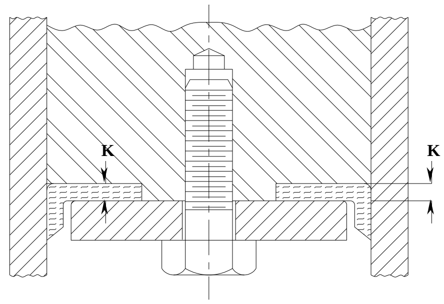


Figure 1.6 Correct method for supporting a cup packing – single acting.

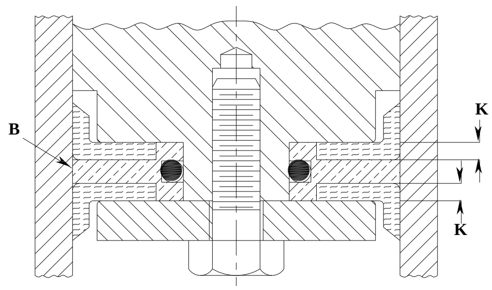


Figure 1.7 Correct method for supporting a cup packing –double acting.

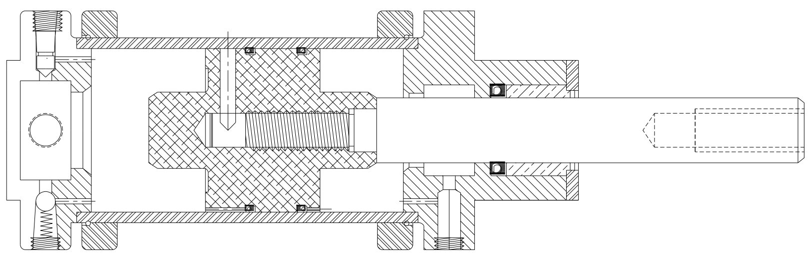


Figure 1.8 The use of U cup packings in piston assembly.

- **Piston rings:** Piston rings are seals that are universally used for cylinder pistons. Metallic piston rings are made of cast iron or steel and are usually plated or given an outer coating of materials such as zinc phosphate or manganese phosphate to prevent rusting and corrosion. Piston rings offer substantially less opposition to motion than synthetic rubber (elastomer) seals. They have a number of non-metallic piston rings made out of tetrafluoroethylene, a chemically inert, tough, waxy solid. Their extremely low coefficient of friction (0.04) permits them to be run completely dry and at the same time prevents scoring of the cylinder walls. This type of piston ring is an ideal solution to many applications where the presence of lubrication can be detrimental or even dangerous. For instance, in an oxygen compressor, just a trace of oil is a fire or explosion hazard.

- **Wiper rings and Scrapper rings:** They are seals designed to prevent foreign abrasive or corrosive materials from entering a cylinder. They are not designed to seal against pressure. They provide assurance against rod scoring and add materiality to packing life. Figure 1.9 shows a number of ways in which wiper rings can be assembled into a cylinder. The wiper ring is molded from a synthetic rubber, which is stiff enough to wipe all dust or dirt from the rod yet pliable enough to maintain a snug fit. The rings are easily installed with a snap fit into a machined groove in the gland. This eliminates the need for and expense of a separate retainer ring.Figure

1.10 shows the application of synthetic scrapper rings. It has been found that wiper rings or scrapper rings made of TFE(Tetrafluorethylene) will not adhere to the piston rod even during extended inoperative periods, and will not score piston rods made of steel or chrome plated steel. TFE is unaffected by temperature from –75℃ to –250℃ and is not affected by grease, paint, or other coatings often used for protective packaging of cylinders and valves.

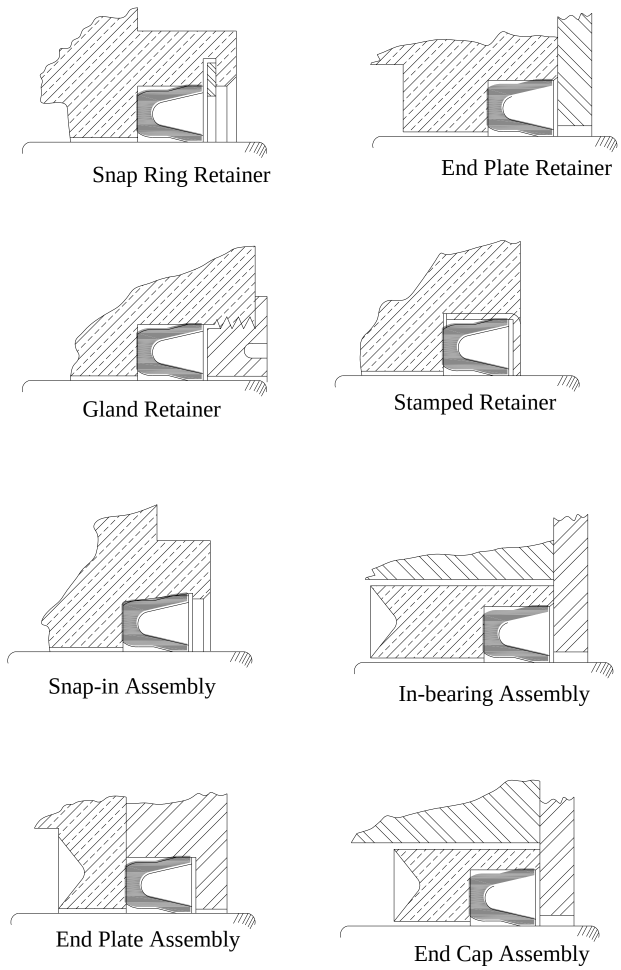


Figure 1.9 Assemblies for wiper rings.

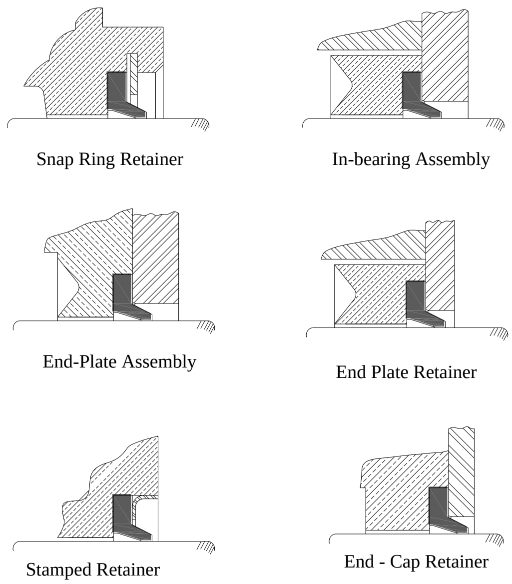


Figure 1.10 Assemblies for scraper rings.

4. **According to the type of seal material used:** Hydraulic seals may be classified according to the type of seal material used. Seals are manufactured from a variety of natural and synthetic materials. Earlier leather, cork and impregnated fibers were used in hydraulic systems. But nowadays, these are replaced by plastic and synthetic rubber materials. Synthetic rubbers (elastomers) are compatible with oil. Elastomers can be made in various compositions to meet various operating conditions:

   - Neoprene (chloroprene).
   - Buna-N.
   - Silicone (Teflon).
   - Tetrafluoroethylene.
   - Viton.

Natural rubber is rarely used as a seal material because it swells and deteriorates with time in the presence of oil. In contrast, synthetic rubber materials are compatible with most oils. The most common types of materials used for seals are leather, Buna-N, silicone, neoprene, tetrafluoroethylene, viton and of course metals.Let us discuss them below:

- **Leather:** This material is rugged and inexpensive. However, it tends to squeezewhen dry and cannot operate above 90°C which is inadequate for many hydraulic systems. Leather does not operate well at cold temperatures to about −50°C .
- **Buna-N:** This material is rugged and inexpensive and wears well. It has a rather wide operating temperature range (−45–110°C) during which it maintains its good sealing characteristics.
- **Silicone:** This elastomer has an extremely wide temperature range (−65–232°C). Hence, it is widely used for rotating shaft seals and static seals where a wide operating temperature is expected. Silicone is not used for reciprocating seal applications because it has a low tear resistance.

- **Neoprene:** This material has a temperature range of 50–120°C. It is unsuitable above 120°C because it has a tendency to vulcanize.
- **Tetrafluoroethylene:** This material is the most widely used plastic for seals of hydraulic systems. It is a tough, chemically inert, waxy solid, which can be processed only by compacting and sintering. It has an excellent resistance to chemical breakdown up to temperatures of 370°C. It also has an extremely low coefficient of friction. One major drawback is its tendency to flow under pressure, forming thin, feathery films. This tendency to flow can be greatly reduced by the use of filler materials such as graphite, metal wires, glass fibers and asbestos.
- **Viton:** This material contains about 65% fluorine. It has become almost a standard material for elastomer-type seals for use at elevated temperatures up to 240°C. Its minimum operating temperature is 28°C.

1.3 Durometer Hardness Tester

The physical properties frequently used to describe the behavior of elastomers are as follows: hardness, coefficient of friction, volume change, compression set, tensile strength, elongation modulus, tear strength, squeeze stretch, coefficient of thermal expansion and permeability. Among these physical properties, hardness is the most important because it has a direct relationship to service performance.

A durometer is an instrument used to measure the indentation hardness of rubber and rubber-like materials. As shown, the hardness scale has a range from 0 to 100. The durometer measures 100 when pressed firmly on flat glass. High durometer readings indicate a great resistance to denting and thus a hard material. A durometer hardness of 70 is the most common value.

A hardness of 80 is usually specified for rotating motion to eliminate the tendency toward side motion and bunching in the groove. The values between 50 and 60 are used for static seals on rough surfaces. Hard seal materials (values between 80 and 90) have less breakaway friction than softer materials, which have a greater tendency to deform and flow into surface irregularities. As a result, harder materials are used for dynamic seals.

1.4 Reservoirs

The functions of a fluid reservoir in a power hydraulic system are as follows:

1. To provide a chamber in which any change in the volume of fluid in a hydraulic circuit can be accommodated. When the cylinder extends, there is an increased volume of fluid in the circuit and consequently there is a decrease in the reservoir level.
2. To provide a filling point for the system.
3. To serve as a storage space for the hydraulic fluid used in the system.
4. It is used as the location where the fluid is conditioned.
5. To provide a volume of fluid which is relatively stationery to allow entrained air to separate out and heavy contaminants to settle. The reservoir is where sludge, water and metal slips settle.
6. It is a place where the entrained air picked up by the oil is allowed to escape.
7. To accomplish the dissipation of heat by its proper design and to provide a radiating and convective surface to allow the fluid to cool.

A reservoir is constructed with steel plates. The inside surface is painted with a sealer to prevent rust due to condensed moisture. At the bottom, it contains a drain plug to allow the complete drainage of the tank when required. A removable head can be provided for easy access during cleaning. A vented breather cap is also included that contains an air filtering screen. This allows the tank to breathe as the oil level changes due to system demand requirements.

A baffle plate extends lengthwise across the center of the tank. The purpose of the baffle plate is to separate the pump inlet line from the return line to prevent the same fluid from recirculating continuously within the tank. The functions of a baffle plate are as follows:

1. To permit foreign substances to settle to bottom.
2. To allow entrained air to escape from oil.
3. To prevent localized turbulence in the reservoir.
4. To promote heat dissipation through reservoir walls.

The return line should enter the reservoir on the side of the baffle plate that is opposite to the pump suction line.

1.4.1 Features of a Hydraulic Reservoir

Schematic diagram of hydraulic reservoir is shown in Fig.1.11. There are many components mounted on reservoir and each one of them having specific features.Following are the features of a hydraulic reservoir:

1. **Filler cap (breather cap):** It should be airtight when closed but may contain the air vent that filters air entering the reservoir to provide a gravity push for proper oil flow.
2. **Oil level gauge:** It shows the level of oil in the reservoir without having to open the reservoir.

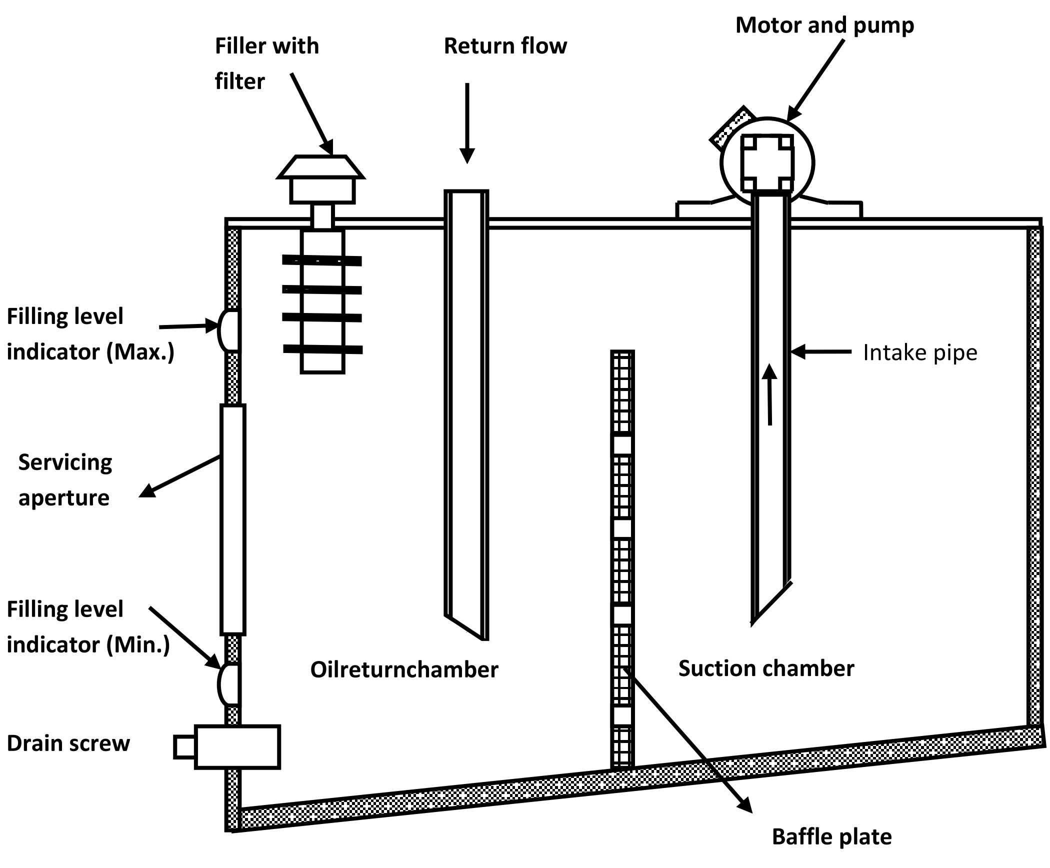


Figure 1.11 Hydraulic reservoir.

3. Baffle plate: It is located lengthwise through the center of the tank and is two-third the height of the oil level. It is used to separate the outlet to the pump from the return line. This ensures a circuitous flow instead of the same fluid being recirculated. The baffle prevents local turbulence in the tank and allows foreign material to settle, get rid of entrapped air and increase heat dissipation.

4. Suction and return lines: They are designed to enter the reservoir at points where air turbulence is the least. They can enter the reservoir at the top or at the sides, but their ends should be near the bottom of the tank. If the return line is above the oil level, the returning oil can foam and draw in air.

5. **Intake filter:** It is usuallya screen that is attached to the suction pipe to filter the hydraulic oil.
6. **Drain plug:** It allows alloil to be drained from the reservoir. Some drain plugs are magnetic to help remove metal chips from the oil.

7. Strainers and filters: Strainers and filters are designed to remove foreign particles from the hydraulic fluid. Strainers and filters are discussed in detail in Section 1.6.

1.4.2 Types of Reservoirs

Industrial reservoirs come in a variety of styles. Some of them are the following:

1. Non-pressurized: The reservoir may be vented to atmosphere using an air filter or a separating diaphragm. The type most commonly used in industry, normally, has an air breather filter, although in very dirty environments, diaphragms or air bags are used.
2. Pressurized: A pressurized reservoir usually operates between 0.35 and 1.4 bar and has to be provided with some method of pressure control; this may be a small air compressor maintaining a set charge pressure. In motor circuits where there is a little change in fluid volume in the reservoir, a simple relief valve may be used to limit the air pressure that alters with changes in temperature. The advantages of a pressurized reservoir are that it provides boost pressure to the main pump and prevents the ingress of atmospheric dirt.

1.4.3 Sizing of the Reservoir

The reservoir capacity should be adequate to cater for changes in fluid volume within the system, and with sufficient surface area to provide system cooling. An oversize reservoir can present some disadvantages such as increased cost, size and longer warming-up periods when starting from cold. There are many empirical rules for sizing reservoirs.
The sizing of a reservoir is based on the following criteria:

1. The minimum reservoir capacity should be twice the pump delivery per minute. This must be regarded as an absolute minimum and may not be sufficient to allow for the volume changes in the system.
2. The reservoir capacity should be three to four times the pump delivery per minute. This may well be too high a volume for mobile application.
3. The reservoir capacity should be 2–15 L per installed horse power. This may result in very large reservoirs when high-pressure systems are used.
4. It must make allowance for dirt and chips to settle and for air to escape.
5. It must be able to hold all the oil.

6. It must maintain the oil level high enough to prevent the whirlpool effect.
7. It should have a large surface area to dissipate heat generated in the system.
8. It should have an adequate air space to allow for the thermal expansion of oil.

All the above rules are based on conventional-shaped reservoirs. Special shapes require special considerations.When heat dissipation from the reservoir is a critical factor, it can be calculated by the following basic formula:

$$H = hA\Delta T \times 3.6 \qquad (1.1)$$

where H is the heat transfer (W), h is the heat transfer coefficient, A is the surface area in m^2 and ΔT is the temperature differential (°C).For a vertical plate of height L,

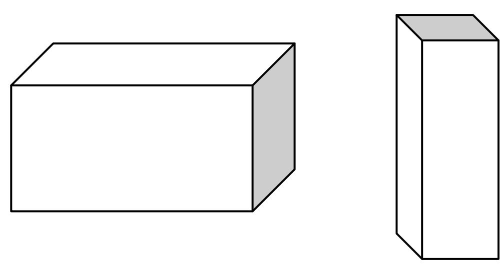
$$h = 1.42\left(\frac{\Delta T}{L}\right)^{1/4} \qquad (1.2)$$

For a horizontal plate of width W,

$$h = 1.32\left(\frac{\Delta T}{W}\right)^{1/4} \qquad (1.3)$$

These formulas apply to natural radiations. Normal air circulation round the reservoir increases the cooling considerably. For maximum heat radiations, the reservoir should be of minimum height and maximum length(see Fig. 1.12). Figure 1.12(a) shows minimum height and is best design from heat transfer point of view and figure shown in Fig. 1.12(b) has poor heat transfer.The equations neglect the cooling effect of the pipe work valves and actuators which may have a surface area comparable with that of the fluid reservoir.

The reservoir should not have any horizontal lips or angles on the vertical face as this interferes with the natural air convection. The cooling efficiency can be increased by using a finned design (Fig. 1.13); the fins should be vertical not horizontal. To assist the free circulation of external air and hence cooling, the reservoir should be mounted clear of the ground.



(a) (b)

Figure 1.12(a)Best design for heat transfer; (b) design for poor heat transfer.

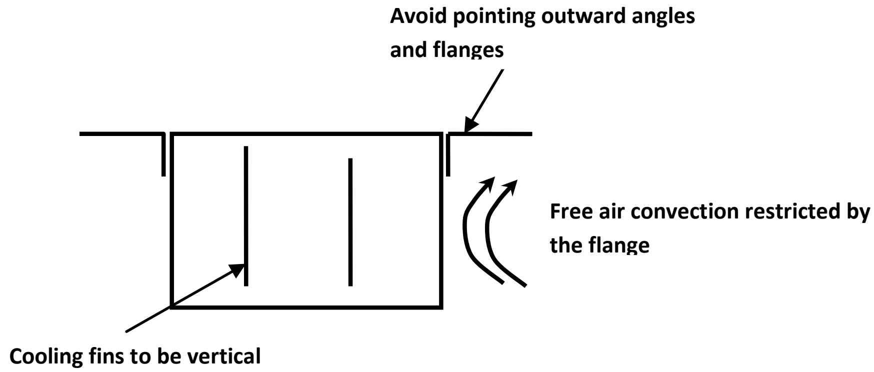


Figure 1.13 Effect of fins and flanges.

The operating temperature of a fluid greatly affects its life. Table 1.1 gives an indication of operating temperatures for various types of fluids. These are typical values only and the exact value for a specific hydraulic fluid should be ascertained from the fluid supplier.

Table 1.1 Operating temperatures for various types of fluids

Temperature	**Mineral Oil** (°C)	**Water in Oil (60/40)** (°C)	**Water Glycol** (°C)	**Phosphate Ester** (°C)
Maximum local temperature	100	65	65	150
Maximum temperature for continuous operation	65	40	40	95
Maximum temperature for optimum fluid life	40	25	25	65

1.4.4 Reservoir Design and Construction

The reservoir must be equipped with a filling point and an air vent point. These are often combined as filler breather unit complete with a filling mesh. Combined units of this nature tend to have a slow filling rate and the filling mesh is often deliberately punctured to speed up the inflow. This defeats its purpose and allows large dirt into the reservoir. It is also very easy to leave the air-filter cap off after filling thereby permitting contaminants to enter. It is better to have a separate filling point complete with a quick release coupling and integral filler unit, in which case all the fluid is pumped into the reservoir through the filter, spin-on air breather are available with a fine filter range: 25 μm absolute is typical (β_{25} = 75).

The pump suction line should be fitted with an adequately sized strainer. The strainer should be located so that there is a well distributed flow and its lowest point should be at least 75 mm above the bottom of the reservoir (to prevent any debris on the bottom of the tank being pulled into the suction line). The top of the strainer should be at least 150 mm below the lowest fluid level to prevent a vortex being formed and air drawn into the suction line.

System return lines and drain lines should terminate below the minimum fluid level to prevent aeration of the fluid. The main returns should be fitted with diffusers that reduce the return oil velocity and prevent turbulence. Where diffusers are not used, the end of return lines should be cut at 45%. Return outlets

should be carefully positioned to minimize interaction between the return flow and the reservoir boundaries. If the return pipes have an anti-syphon hole, aeration may be caused by the jet of fluid emanating from them during normal operation. This can be reduced by fitting deflector plates.
The reservoir must be fitted with some form of level indicator, generally a sight glass that should have the minimum and maximum levels clearly marked. A float switch can be incorporated to give an alarm or shut the pump down if the oil level falls below a certain value. This is very important in large systems where there is a possibility of a pipe failure or other sudden leak. Shutting the pump down protects the pump from cavitation and reduces the spillage of fluid.
Baffles are fitted in the reservoir to separate the return lines from the suction lines. They create a long flow path, which reduces surging effects, encourages dirt separation and improves cooling. A fine wire mesh baffle between 60 and 200 mesh size (250–75 μm aperture), set at an angle of 20–40°to horizontal, with the top edge terminating below the fluid surface will considerably aid the separation of entrained air. Wherever the hydraulic fluid is an oil and water emulsion, baffles should not be used as they tend to separate emulsion. With these fluids, good circulation in the reservoir is important to maintain emulsion. Figure 1.14 shows a reservoir designed to include most of these features. Although it is not the best shape for heat dissipation, it has other merits. If the fluid temperature inside the reservoir varies outside specified limits, temperature switches can be used to give a warning, switch in coolers or heaters, shut down pumps, prevent startup, etc. Wherever systems operate outdoors in cold temperatures or, for example, servo systems, which in order to preserve accuracy need to work at a constant temperature, heaters may be incorporated. These are a special type of immersion heaters having a low wattage relative to the surface area (maximum 6 W/m^2).

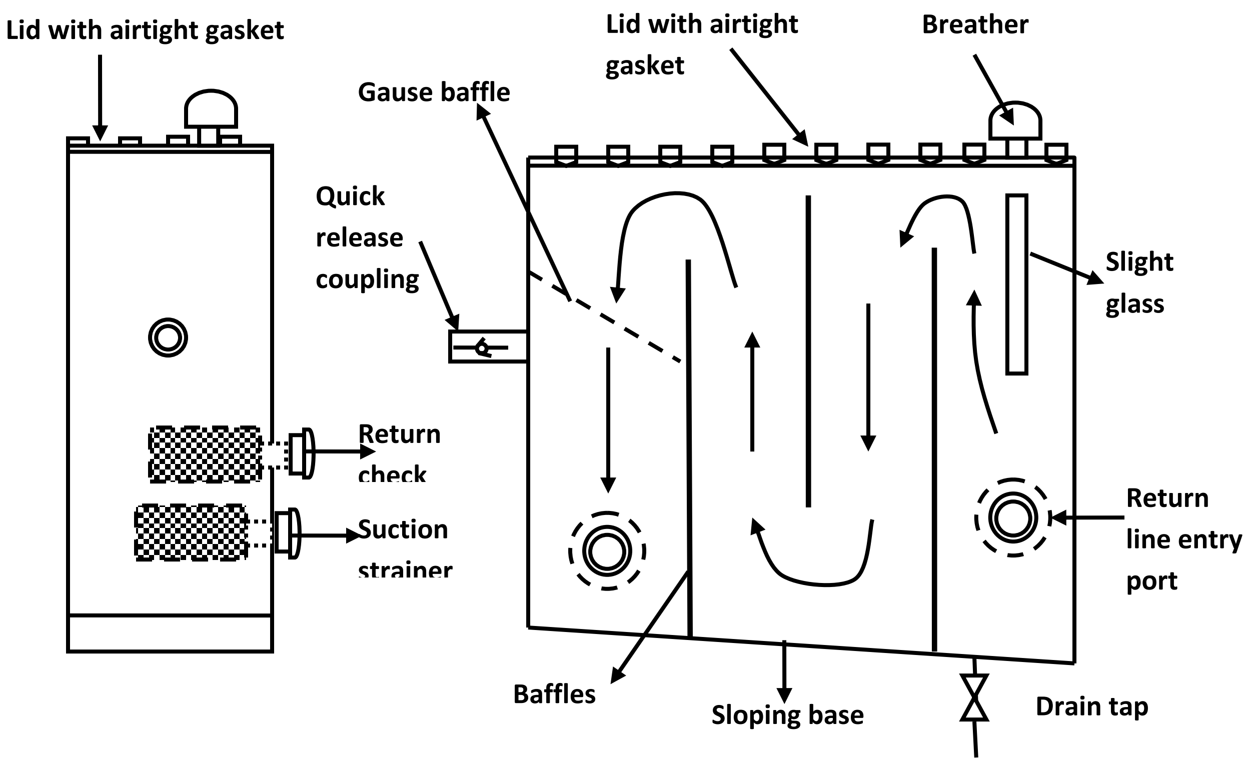


Figure 1.14 Reservoir design and construction.

1.5 Fluid Conditioners

In the case of large reservoirs and systems where online filtration is impossible, separate clean-up filtration loops should be fitted. These may include an oil cooler. There are two types of oil coolers in use:

1. Water tube coolers: Water tube oil coolers consist of a series of interconnected copper tubes through which the cooling water passes surrounded by a jacket through which the hydraulic fluid passes. It is quiet in operation and can be arranged so that the oil pressure is higher than the water pressure; consequently, any leakage is more likely to be of oil into the water that is less serious than the contamination of the hydraulic fluid. When a water tube cooler is used, there is considerable flow of water involved and a separate water cooling tower and a circulating water supply may be necessary. Usually, the water supply is thermostatically controlled so that it is only switched ON when required.
2. Air blast coolers: An air blast cooler is similar in construction to a vehicle radiator with a powered air fan. It should be situated in a cool area so that cold air is blown over the radiator. An air blast cooler tends to be noisy but on small installations is preferable to water coolers owing to running and installation costs. Air blast coolers are now available to fit between the pump and the electric motor as part of the bell housing and coupling. Such coolers do not need a separate electric motor drive, but it is necessary to take into account the extra power needed when sizing the pump drive.

1.5.1 Centralized Hydraulic System

The forerunner of modern gas, electricity and water authorities were a large number of local and municipal undertakings meeting the demands of large towns and industrially customers. In the late 19th and early 20th century, there existed several public hydraulic power companies providing water power in a similar manner.
The earliest system was in dockyards driving machinery of cranes, dock gates and sluices, commencing with an installation at hull docks in 1876. During that time, there were sizable undertakings in most parts of the world.
The change to hydraulic oil as a working medium fluid made such large installation impractical, but there has recently been a renaissance of centralized systems operating on a smaller scale. The idea of a hydraulic ring main running around a factory in a manner similar to the pneumatic ring mains which are in everyday use has many attractions. The principal difference, apart from the higher pressures involved and the obvious seriousness of leaks, is that a return pipe has to be provided as well as supply. Their most frequent use is in a factory or process plant where a group of related machines is powered from a central source.

1.5.2 Individual versus Centralized Systems

There is still much controversy over the use of centralized hydraulic systems as opposed to individual power packs. The features of both are being compared. In general, what is an advantage for one is a disadvantage for the other. Both have their merits and every application must be assessed on its own particular requirement.

1.5.2.1 Individual Power-Pack Systems

Individual power packs are provided for each machine and components mounted are specific to the operation. They are usually supplied as modular units.

The advantages of individual power-pack systems are as follows:

1. They are completely independent of each other.
2. Different grades or types of fluids can be used as appropriate to each system.
3. Each can operate at different pressures.
4. If one circuit fails, others are still operative.
5. Power packs can be adjacent to the machine.

The disadvantages of individual power-pack systems are as follows:

1. More power packs and components to maintain.
2. Increased cost.
3. More floor space required.
4. Greater total power.

1.5.2.2 Centralized Hydraulic Systems

In centralized hydraulic system, reservoir will cater to many machines. This is like a hydraulic ring which runs around a factory in a manner similar to pneumatic ring.

The advantages of centralized hydraulic systems are as follows:

- Single oil reservoir.
- Standby pumps can be designed into the power pack.
- Reduced cost.
- Single unit to be maintained.
- Less space required.
- Fluid conditioning can be incorporated either offline or in the return line at much lower cost than when individual power packs are used.
- Total power-pack capacity may be less than that for individual power packs.

The disadvantages of centralized hydraulic systems are as follows:

- If there is a failure in the power pack, all the systems fail (unless standby facility included).
- Longer pipe runs are involved than with individual power packs.
- If more than one operating pressure is involved, pressure-reducing valves have to be fitted to the appropriate circuits.
- A large volume of fluid in the reservoir can be a considerable fire hazard.
- There may be interaction between the circuits.

Lecture 31

ACCESSORIES USED IN FLUID POWER SYSTEMS [CONTINUED]

1.6 Filters and Strainers

For proper operation and long service life of a hydraulic system, oil cleanliness is of prime importance. Hydraulic components are very sensitive to contamination. The cause of majority of hydraulic system failures can be traced back to contamination. Hence, filtration of oil leads to proper operation and long service life of a hydraulic system.

Strainers and filters are designed to remove foreign particles from the hydraulic fluid. They can be differentiated by the following definitions:

1. Filters: They are devices whose primary function is the retention, by some fine porous medium, of insoluble contaminants from fluid. Filters are used to pick up smaller contaminant particles because they are able to accumulate them better than a strainer. Generally, a filter consists of fabricated steel housing with an inlet and an outlet. The filter elements are held in position by springs or other retaining devices. Because the filter element is not capable of being cleaned, that is, when the filter becomes dirty, it is discarded and replaced by a new one. Particle sizes removed by filters are measured in microns. The smallest sized particle that can be removed is as small as 1 µm. A strainer is a device whose function is to remove large particles from a fluid using a wire screen. The smallest sized particle that can be removed by a strainer is as small as 0.15 mm or 150 µm.

2. Hydraulic strainers: A strainer is a coarse filter. Fluid flows more or less straight through it. A strainer is constructed of a fine wire mesh screen or of screening consisting of a specially processed wire of varying thickness wrapped around metal frames. It does not provide as fine a screening action as filters do, but offers less resistance to flow and is used in pump suction lines where pressure drop must be kept to a minimum. A strainer should be as large as possible or wherever this is not practical, two or more may be used in parallel.

1.6.1 Causes of Contamination

The causes of contamination are as follows:

- Contaminants left in the system during assembly or subsequent maintenance work.
- Contaminants generated when running the system such as wear particles, sludge and varnish due to fluid oxidation and rust and water due to condensation.
- Contaminants introduced into the system from outside. These include using the wrong fluid when topping up and dirt particles introduced by contaminated tools or repaired components.

1.6.2 Types of Filters

Filters may be classified as follows:

1. **According to the filtering methods:**

- **Mechanical filters:** This type normally contains a metal or cloth screen or a series of metal disks separated by thin spacers. Mechanical filters are capable of removing only relatively coarse particles from the fluid.
- **Absorption filters:** These filters are porous and permeable materials such as paper, wood pulp, diatomaceous earth, cloth, cellulose and asbestos. Paper filters are impregnated with a resin to provide added strength. In this type of filters, the particles are actually absorbed as the fluid permeates the material. Hence, these filters are used for extremely small particle filtration.

- **Adsorbent filters:** Adsorption is a surface phenomenon and refers to the tendency of particles to cling to the surface of the filters. Thus, the capacity of such a filter depends on the amount of surface area available. Adsorbent materials used include activated clay and chemically treated paper.

2. According to the size of pores in the material:

- **Surface filters:** These are nothing but simple screens used to clean oil passing through their pores. The screen thickness is very thin and dirty unwanted particles are collected at the top surface of the screen when the oil passes, for example, strainer.
- **Depth filters:** These contain a thick-walled filter medium through which the oil is made to flow and the undesirable foreign particles are retained. Much finer particles are arrested and the capacity is much higher than surface filters.

3. According to the location of filters:

- **Intake or inline filters (suction strainers):** These are provided first before the pump to protect the pump against contaminations in the oil as shown in Fig. 1.15. These filters are designed to give a low pressure drop,otherwise the pump will not be able to draw the fluid from the tank. To achieve low pressure drop across the filters, a coarse mesh is used. These filters cannot filter out small particles.

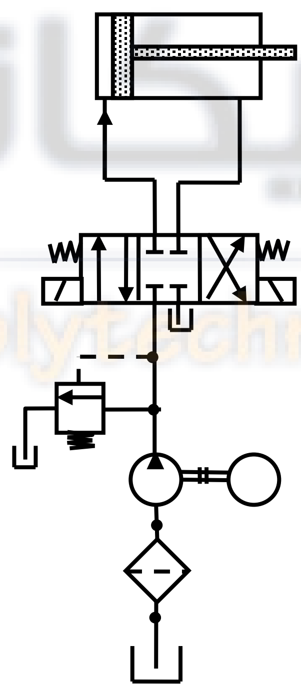


Figure 1.15 Suction filter**.**

***Advantages of suction filters**:*

(a) A suction filter protects the pump from dirt in the reservoir. Because the suction filter is outside the reservoir, an indicator telling when the filter element is dirty can be used.
(b) The filter element can be serviced without dismantling the suction line or reservoir (easy to maintain).

Disadvantages of suction filters:

(a) A suction filter may starve the pump if not sized properly.

- **Pressure line filters (high-pressure filters):** These are placed immediately after the pump to protect valves and actuators and can be a finer and smaller mesh (Fig. 1.16). They should be able to withstand the full system pressure. Most filters are pressure line filters.

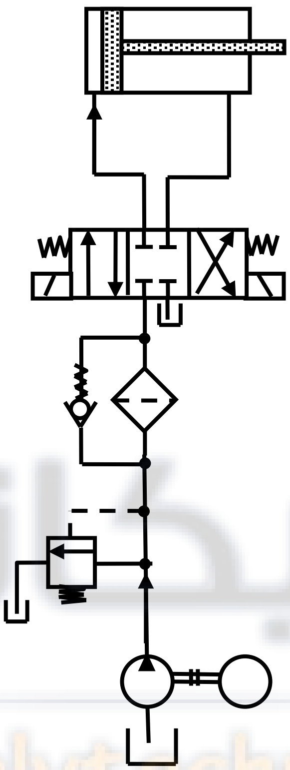


Figure 1.16 Pressure filter**.**

Advantages of a pressure line filter:

(a) A pressure filter can filter very fine contaminants because the system pressure is available to push the fluid through the element.
(b) A pressure filter can protect a specific component from the harm of deteriorating particles generated from an upstream component.

Disadvantages of a pressure line filter:

(a) The housing of a pressure filter must be designed for high pressure because it operates at full system pressure. This makes the filter expensive.
(b) If pressure differential and fluid velocity are high enough, dirt can be pushed through the element or the element may tear or collapse.

- **Return line filters (low-pressure filters):** These filters filter the oil returning from the pressure-relief valve or from the system, that is, the actuator to the tank (Fig. 1.17). They are generally placed just before the tank. They may have a relatively high pressure drop and hence can be a fine mesh. These filters have to withstand low pressure only and also protect the tank and pump from contamination.

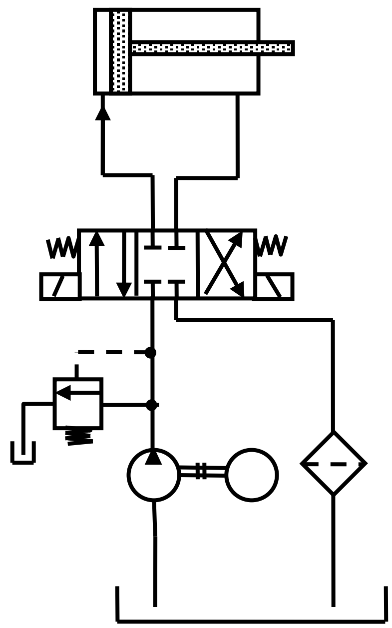


Figure 1.17 Return line filter.

Advantages of a return line filter:

(a)A return line filter catches the dirt in the system before it enters the reservoir.
(b)The filter housing does not operate under full system pressure and is therefore less expensive than a pressure filter.

Disadvantages of a return line filter:

(a) There is no direct protection for circuit components.
(b) In return line full flow filters, flow surges from discharging cylinders, actuators and accumulators must be considered when sizing.

1. **Depending on the amount of oil filtered by a filter:**

- **Full flow filters:** In this type, complete oil is filtered. Full flow of oil must enter the filter element at its inlet and must be expelled through the outlet after crossing the filter element fully(Fig. 1.18). This is an efficient filter. However,it incurs large pressure drops. This pressure drop increases as the filter gets blocked by contamination.

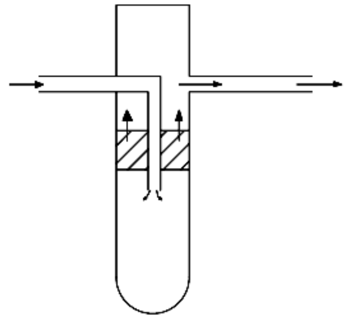


Figure 1.18 Full flow filter.

- **Proportional filters (bypass filters):** In some hydraulic system applications, only a portion of oil is passed through the filter instead of entire volume and the main flow is directly passed without filtration through a restricted passage (Fig. 1.19).

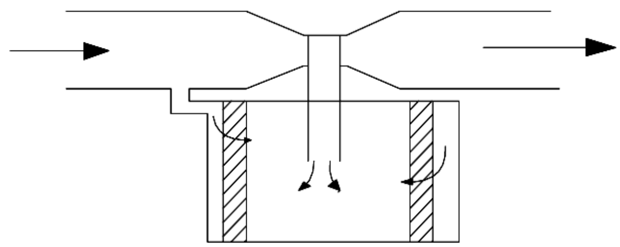


Figure 1.19 Proportional filter.

1.6.3 Beta Ratio of Filters

Filters are rated according to the smallest size of particles they can trap. Filter ratings are identified by nominal and absolute values in micrometers. A filter with a nominal rating of 10 μm is supposed to trap up to 95% of the entering particles greater than 10 μm in size. The absolute rating represents the size of the largest pore or opening in the filter and thus indicates the largest size particle that could go through. Hence, absolute rating of a 10 μm nominal size filter would be greater than 10 μm.

A better parameter for establishing how well a filter traps particles is called the beta ratio or beta rating. The beta ratio is determined during laboratory testing of a filter receiving a steady-state flow containing a fine dust of selected particle size. The test begins with a clean filter and ends when pressure drop across the filter reaches a specified value indicating that the filter has reached the saturation point. This occurs when contaminant capacity has been reached.

By mathematical definition, the beta ratio equals the number of upstream particles of size greater than Nμm divided by the number of downstream particles having size greater than Nμm where N is the selected particle size for the given filter. The ratio is represented by the following equation:

$$\text{Beta ratio} = \frac{\text{No. of upstream particles of size} > N\ \mu\text{m}}{\text{No. of downstream particles of size} > N\ \mu\text{m}} \quad (1.4)$$

A beta ratio of 1 would mean that no particle above specified N are trapped by the filter. A beta ratio of 50 means that 50 particles are trapped for every one that gets through. Most filters have a beta ratio greater than 75:

$$\text{Beta efficiency} = \frac{\text{No. of upstream particles} - \text{No. of downstream particles}}{\text{No. of upstream particles}}$$

Thus,

$$\text{Beta efficiency} = 1 - \frac{1}{\text{Beta ratio}} \qquad (1.5)$$

1.7 Heat Exchangers

The heating up of hydraulic oil beyond tolerable limits in an otherwise well-designed hydraulic systemis usually a phenomenon associated with high-pressure, high-flow systems cycling at high frequencies. The input power in such systems is usually more than 40 kW.

Heat is generated in hydraulic systems because no component can operate at 100% efficiency. Significant sources of heat include the pump, pressure-relief valves and flow control valves. Heat can cause the hydraulic fluid temperature to exceed its normal operating range of 35–70°C. Excessive temperature hastens the oxidation of the hydraulic oil and causes it to become too thin. This promotes deterioration of seals and packing and accelerates wear between closely fitting parts of hydraulic components of valves, pumps and actuators.

The steady-state temperature of fluid of a hydraulic system depends on the heat-generation rate and the heat-dissipation rate of the system. If the fluid operating temperature in a hydraulic system becomes excessive, it means that the heat-generation rate is too large relative to the heat-dissipation rate. Assuming that the system is reasonably efficient, the solution is to increase the heat-dissipation rate. This is accomplished by the use of coolers, which are commonly called "heat exchangers."
It is basically a problem associated with fluid power systems using constant delivery pumps in which, for most part of the cycle, the fluid is dumped to the tank through the relief valves at high pressures resulting in wasted power. The consequent heating of oil should not be a cause for concern if oil temperatures do not exceed 30°C above ambient even after 8 h of continuous operation.

Injection molding machines and huge hydraulic presses are typical examples for high-pressure, high-flow systems. If the frequency of the operating cycle is also high, then heat builds up because the high operating cycle does not permit heat dissipation. The consequent temperature raise shall be between 40 and 60°C above ambient. In places where the ambient temperatures are above 30°C, this heating up of oil would be a cause for concern. Electrical valves begin to malfunction and leakage past the seals and spool valves due to viscosity changes would affect the system performance. A poorly designed hydraulic system is another cause for heating up of oil. The glow ratings of all valves and conductors must be adequate to handle the pump flow. In a double-acting cylinder with a large piston area to annular area ratio, valves must be selected to handle the discharge out of the rod end of the cylinder and not based on the flow into the cap end. Heat exchangers are designed primarily to overcome this problem. If installed at an appropriate location, which is the tank return line, heat exchangers help the system to dissipate heat and maintain oil temperatures within normal limits.

In some applications, the fluid must be heated to produce a satisfactory value of viscosity. This is typical when, for example, mobile hydraulic equipment is to operate below 0°C. In these cases, the heat exchangers are called "heaters." However, for most hydraulic systems, the natural heat-generation rate is sufficient to produce high enough temperatures after an initial warm-up period.

Basically, there are two types of heat exchangers: liquid-to-liquid and liquid-to-air type. Liquid-to-liquid types are of shell and tube construction consisting of a bundle of small tubes held inside a shell. The coolant flows through the small tubes, while the hydraulic oil passes around and between these tubes. The tubes can be either of the straight-type or of U-type. Straight-type units have higher thermal efficiency and are less expensive. U-types are best suited for high-temperature and high-pressure applications. They can accommodate thermal expansion but the tube banks are difficult to clean. The heat exchangers can be of single pass or double pass and can be of parallel-flow or counter-flow type. The double-pass counter-flow type provides maximum heat transfer for a given size.

For hydraulic applications, the commonly used coolant is water. Standard liquid-to-liquid units can handle pressures up to 15 bar and temperatures up to 150°C although the actual temperature difference between oil and water should not exceed 90°C. The mechanism of action of liquid-to-liquid heat exchangers is one of evaporation and condensation. The tubes are basically an enclosure containing a fluid that can be vaporized and a material that provides capillary action. In what is basically an isothermal process, heat at the input end causes the fluid to evaporate. Vapor travels through the tube to the condenser or output end. Here the vapor condenses giving up its latent heat and returns to the fluid state and travels back to the system through the evaporator end. Clean and soft water should be used to prevent corrosion and scaling in the tubes. If the water is hard and has excessive salt, it can lead to clogging of the insides of the narrow tube due to scale and dirt deposits. Liquid-to-air heat exchangers transfer heat from the hydraulic fluid to the atmosphere, just like an automobile radiator. The air passes over-finned tubes made of either copper or aluminum in which the hot hydraulic fluid circulates.

Heat exchangers are available as of the shelf components in various sizes and configurations. They are available with temperature control valves, water strainers and bypass check valves. Bypass valves protect heat exchangers by diverting the excess oil to the tank during surge and peak pressures.

1.7.1 Sizing of Heat Exchangers

The selection of a suitable heat exchanger involves determining the total heat equivalent to wasted energy or the heat load. This can be assessed by analyzing the duty cycle of the hydraulic machine. The total time period of a cycle must be broken up into its phases consisting of idling, approach, work, return, etc. and the wasted energy calculated for each.

The sum of the energy less the energy dissipated through the reservoirs and flow lines by convection and radiation constitutes the net energy that the heat exchanger must dissipate or the rate of heat transfer. The maximum oil temperature at the inlet point to the heat exchanger can now be determined by using the equation

$$H = mC_{\mathrm{p}}(T_1 - T_0) \tag{1.6}$$

Here H is the heat transfer rate in kJ/s;C_{p} is the specific heat at constant pressure that for hydraulic oil = 1.97 kJ/kg°K;T_1 is the oil temperature at the inlet to the heat exchanger, which typically lies between 55 and 65°C;T_0 is the oil temperature at the outlet, which is the desired oil temperature that the heat exchanger must maintain. A suitable value may be assumed forT_0which is consistent with the system requirement. Also m is the mass flow rate determined by the equation

$$m = \rho \times q \ \ (\mathrm{kg/s}) \tag{1.7}$$

where ρ is the density of hydraulic oil in kg/m^3 and q is the oil flow rate in m^3/s. If the difference between T_1 and T_0 is large, then it justifies the need for a heat exchanger. If the difference is only marginal, the heat exchanger may be dispensed with. By sizing the surface area of the reservoir, so that it can transfer enough heat from oil to the atmosphere by conduction, convection or radiation, the net wasted energy that the heat exchanger must dissipate can be reduced, thus reducing the size of heat exchanger.
After ascertaining the need for the heat exchanger, the size of heat exchanger in terms of quantity of water required to dissipate the heat and maintain the desired temperature must be determined based on the equation

$$Q = \frac{(860)(P_w)}{T_1 - T_w} \qquad (1.8)$$

where P_w is the wasted energy in kW, Q is the water flow rate through the heat exchanger in L/min, T_1 is the temperature of the oil and T_w is the water temperature. When the flow capacity as determined from the equation is much larger than the capacity of available heat exchangers, then two smaller heat exchangers of equal sizes can be used in parallel. Standard heat exchangers would have the oil-to-water ratio of either 1:1 or 2:1.This means that for every liter of oil circulated, one or half a liter of water must be circulated to achieve the desired oil temperature. The size and therefore the cost of a heat exchanger depend largely on the initial temperature difference between the inlet oil and water. If this difference decreases, the cost decreases. The pressure drop that the hydraulic system can permit is another factor that has a direct bearing on the size and cost of the heat exchanger. Ideal allowable pressure drops should be between 3.5 and 5 bar.

If it becomes necessary to install a heat exchanger in an existing system, then the waste energy may safely be assumed equal to 20% of the connected power. The value of T_1 may be obtained directly by measuring the maximum temperature of the oil in the tank and heat exchanger size, based on Eq. (1.8) above. Shell and tube heat exchangers can be mounted either vertically or horizontally.

Example 1.1
In a hydraulic system operating at 200 bar, the pump delivery is 25 LPM and input power to the pump drive is 10 kW. The system cycle is such that the pump is unloaded at 60% of the operating time. The overall efficiency of the systems when it is on load is 65%. If the ambient temperature is 15°C and the maximum permissible fluid temperature in the reservoir is 50°C, calculate a suitable size for the fluid reservoir assuming that
(a) Normal air circulation around the fluid reservoir doubles the cooling owing to the natural radiation.
(b) The fluid reservoir is of square section of side *a* with a length of 2*a*.

Solution: Heat dissipation from the vertical plate:

$$H_v = h_v A \Delta T \times 3.6$$

where ΔT = 50 – 15 = 35°C and $A = 6a^2$. Also $h_v = 1.42\left(\frac{35}{a}\right)^{1/4} = 3.45\,(a)^{-1/4}$

So

$$H_v = 3.45\,(a)^{-1/4}\, 6a^2 \times 35 \times 3.6\,(\text{W}) = 2608\ (a)^{7/4}\ (\text{W})$$

Heat dissipation from the horizontal top plate:

$$H_H = h_H A \Delta T \times 3.6$$

where ΔT = 50 – 15 = 35°C and $A = 6a^2$. Also

$$h_H = 1.32\left(\frac{35}{a}\right)^{1/4} = 3.21\,(a)^{-1/4}$$

So

$$H_H = 3.21\,(a)^{-1/4}\, 2a^2\, 35 \times 3.6 = 809\ (a)^{7/4}\ \text{W}$$

Heat dissipation owing to the natural radiation:

$$H_v + H_H = 2608\ (a)^{7/4} + 809\ (a)^{7/4} = 3417\,(a)^{7/4}$$

Heat dissipation with normal air circulation may be taken as twice that owing to natural radiation and is equal to $6834\,(a)^{7/4}$ W.
Heat input to the system during "on load" part of cycle is

$$10 \times (1 - 0.65) = 3.5 \text{ kW}$$

But the system is only on load for 40% of the time; thus the average heat energy input to the fluid is given by

$$\text{Average input} = 0.4 \times 3.5 = 1.4 \text{ kW} = 1400 \text{ W}$$

For thermal equilibrium, the heat energy entering the system must be equal to the heat energy dissipated from the system; therefore,

$$6834\,(a)^{7/4} = 1400$$

Solving, we get a = 0.404 m = 40.4 cm.
Assume the tank material measurement to be 0.4 m wide, 0.4 m high and 0.8 m long; then the volume of the fluid contained is given by

$$Q = 0.4 \times 0.4 \times 0.8 = 0.128 \text{ m}^3 = 128 \text{ L}$$

In practice, the tank has to be higher than 0.4 m as there must be clearance volume above the oil. Size the reservoir from the rule of thumb formula, that is, the reservoir capacity is equal to three to four times the pump delivery per minute.

$$\text{Reservoir capacity} = -75 \text{ to } 100 \text{ L}$$

This value is similar to the value calculated.
If a very large fluid reservoir is needed to dissipate the heat energy, it may be advantageous to use a fluid cooler.

Example 1.2
Determine the beta ratio of a filter when, during test operation, 30000 particles greater than 20 μm enter the filter and 1050 of these particles pass through the filter. What is the beta efficiency?

Solution: We have

$$\text{Beta ratio} = \frac{\text{No. of upstream particles of size} > N\,\mu\text{m}}{\text{No. of downstream paricles of size} > N\,\mu\text{m}} = \frac{30000}{1050} = 28.6$$

$$\text{Beta efficiency} = \frac{\text{No. of upstream particles} - \text{No. of downstream particles}}{\text{No. of upstream particles}} = \frac{30000 - 1050}{30000} = 96.5\%$$

Beta efficiency could also be calculated as

$$\text{Beta efficiency} = 1 - \frac{1}{\text{Beta ratio}} = 1 - \frac{1}{28.6} = 96.5\%$$

Example 1.3
A hydraulic machine has the following duty cycle: Idle at 15 bar for 2 s, clamp workpiece at 100 bar for 5 s, approach at 15 bar for 2 s, perform work at 300 bar for 3 s, declamp, return at 15 bar for 2 s.The pump flow is 100 LPM, the total surface area of the oil reservoir is 2.5 m^2 and the hydraulic pipeline is 25 mm × 2500 mm. Calculate the net wasted energy that needs to be dissipated and recommend a suitable heat exchanger if necessary. Assume the following values: Ambient temperature = 20°C, volumetric efficiency of the pump = 0.85, density of oil = 860 kg/m^3, 1 kWh = 3.6 × 10^6 J.

Solution: Power dissipated in kW is given by the equation

$$\frac{pQ}{600} \times \eta$$

where$p$ is the pressure in bar, Q is the flow in LPM and η is the volumetric efficiency of the pump.
Power dissipated during the idle cycle while passing oil through valves and pipelines is

$$15 \times \frac{100}{600} \times 0.85 = 2.94 \text{ kW}$$

Because this phase lasts for 30 s, the energy consumed is

$$2.94 \times \frac{30}{3600} 0.0245 \text{ kWh or } 0.0245\,(3.6\times10^6) = 8.82 \text{ J}$$

Power dissipated during clamping is $100 \times \frac{100}{600} \times 0.85 = 19.6$ kW

Because this phase lasts for 5 s, the energy consumed is

$$19.6 \times \frac{15}{3600} = 0.027 \text{ kWh}$$
$$= 0.027 \times (3.6 \times 10^6)$$
$$= 9.72\times10^4 \text{ J}$$

Power dissipated during approach is $100 \times \frac{15}{600} \times 0.85 = 2.94$ kW

Because this phase lasts for 2 s, the energy consumed is

$$2.94 \times \frac{2}{3600} = 0.001 \text{ kWh}$$
$$=0.001 \times (3.6 \times 10^6)$$
$$= 3.6 \times 10^3 \text{ J}$$

Power dissipated during work phase is

$$100 \times \frac{300}{600} \times 0.85 = 58.8 \text{ kW}$$

Because this phase lasts for 3 s, the energy consumed is

$$58.8 \times \frac{3}{3600} = 0.049 \text{ kWh}$$
$$= 0.049 \times (3.6\times10^6)$$
$$= 17.64 \times 10^4 \text{ J}$$

Power dissipated during return is

$$100 \times \frac{15}{600} \times 0.85 = 2.94 \text{ kW}$$

Because this phase lasts for 2 s, the energy consumed is

$$2.94 \times \frac{2}{3600} = 0.001 \text{ kWh}$$
$$= 0.001 \times (3.6 \times 10^6)$$
$$= 3.6 \times 10^3 \text{ J}$$

Total power dissipated in 42 s = 2.1×10^5 J.In terms of average wasted power, this will be

$$2.1 \times \frac{10^5}{3.6\times10^6} = 0.058 \text{ kWh}$$
$$= 0.058 \times \frac{3600}{42}$$
$$= 5 \text{ kW.}$$

Available hydraulic power is

$$300 \times \frac{30}{3600} \times 0.85 = 58.8 \text{ kW}$$

Wasted power is 8.5% of available power.
Now

$$(T_1 - T_0) = \frac{H}{m} \times C_p \qquad (1.9)$$

We have

$$H = 2.1 \times \frac{10^5}{1000 \times 42} = 5 \text{ kJ/s}$$

$$m = 860 \times 0.001 = 0.86 \text{ kg/s}$$

Substituting the above values in Eq. (1.9), we get

$$(T_1 - T_0) = \frac{5}{0.86} \times 1.97 = 2.95$$

Because this difference is negligible, there is no need for a heat exchanger. Heat dissipated by the oil reservoir and hydraulic pipelines can be calculated as follows: Heat dissipated by the oil reservoir

$$\Delta T \times A \times \frac{K}{3600} \text{ kW}$$

where ΔT is the difference between the maximum oil temperature and the ambient temperature in °C, A is the surface area of the reservoir in m^2 and K the average rate of convection from the reservoir is 6–10 kJ/cm^2 h, depending upon the condition of the surroundings. For poor air circulation, assume a value equal to 6. The surface area of hydraulic pipelines is given by *pdl*, where *d* is the outside diameter of the pipe and *l* is the length in appropriate units.

Example 1.4

Oil at 49°C and 69 bar is flowing through a pressure-relief valve at 38 LPM. What is the downstream oil temperature?

Solution: First we calculate the power lost/wasted:

Power lost = p (bar) × Q (LPM)

$= (69 \times 10^5 \text{ N/m}^2)(38 \times 10^{-3} \text{ m}^3\text{/min}) \times 1/60 \text{ (s)}$

$= 4370 \text{ W} = 4.37 \text{ kW}$

Next we calculate the oil flow rate in units of kg/s and the temperature increase.

Oil flow rate (kg/s) = 895 × Oil flow rate (m^3/s)

$$= 895 \times 38 \times \frac{10^{-3}}{60} = 0.6 \text{ kg/s}$$

Now

$$\text{Temperature } (^\circ\text{C}) = \frac{\text{Heat generation rate (kW)}}{\text{Oil specific heat}\left(\frac{\text{kJ}}{\text{kg}}{}^\circ\text{C}\right) \times \text{Oil flow rate (kg/s)}}$$

$$= \frac{4.37}{1.8 \times 0.6} = 4^\circ\text{C}$$

Downstream oil temperature = 49 + 4 = 53 °C

Example 1.5

The hydrostatic transmission receives 6 kW from the engine at the pump. The transmission output delivers 3.7 kW to the rear wheel output drive. Assuming that the difference is lost to heat, compute the total heat loss over the period of 4 h of operation.

Solution: , Energy loss is given by the difference between the energy supplied(to pump) and energy taken out(at the actuator), mathematically, we get

Heat energy loss = Operation time (Pump input power – Actuator output power)

$$= 4 \text{ h } (6 - 3.7) = 9.2 \text{ kWh}$$

The temperature increase resulting from the heat generated in the fluid computed in °C/kg of fluid is attributed to discharging fluid across such a component as pressure-relief valves and from the mechanical friction of moving parts. In most hydraulic systems, mechanical friction accounts for less than 20% of the input power.

The formula for computing temperature increase due to heat generation by a fluid passing through restriction is

$$\text{Temperature increase (°C)} = \frac{\text{Heat generated (kW)}}{\text{Oil specific weight}\left(\dfrac{\text{kJ}}{\text{kg}}\text{°C}\right) \times \text{Oil flow rate (kg/s)}}$$

where the specific heat of oil is given as 1.8 kJ/kg°C. . When the flow rate is given in m^3/s and the mass density of the oil is taken as 895 kg/m^3, the mass flow rate of oil is computed from

$$\text{Mass flow rate (kg/s)} = 895 \text{ kg/m}^3 \times \text{ Oil flow rate } (\text{m}^3/\text{s})$$

Example 1.6

The pressure drop across a sticking control valve is observed to be 68.9 bar. If the fluid has a specific gravity of 0.895 and a flow rate of 0.19 LPS, estimate the rise in temperature of the fluid that can be attributed to the control valve.

Solution: The heat generated by the pressure drop across the control valve is

$$\text{Heat loss generated} = 68.9 \times 10^5 \times 0.19 \times 10^{-3} = 1309 \text{ W/s}$$

The mass flow rate of fluid through the valve is computed as

$$\text{Mass flow rate kg/s} = 895 \text{ kg/m}^3 \times 0.19 \times 10^{-3} \text{ m}^3/\text{s} = 0.17 \text{ kg/s}$$

Solving for the temperature rise in °C of the fluid we get

$$\text{Temperature increase } (^\circ\text{C}) = \frac{\text{Heat generated (kW)}}{\text{Oil specific weight}\left(\dfrac{\text{kJ}}{\text{kg}}\text{°C}\right) \times \text{ Oil flow rate (kg/s)}}$$

$$= \frac{1309}{0.17 \times 1.8 \times 1000} = 4.28\text{°C}$$

A heat exchanger relieves the hydraulic fluid of excess heat to lower its operating temperature. In gross terms, the amount of heat to be removed and transferred to the cooling medium equals the difference between the input to the hydraulic pump and the output of all system actuators. This assumes, of course, that the existing ambient temperature is appropriate for the system operation and the environmental conditions do not add or subtract heat from the fluid. This is seldom the case. In extreme environments, both cold and hot heat exchangers may be employed to counter act primarily environmental circumstances rather than operating conditions to maintain the temperature of oil within operable limits. For example, in a cold region, heat is frequently added to the fluid to decrease viscosity, whereas locating pumps, transmission lines and actuators near furnaces require that heat be subtracted from the fluid to increase viscosity and reduce the temperature. This is a different problem than lowering the temperature of fluids that are heated by working cycle itself, for example, by being discharged across a relief valve.

Example 1.7

A hydraulic machine has the following duty cycle: Idle at 12 bar for 30 s, clamp workpiece at 98 bar for 6 s, approach at 16 bar for 2 s, perform work at 300 bar for 3 s, declamp, return at 15 bar for 2 s.The pump flow is 100 LPM, the total surface area of the oil reservoir is 2.5 m^2 and the hydraulic pipeline is 25 mm in diameter and 2500 mm in length. Calculate the net wasted energy that needs to be dissipated and recommend a suitable heat exchanger if necessary. The room temperature is 22°C. The volumetric efficiency of pump = 0.86. The density of the oil used = 860 kg/m^3.

Solution: Power dissipated in kW

$$\frac{P \times Q}{600 \times \eta} = \frac{12 \times 100}{600 \times 0.86} = 2.33 \text{ kW}$$

Energy consumed during 30 s is

$$\frac{2.33 \times 30}{3600} = 0.0194 \text{kWh}$$

Power dissipated during clamping is

$$\frac{98 \times 100}{600 \times 0.86} = 18.99 \text{kW}$$

Because clamping takes place in 5 s, the energy consumed is

$$\frac{18.99 \times 6}{3600} = 0.032 \text{kWh}$$

Power dissipated during approach is

$$\frac{100 \times 16}{600 \times 0.86} = 3.1 \text{ kW}$$

Because approach takes place in 2 s, the energy consumed is

$$\frac{3.1 \times 2}{3600} = 0.0017 \text{kWh}$$

Power dissipated during forward (work) is

$$\frac{100 \times 300}{600 \times 0.86} = 58.4 \text{kW}$$

Because work takes place in 3 s, the energy consumed is

$$\frac{58.4 \times 3}{3600} = 0.0485 \text{kWh}$$

Power dissipated during return is

$$\frac{100 \times 15}{600 \times 0.86} = 2.91 \text{kW}$$

Because return takes place in 2 s, the energy consumed is

$$\frac{2.91\times 2}{3600}=0.0016\,\text{kWh}$$

Total power dissipated in 59 s = 0.1032 kWh. We can express this in kW as

$$\frac{0.058\times 3600}{59}=6.29\,\text{kW}$$

Also

$$\text{Mass flow rate} = 860 \times 0.001 = 0.86 \text{ kg/s}$$

We have

$$\text{Temperature }(°\text{C})=\frac{\text{Heat generation rate (kW)}}{\text{Oil specific heat}\left(\frac{\text{kJ}}{\text{kg}}°\text{C}\right)\times\ \text{Oil flow rate (kg / s)}}$$

$$=\frac{6.29}{1.8\times 0.86}=4.06°\text{C}$$

Temperature rise is small; therefore, there is no need for a heat exchanger.

Example 1.8

A hydraulic pump operates at 140 bar and delivers oil at 0.001 m^3/s to a hydraulic actuator. Oil discharges through a pressure relief valve during 60% of the cycle time. The pump has an overall efficiency of 82% and 15% of power is lost due to frictional pressure losses in the hydraulic lines. What heat exchanger rating is required to dissipate all the generated heat?

Solution: We have

$$\text{Pump power loss} = \text{Pump power input} - \text{Pump power output}$$

$$\text{Pump power loss} = \frac{\text{Power output}}{\text{Overall efficiency}} - \text{Power output}$$

$$\text{Pump power loss} = \left\{\frac{1}{\eta_0}-1\right\}\text{pump power output}$$

$$=\left(\frac{1}{82}-1\right)\times\left(\frac{140\times 10^5\times 0.001}{1000}\right)=3.073\text{ kW}$$

$$=(0.60)\times\left(\frac{140\times 10^5\times 0.001}{1000}\right)=8.4\text{ kW}$$

Also

$$\text{Line average loss} = \{0.60\}\times 0.15\times\left(\frac{140\times 10^5\times 0.001}{1000}\right)=1.26\text{ kW}$$

Therefore

$$\text{Total loss} = 3.073 + 8.4 + 1.26 = 12.77\text{ kW}$$

Select heat exchanger rating of 12.77 kW.

Example 1.9

What would be an adequate size of a reservoir for a hydraulic system using 0.0005 m^3/s pump?

Solution: Size of the reservoir is three to four times the capacity of the pump, that is

$$\text{Size of reservoir} = 4 \times \text{capacity of pump (LPM)}$$

$$= 4 \times 0.0005 = 4 \times 30\,(\text{LPM}) = 120 \text{ L tank is required}$$

Example 1.10

A pump delivers oil to a hydraulic motor at 20 LPM at a pressure of 15 MPa. If the motor delivers 4 kW and 80% of the power loss is due to internal leakage, which heats the oil, calculate the heat-generation rate in kJ/min.

Solution: We have

$$\text{Pump power } = \frac{0.02}{60} \times 150000 = 5 \text{ kW}$$

Motor delivers 4 kW; therefore loss is 1 kW. Now

$$\text{Power loss due to leakage} = 0.8 \times 1 = 0.8 \text{ kW}$$

$$\text{Power loss due to leakage} = 0.8 \times 60 = 48 \text{ kJ / min}$$

Example 1.11

A certain static O-ring has a diameter of 11 mm and is given an initial squeeze of 10%. The swell of the seal is equal to 20% increase in squeeze while the compression set is 10%. Calculate the final squeeze percentage of the seal**.**

Solution: We have

Final squeeze = Initial squeeze – Compression set + Swell

= 11 (0.10−0.10 +.20) = 2.2 mm

So

$$\text{Percentage of final squeeze} = \frac{2.2}{11} \times 100 = 20\%$$

Objective-Type Questions
Fill in the Blanks

1. Seals are used in hydraulic systems to prevent excessive internal and external leakage and to keep out ________.
2. ________rings are seals designed to prevent foreign abrasive or corrosive materials from entering a cylinder.
3.________ is the most widely used plastic for seals of hydraulic systems.
4. ________ almost a standard material for elastomer-type seals for use at elevated temperatures up to 240°C.
5. ________is an instrument used to measure the indentation hardness of rubber and rubber-like materials.
6. A vented ________ allows the tank to breathe as the oil level changes due to system demand requirements.
7. The pressurized reservoir usually operates at between 0.35 and ________ bar.
8. The reservoir capacity should be ________ times the pump delivery per minute. This may well be too high a volume for mobile application.
9. A hydraulic strainer is a ________ filter.
10. ________ is a surface phenomenon and refers to the tendency of particles to cling to the surface of the filters.
11. ________ filters are porous and permeable materials such as paper, wood pulp, diatomaceous earth, cloth, cellulose and asbestos.
12. Mostly filters have a beta ratio ________ than 75.
13. For hydraulic applications, the commonly used coolant in a heat exchanger is ________.

State True or False

1. Internal leakage does not cause loss of fluid because the fluid returns to the reservoir.
2. A non-positive seal allows a small amount of internal leakage.
3. Dynamic seals are subject to less wear compared to static seals.
4. V- and U-ring seals are compression-type seals used in virtually all types of rotary motion applications.
5. T-ring seals are static seals.
6. Natural rubber is commonly used as a seal material.
7. Standard liquid-to-liquid units can handle pressures up to 15 bar and temperatures up to 150°C although the actual temperature difference between oil and water should not exceed 90°C.

Review Questions

1. Explain the two types of leakages in a hydraulic system. In what way do they affect the performance of a fluid system?
2. What is a seal and what are its functions?
3. How are hydraulic seals classified? What is meant by positive sealing and non-positive sealing?
4. Distinguish between a static seal and a dynamic seal.
5. How are seals classified based on geometrical cross-section?
6. Explain the different types of sealing materials commonly used.
7. What are the primary and secondary functions of a reservoir system?
8. Explain the important elements of a reservoir system and explain the function of each.
9. What is a filter and how is it classified?
10. What are surface and depth filters?
11. What are the important locations of filters? Explain the advantages and disadvantages of each location.
12. Why should the temperature of a working fluid be properly maintained?
13. What is the purpose of seals in a hydraulic system and how are they classified?
14. List out the most commonly used types of seal configuration.
15. What is the difference between a strainer and a filter?
16. Name various filter media.
17. List out the basic types of filtering methods used in a fluid system.
18. List various locations where filters are installed in fluid power systems.
19. What are the main criteria in the design of a hydraulic system?
1. What is the purpose of a baffle plate in the fluid power pack?
.

Answers
Fill in the Blanks

1. Contamination
2. Wiper
3. Tetrafluoroethylene
4. Viton
5. A durometer
6. Breather cap
7. 1.4 bar
8. Three–four times
9. Coarse
10. Adsorption
11. Absorption
12. Greater
13. Water

State Tue or False

1 True
2. True
3. False
4. False
5. False
6. False
7. True

Lecture 32

MAINTENANCE OF FLUID POWER SYSTEMS

Learning Objectives

Upon completion of this chapter, the student should be able to:
- ☐ List the most common causes of hydraulic system breakdown.
- ☐ Explain the importance of cleanliness of hydraulic systems.
- ☐ Explain the problems caused by air in hydraulic systems.
- ☐ List the causes and remedy for excessive noise, incorrect flow, pressure and faulty operations.
- ☐ Describe the various general safety rules for electricity and electronics.
- ☐ List possible faults in solenoid valves.
- ☐ Explain the method for maintaining and disposing of fluids.

1.1 Introduction

The working medium in hydraulic systems is a fluid. Till the early 20th century, water was used as a fluid. Water as a working fluid had many drawbacks, such as low freezing point, corrosive (rust formation) nature and poor lubrication characteristics. Gradually, various oil-based fluids that had the desirable properties were developed for use in hydraulic systems.
In a hydraulic system, a hydraulic fluid has to perform various functions, such as follows:

- To transmit power, which is the primary function?
- To lubricate various moving parts, so as to avoid metal-to-metal contact, and reduce wear and noise.
- To carry the heat generated in the system due to friction between moving parts and moving fluid, and to dissipate to the environment either through a suitable heat exchanger or through the reservoir.

To perform these functions and make the system work efficiently, a hydraulic fluid must be clean and should possess certain properties.A hydraulic system is fairly easy to maintain: the fluid provides a lubricant and protects against overload. But like any other mechanism, it must be operated properly. You can damage a hydraulic system by too much speed, too much heat, too much pressure or too much contamination.

The following is a list of most common causes of hydraulic system breakdown:

- Clogged or dirty oil filters.
- Inadequate supply of oil in the reservoir.
- Leaking seals.
- Loose inlet lines that cause the pump to take in air.
- Incorrect type of oil.
- Excessive oil temperature.
- Excessive oil pressure.

Most of these and similar kinds of problems can be eliminated if a plant-preventive maintenance program is undertaken. This starts with the fluid power designer in the selection of high-quality, properly sized components. It is important for the total system to provide easy access to components requiring periodic inspection such as filters, strainers, sight gauges, drain and fill plugs, flow meters, and pressure and temperature gauges.

Over half of all hydraulic system problems have been traced directly to the oil. The test kit may be used on the spot to determine whether fluid quality permits continued use. Test that can be performed include the determination of viscosity, water content and particulate contamination level. Viscosity is measured using a visage viscosity comparator. Water content is determined by the hot-plate method. Contamination is eliminated by filtering a measured amount of hydraulic fluid, examining the particles caught on the filters under microscope and comparing what is seen with the series of photos indicating contamination levels.

For preventive maintenance techniques to be truly effective, it is necessary to have a good report and record system. This report should include the following:

- The types of symptoms encountered, how they were detected and the date.
- The description of the maintenance performed. This should include the replacement of parts, the amount of downtime and the date.
- Records of dates when oil was tested added or changed. Dates of filter changes should also be recorded.

Proper maintenance reduces hydraulic troubles. By caring for the system using a regular maintenance program, we can eliminate common problems and anticipate special ones.

1.2 The Importance of Cleanliness

Cleanliness is the first requirement when it comes to servicing hydraulic systems. Keep dirt and other contaminants out of the system. Small particles can score valves, seize pumps, clog orifices and cause expensive repair jobs.
How to keep the hydraulic system clean? Let us put it this way:

- Keep the oil clean.
- Keep the system clean.
- Keep your work area clean.
- Be careful when you change or add oil.

1.3 Importance of Oil and Filter Changes

We cannot get peak performance out of a hydraulic system that is not clean. Despite all the precautions taken when working with the hydraulic system, some contaminants get into the system anyway. Good hydraulic oils hold these contaminants in suspension and filters collect them as the oil passes through. Good hydraulic oil contains many additives that work to keep contaminants from damaging or plugging the system. However, these additives lose their effectiveness after a period of time. Therefore, oil should be changed at the recommended intervals to make sure that the additives do their job. The system filters can absorb only a limited amount of dirt particles and other contaminants from the oil. After that the filters stop working. At this point, the filters should be cleaned or replaced with new ones so that the cleaning process can be maintained.

1.3.1 Draining the System

Periodic draining of the entire hydraulic system is very important. This is the only positive way to completely remove contaminants, oxidized fluid and other substances from the system. The machine operator's manual tells the method to be used, and the frequency, depending on conditions.

1.3.2 Cleaning and Flushing the System

In some hydraulic systems, there might be deposits left in the system. It is advisable to clean and flush the system after draining the oil out. After draining the system, clean any sediment from the reservoir and clean or replace the filter elements. Flush out the old oil remaining in the system after draining, particularly if the oil is badly contaminated. Drain out the flushing oil and refill the system with clean hydraulic oil of the recommended type. Be sure to clean or replace the system filters before refilling the system.

1.3.3 Filling the System

Before filling the system, be sure the area around the filler cap is clean. Fill the reservoir to the specified level with the recommended hydraulic oil. Use only clean oil and funnels or containers, and then be sure to replace the filler cap before operating the equipment.

1.3.4 Preventing Leaks

There are internal and external leakages. Internal leakage does not result in actual loss of oil but it does reduce the efficiency of the system. External leakage does result in direct loss of oil and can have other undesirable effects as well. A hydraulic system should always be monitored for leaks and remedial actions should be taken immediately.

1.3.5 Preventing Overheating

Heat causes hydraulic oil to breakdown faster and lose its effectiveness. This is why cooling of the oil is needed. In many systems, enough heat is dissipated through the lines, the components and the reservoir to keep the oil fairly cool. But on high-pressure, high-speed circuits, oil coolers are needed to dissipate the extra heat.

To help prevent overheating, keep the oil at the proper level; clean dirt and mud from lines, reservoirs and coolers; check for dented and kinks lines; and keep relief valves adjusted properly. Also be careful to not over speed or overload the system and never hold the control valve in power position longer than necessary.

1.4 Problems Caused By Gases in Hydraulic Fluids

Gases can be present in a hydraulic fluid (or any other fluid) in three ways: free air, entrained gas and dissolved air.

1.4.1 Free Air

Air can exist in a free pocket located at some high point of a hydraulic system (such as the highest elevation of a given pipeline). This free air either existed in the system when it was initially filled or was formed due to air bubbles in the hydraulic fluid rising into the free pocket. Free air can cause the hydraulic fluid to possess a much lower stiffness (bulk modulus), resulting in a spongy and unstable operation of hydraulic actuators.

1.4.2 Entrained Gas

Entrained gas (gas bubbles within the hydraulic fluid) is created in two ways. Air bubbles can be created when the flowing hydraulic fluid sweeps air out of a free pocket and carries it along the fluid stream. Entrained gas can also be created when the pressure drops below the vapor pressure of the hydraulic fluid. When this happens, the bubbles of hydraulic fluid are created within the fluid stream. Entrained gas can cause cavitation problems in pumps and valves. These gases can greatly reduce the effective bulk modulus of hydraulic fluids, resulting in spongy and unstable operation of hydraulic actuators.

Vapor pressure is defined as the pressure at which a liquid starts to boil and thus begins changing into a vapor. The vapor pressure of a hydraulic fluid (or any other liquid) increases with an increase in temperature. Petroleum-based and fire-resistant phosphate ester fluids have very low vapor pressures. Cavitation occurs because the vapor bubbles collapse as they are exposed to the high pressure at the outlet port of the pump, creating extremely high local fluid velocities. This high-velocity fluid impacts internal metal surfaces of the pump. The resulting high impact forces cause flaking or pitting the surfaces of internal components such as gears, vanes, etc. Cavitation also interferes with the lubrication of mating moving surfaces and thus produces increased wear.

One indication of cavitation is a loud noise emanating from the pump. The rapid collapsing of gas bubbles produces vibrations of pump components, which are transmitted into pump noise. Cavitation also causes a decrease in the pump flow rate because the pumping chambers do not completely fill with the hydraulic fluid. As a result, the system pressure becomes erratic.

1.4.3 Dissolved Air

Dissolved air is in the solution and thus cannot be seen and does not add to the volume of the hydraulic fluid. Hydraulic fluids can hold an amazingly large amount of air in the solution. A hydraulic fluid, as received at atmospheric pressure, typically contains about 6% of dissolved air by volume that increases to 10% when pumped.

Dissolved air creates no problem in hydraulic systems as long as the air remains dissolved. However, if the dissolved air comes out of the solution, it forms bubbles in the hydraulic fluid and thus becomes entrained air.

The following will help control or eliminate pump cavitation by keeping the suction pressure above the vapor pressure of the fluid:

- Keep suction velocities below 1.5 m/s.
- Keep pump inlet lines as short as possible.
- Minimize the number of fittings in the pump inlet line.
- Mount the pump as close as possible to the reservoir.
- Use low-pressure drop-pump inlet filters or strainers.
- Use a properly designed reservoir that can remove entrained air from the fluid before it enters the pump inlet line.
- Use proper oil as recommended by the manufacturer.
- Keep the oil from exceeding the recommended maximum temperature level.

1.5 Troubleshooting Guides

The following troubleshooting guides are arranged in five main categories. The heading of each is a symptom that indicates some malfunction in the system. The causes and remedies are given in Tables 1.1–1.5.

1.5.1 Fluid Maintenance

Fluid maintenance can be done in the following ways:

- Before opening a drum, clean the top and the bung
- Use only clean containers and hoses to transfer the hydraulic fluid.
- Provide a 200-mesh screen in the reservoir filter pipe.
- Store drums indoor or under a roof.

1.5.2 In-Operation Care of Hydraulic Fluid

This can be done in the following ways:

- Keep the system tight and repair all leaks immediately.
- Use proper air and fluid filtration.
- Establish fluid change intervals.
- Keep the reservoir filled properly to take the advantage of its heat-dissipating characteristics and prevent moisture from condensing on inside walls.

Table 1.1 Excessive noise

Symptom	Cause	Remedy
Pump noisy	Cavitation	Any or all of the following: ☐ Replace dirty filters. ☐ Wash strainers. ☐ Clean the clogged inlet line. ☐ Clean the reservoir breather vent. ☐ Change the system fluid. ☐ Change to proper pump drive motor speed. ☐ Overhaul or replace the pump. ☐ Check fluid temperature.
	Air in fluid	Any or all of the following: ☐ Tighten leaky inlet connections. ☐ Fill the reservoir to proper level. ☐ Bleed air from the system. ☐ Replace the pump shaft seal.
	Coupling misaligned	All of the following: ☐ Align unit. ☐ Check the condition of seals, bearings and couplings.
	Pump worn or damaged	Overhaul or replace defective parts
Motor noisy	Coupling misaligned	All of the following: ☐ Align unit. ☐ Check the condition of seals, bearings and couplings.

	Motor or coupling worn	Overhaul or replace defective parts
Relief valve noisy	Setting too low or too	Install and adjust pressure gauge

Table 1.2 Excessive heat

Symptom	**Cause**	**Remedy**
Pump heated	Fluid heated	See symptom "fluid heated"
	Cavitation	Any or all of the following: ☐ Replace dirty filters. ☐ Wash strainers. ☐ Clean the clogged inlet line. ☐ Clean the reservoir breather vent. ☐ Change the system fluid. ☐ Change to proper pump drive motor speed. ☐ Overhaul or replace the pump. ☐ Check fluid temperature.
	Air in fluid	Any or all of the following: ☐ Tighten leaky inlet connections. ☐ Fill the reservoir to proper level. ☐ Bleed air from the system. ☐ Replace the pump shaft seal.
	Excessive load	All of the following: ☐ Align unit. ☐ Check the condition of seals, bearings and couplings. ☐ Locate and correct mechanical binding. ☐ Check for workload in excess of circuit design.
	Pump worn or damaged	Overhaul or replace defective parts
	Relief or unloading valve set	Install and adjust pressure gauge
Motor heated	Fluid heated	See symptom "fluid heated"
	Relief or unloading valve set	Install and adjust pressure gauge
	Excessive loading	All of the following: ☐ Align unit. ☐ Locate and correct mechanical binding. seals, bearings and couplings. ☐ Check for workload in excess of circuit design.
	Motor or coupling worn or	Overhaul or replace defective parts
Relief valve heated	Fluid heated	See symptom "fluid heated"
	Valve setting	Install and adjust pressure gauge

	Worn or damaged valve	Overhaul or replace defective parts
Fluid heated	System pressure too	Install and adjust pressure gauge
	Unloading valve set	Install and adjust pressure gauge
	Fluid dirty or low supply	☐ Change filters. ☐ Check system fluid viscosity, change if necessary. ☐ Fill the reservoir to proper level.
	Incorrect fluid viscosity	☐ Change filters. ☐ Check system fluid viscosity, change if necessary. ☐ Fill the reservoir to proper level.
	Faulty fluid cooling system	☐ Clean the cooler and/or strainer. ☐ Replace the cooler control valve. ☐ Repair or replace the cooler.
	Worn pump, valve, motor,	Overhaul or replace defective parts

Table 1.3 Incorrect flow

Symptom	Cause	Remedy
No flow	Pump not receiving fluid	Any or all of the following: ☐ Replace dirty filters. ☐ Clean the clogged inlet line. ☐ Clean the reservoir breather vent. ☐ Change the system fluid. ☐ Overhaul or replace the pump.
	Pump drive motor not operating	Overhaul

	Pump to drive coupling sheared	Check for the damaged pump Replace and align coupling.
	Pump drive motor turning in the wrong	Reverse rotation
	Directional control set in the wrong direction	Any or all of the following: ☐ Check the position of manually operated controls. ☐ Check the electrical circuit on solenoid- operated controls. ☐ Repair or replace pilot pressure pump.
	Entire flow passing over the relief	Adjust part
	Damaged pump	Check for the damaged pump Replace and align coupling.
	Incorrectly assembled pump	Overhaul or replace part
Excessive flow	Flow control set too high	Adjust part
	Yoke actuating device inoperative (variable	Overhaul or replace part
	Rotation per minute (RPM) of pump drive	Replace with correct unit
	Improper size pump used for replacement	Replace with correct unit
Low flow	Flow control set too low	Adjust part
	Relief or unloading valve set too low	Adjust part
	Flow bypassing through the partially open valve	Overhaul or replace part or any or all of the following: ☐ Check the position of manually operated controls ☐ Check the electrical circuit on solenoid- operated controls. ☐ Repair or replace the pilot pressure pump.
	External leak in the system	☐Bleed air from the system.
	Yoke actuating device inoperative (variable displacement pump)	Overhaul or replace part
	RPM of pump drive motor incorrect	Replace with correct unit
	Worn pump, valve motor, cylinder or other	Overhaul or replace part

Table 1.4 Incorrect pressure

Symptom	Cause	Remedy
No pressure	No flop	See "incorrect flow", symptom "no flow"
Low pressure	Pressure relief path exists	See "incorrect flow," symptom "no flow" and "low flow"
	Pressure-reducing valve set too low	Adjust part
	Pressure-reducing valve damaged	Overhaul or replace part
	Damaged pump, motor or cylinder	Overhaul or replace part
Erratic pressure	Air in fluid	☐ Tighten leaky connections. ☐ Fill the reservoir to proper level. ☐ Bleed air from the system.
	Worn relief valve	Overhaul or replace part
	Contamination in fluid	Replace dirty filters and system fluid
	Accumulator defective or had lost charge	☐ Check the gas valve for leakage. ☐ Change to correct pressure. ☐ Overhaul if defective.
	Worn pump, motor or cylinder	Overhaul or replace part
Excessive pressure	Pressure-reducing, relief or unloading valve misadjusted	Adjust part
	Yoke actuating device inoperative (variable displacement pumps)	Overhaul or replace part
	Pressure-reducing, relief or unloading valve worn or damaged	Overhaul or replace part

Table 1.5 Faulty operation

Symptom	Cause	Remedy
No movement	No flow or pressure	See “incorrect flow”
	Limit or sequence device inoperative or misadjusted	Overhaul or replace part
	Mechanical bind	Locate bind and repair
	No command signal to the servo amplifier	Repair command console or interconnecting wires
	Inoperative or misadjusted servo amplifier	Adjust, repair or replace part
	Inoperative servo valve	Overhaul or replace part
	Worn or damaged cylinder or motor	Overhaul or replace part
Slow movement	Low flow	See “incorrect flow”
	Fluid viscosity to high	☐ Check fluid temperature. ☐ Check system fluid viscosity, change if necessary.
	Insufficient control pressure for valves	☐ See “incorrect pressure”
	No lubrication of machine ways or linkage	Lubricate
	Misadjusted or malfunctioning servo amplifier	Adjust, repair or replace part
	Sticking servo valve	☐ Clean and adjust or replace part. ☐ Check the condition of system fluid and filters.
	Worn or damaged cylinder or motor	Overhaul or replace part
Erratic movement	Erratic pressure	See “incorrect pressure”
	Air in fluid	See “excessive noise”
	No lubrication of machine ways or linkage	Lubricate
	Erratic command signal	Repair command console or interconnecting wires
	Misadjustment of malfunctioning servo amplifier	Adjust, repair or replace part
	Malfunctioning feedback transducer	Overhaul or replace part
	Sticking servo valve	☐ Clean and adjust or replace part. ☐ Check the condition of system fluid and filters.
	Worn or damaged cylinder or motor	Overhaul or replace part
Excessive speed or movement	Excessive flow	See “incorrect flow”
	Feedback transducer	Overhaul or replace part

1.6 General Safety Rules for Electricity and Electronics

Following are the general safety rules for electricity and electronics:

- Use approved tools, equipment and protective devices.
- Avoid wearing rings, bracelets and similar meal items when working around exposed electric circuits.
- Never assume that a circuit is OFF. Double check it with an instrument that is supposed to be surely operational.
- Some situations require a "buddy system" to guarantee that power would not be turned ON
- Never tamper with or try to override safety devices such as interlock (a type of switch that automatically removes power when a door is opened or a panel removes).
- Keep tools and test equipment clean and in good working condition. Replace insulated probes and leads at the first sign of deterioration.
- Some devices, such as capacitors, can store a lethal charge. They may store this charge for long periods of time. It must be certain that these devices are discharged before working around them.
- Do not remove grounds and do not use adapters that defeat the equipment ground.
- Use protective clothing and safety glasses when handling high vacuum devices such as picture tubes and cathode ray tubes.
- Do not work on equipment before knowing proper procedures and having awareness of any potential safety hazards.

1.6.1 Solenoid Valves

Pneumatics will continue to dominate the power section and retain its positions in the control section. On the other hand, it is impossible to deny the advances made by electronic controls, systems and components due to the following advantages:

- Low power consumption.
- Short switching times.
- Higher contact ratings.
- Long service life.
- Maintenance-free operation.

1.6.1.1 Possible Faults in Solenoid Valves

Following are the possible faults:

- Directed short (at power supply, electrical bus and load): A direct short is when too much current is sent back to the power supply overloading it, generally blowing a fuse.
- Cross short: A cross short is created by one or more wires (cables) bypassing the load causing a direct short to occur.
- High resistance connections (too many connections at the terminal eye).
- Low voltage or over voltage at the solenoid.
- Corrosion.
- Partially or fully blocked hoses.

- Wire connections are open internally
- Lack of source pressure (at the compressor or on the service unit).
- Sticking spool.
- Diaphragm not working.
- Exhaust ports blocked.
- Gaskets mounted incorrectly.
- Faults caused by wear or external influences.
- Caution: Short circuiting of the power supply is not recommended without the installation of a “circuit breaker” to protect the equipment and the user.

Causes and remedy for troubleshooting for direct shorts and faults in relay coil is given in Tables 1.6 and 1.7.

Table 1.6 Troubleshooting for direct shorts

Fault	**Cause**	**Remedy**
When the push button is pressed, solenoid Y_1 is not activated and the circuit breaker must be reset.	Relay coil K_1 has been shorted directly to the ground bypassing the load (relay coil K_1)	Pull the ground cable connected to solenoid Y_1, retry the circuit. If the power supply continues to short, the location of the direct short is at relay coil K_1. If not, the direct short is at solenoid Y_1.
	Solenoid Y_1 has been shorted directly to ground bypassing the load (solenoid coil). This causes the relay coil to buzz (noise like a bee) and the circuit breaker must be reset.	
When proximity limit switch a_1 and pressure switch a_2 are activated, solenoid Y_3 is not activated and the circuit breaker must be reset.	Relay coil K_2 has been shorted directly to ground, bypassing the load (relay load).	Pull the ground cable connected to solenoid Y_3. Retry the circuit. If power supply continues to short, the location of the direct short is at relay coil K_2. If not, the direct short is at solenoid Y_3.
	Solenoid Y_3 has been shorted directly to ground, bypassing the load (solenoid). The circuit breaker must be reset.	

Table 1.7 Faults in relay coils

Fault	**Cause**	**Remedy**
When powered, the relay coil does not function.	One or more of the wires (leads) have an infinite resistance.	Use the voltmeter to check the potential difference across each cable and the push button when switch is open, then closed. If the problem is with any of these components
	Open (infinite resistance) in the coil.	
	Low voltage: Below specification.	
Relay coil is not energized by electrical limit switch (or proximity limit switch).	The electrical limit switch is not being activated.	Visually examine the electrical limit switch to make sure that the roller is fully activated. If not, reposition the sensor so that physical contact is achieved (or see LED).
		Use the voltmeter to test the difference in potential across the limit switch (24 V when open, 0 V when closed); replace if not functioning or remove the limit switch from the circuit and use the ohmmeter to measure the resistance of the limit switch (infinite pressure when open, approximately 0 when closed).
	One of the cables (wires) is either not connected or there is infinite resistance in the cable (wire).	Use the multimeter to check the difference in potential across each cable (wire) in the corresponding ring. Then remove the suspect cable(s) and use the ohmmeter to confirm your findings, replace the cables if required.
	The relay coil itself is malfunctioning, that is, low voltage, an open (infinite resistance) circuit, loose connection, high-resistance connection.	If the voltmeter identifies low voltage, then check the power supply and original source. Replace or modify the source as required. If an “open” infinite resistance, loose or high connection is suspected, use the ohmmeter to determine the exact location of the problem and repair/replace as required.

1.7 Maintaining and Disposing of Fluids

Controlling pollution and conserving natural resources are important goals to achieve for the benefit of society. Thus, it is important to minimize the generation of waste hydraulic fluids and to dispose them in an environmentally friendly manner. The following are some recommendations that should be adhered to strictly for properly maintaining and disposing hydraulic fluids:

- Select the optimum fluid for the application involved. This includes the consideration of the system operating pressures and temperatures as well as the desired fluid properties.

- Utilize a well-designed filtration system to reduce contamination and increase the useful life of the fluid. Filtration should be continued and filters should be changed at regular intervals.

- **3.** Follow a proper storage procedure for the unused fluid supply. For example, outdoor storage is not recommended, especially if the fluid is stored in drums as it is affected by increment weather and resulting weakening of drum seams may occur and cause leakage and contamination.

- Fluids from the storage containers to the hydraulic systems should be transported carefully as the chances of contamination depend to a large extent on handling. The transfer containers should be covered when not in use.

- Operating fluids should be checked regularly for viscosity, acidity, bulk modulus, specific gravity, water content, additive levels and particle contamination.

- The entire hydraulic system, including pumps, piping, filters, actuators and reservoir, should be maintained according to the manufacturer's specifications.

- Corrective action should be taken to reduce or eliminate leakage from operating hydraulic systems. Typically leakage occurs due to worn seals or loose fittings. A preventive maintenance program should be implemented to ensure ideal operating conditions and reduce contamination, leakage, etc.

- Fluids must be disposed properly because a hydraulic fluid is considered to be a waste material when it has deteriorated to the point where it is no longer suitable for use in hydraulic systems. The various environmental government agencies also suggest against mixing hazardous wastes with waste hydraulic fluids being disposed. It is also not allowed to burn these waste fluids in non-industrial boilers.

- Pollution control and conservation of natural resources are critical environmental issues for society. Properly maintaining and disposing of fluids not only protects the environment but also conserves our natural resources.

Objective-Type Questions

Fill in the Blanks

1. The primary function of a hydraulic fluid is to transmit ________.
2. Over half of all hydraulic system problems have been traced directly to the ________.
3. Entrained gas can also occur when the pressure drops below the ________of the hydraulic fluid.
4. Cavitation occurs because the vapor bubbles collapse as they are exposed to the ________ pressure at the outlet port of the pump, creating extremely high local fluid velocities.
5. Oxidation is caused by the chemical reaction of oxygen from the air with particles of ________.

State True or False

1. Dissolved air creates no problem in hydraulic systems as long as the air remains dissolved.
2. Most of fire-resistant fluids are compatible with most natural or synthetic rubber seals.
3. The neutralization number is a measure of the relative acidity.
4. A 10 µm filter is one capable of removing contaminants as small as 10 µm in size.
5. Free air can cause the hydraulic fluid to possess a much lower stiffness.

Review Questions

1. Name five reasons for the overheating of the fluid in a hydraulic system.
2. Name four causes of low or erratic pressure.
3. What three devices are commonly used in the troubleshooting of hydraulic circuits?
4. Name five of the most common causes of hydraulic system breakdown.

5.List eight recommendations that should be followed for properly maintaining and disposing hydraulic fluid.
6.Name two items that should be included in reports dealing with a maintenance procedure.

7. Name the three ways in which a hydraulic fluid becomes contaminated.
8. Name five things that can cause a noisy pump.
9.Name four causes of low or erratic pressure.
10.Name four causes of no pressure.
11.If an actuator fails to move, name five possible causes.
12.If an actuator has slow or erratic motion, name five possible causes.
13. Why is loss of pressure in a hydraulic system not a symptom of pump malfunction?

Answers
Fill in the Blanks

1. Power
2. Oil
3. Vapor pressure
4. High
5. Oil

State True or False

1. True
2. False
3. True
4. True
5. True

Lecture 33

INTRODUCTION TO PNEUMATICS

Learning Objectives

Upon completion of this chapter, Student should be able to

- Explain the meaning of Pneumatics
- Describe the various properties desired of a air medium in pneumatic system
- Explain the advantages and disadvantages of compressed air
- Identify and appreciate the application of pneumatic systems in various Industries
- Describe the various gas laws
- List the basic components required for a pneumatic systems
- Describe the various power transmission systems
- Compare hydraulic, pneumatic and mechanical systems

1.1 PNEUMATICS AND ITS MEANING.

The English word pneumatic and its associate noun pneumatics are derived from the Greek "**pneuma**" meaning breath or air. Originally coined to give a name to the science of the motions and properties of air. Compressed air is a vital utility- just like water, gas and electricity used in countless ways to benefit everyday life. Pneumatics is application of compressed air (pressurized air) to power machine or control or regulate machines. Simply put, Pneumatics may be defined as branch of engineering science which deals with the study of the behavior and application of compressed air. Pneumatics can also be defined as the branch of fluid power technology that deals with generation, transmission and control of power using pressurized air. Gas in a pneumatic system behaves like a spring since it is compressible.

Any gas can be used in pneumatic system but air is the most usual, for obvious reasons. Exceptions are most likely to occur on aircraft and space vehicles where an inert gas such as nitrogen is preferred or the gas is one which is generated on board. Pure nitrogen may be used if there is a danger of combustion in a work environment. In Pneumatic control, compressed air is used as the working medium, normally at a pressure from 6 bar to 8 bar. Using Pneumatic

Control, maximum force up to 50 kN can be developed. Actuation of the controls can be manual, Pneumatic or Electrical actuation. Signal medium such as compressed air at pressure of 1-2 bar can be used [Pilot operated Pneumatics] or Electrical signals [D.C or A.C source- 24V – 230V] can be used [Electro pneumatics]

1.2 CHOICE OF WORKING MEDIUM AND SYSTEM.

The choice of medium depends on the application. Some of the general, broad rules followed in the selection of a working medium are listed below.

- When the system requirement is high speed, medium pressure (usually 6 to 8 bar) and less accuracy of position, then pneumatic system is preferred.
- If the system requirement is high pressure and high precision, a fluid system with oil is good.
- When the power requirement is high like in forging presses, sheet metal press, it is impossible to use air system. Oil hydraulics is the only choice
- Air is used where quick response of actuator is required.
- If temperate variation range in the system is large, then use of air system may run into condensation problems and oil is preferred.
- If the application requires only a medium pressure and high positional accuracy is required then hydro –pneumatic system is preferred
- Air is non-explosive, it is preferred where fire/electric hazard are expected. Oil systems are more prone to fire and electrical hazards and are not recommended in such applications.
- Because air contains oxygen (about 20%) and is not sufficient alone to provide adequate lubrication of moving parts and seals, oil is usually introduced into the air stream near the actuator to provide this lubrication preventing excessive wear and oxidation.

In a practical sense, compressed air is a medium that carries potential energy. However it can be expensive to produce, and from a simple energy efficiency point of view compressed air may not appear advantageous at first. Considering that it takes about 6 kW of electrical energy to generate

0.75 kW output on an air motor, compressed air has an efficiency rating of only 12%. In spite of that compressed air is used due to its other advantages.

Table 1.1: The advantages and disadvantages of compressed air.

Advantages of compressed air	**Disadvantages of compressed air**
Air is available in unlimited quantities	Compressive air is relatively expensive means of conveying energy The higher costs are, however. Largely compensated by the cheaper elements. Simpler and more compact equipment
Compressed air is easily conveyed in pipelines even over longer distances	
Compressed air can be stored	Compressed air requires good conditioning. No dirt or moisture residues may be contained in it. Dirt and dust leads to wear on tools and equipment
Compressed air need not be returned. It can be vented to atmosphere after it has performed work	It is not possible to achieve uniform and constant piston speeds(air is compressible)
Compressed air is insensitive to temperature fluctuation. This ensures reliable operation even in extreme temperature conditions	Compressed air is economical ony up to certain force expenditure. Owing to the commonly used pressure of 7 bar and limit is about 20 to 50 kN, depending on the travel and the speed. If the force which is required exceeds this level, hydraulics is preferred
Compressed air is clean. This is especially important in food, pharmaceutical, textile, beverage industries	The exhaust is loud. As the result of intensive development work on materials for silencing purposes, this problems has however now largely been solved
Operating elements for compressed air operation are of simple and inexpensive construction.	The oil mist mixed with the air for lubricating the equipment escapes with the exhaust to atmosphere.
Compressed air is fast. Thus, high operational speed can be attained.	Air due to its low conductivity , cannot dissipate heat as much as hydraulic fluid
Speeds and forces of the pneumatics elements can be infinitely adjusted	Air cannot seal the fine gaps between the moving parts unlike hydraulic system
Tools and operating elements are overload proof. Straight line movement can be produced directly	Air is not a good lubricating medium unlike hydraulic fluid.

Differences between hydraulic and pneumatic systems.

One of the main differences between the two systems is that in pneumatics, air is compressible. In hydraulics, liquids are not. Other two distinct differences are given below.

Pneumatic Systems

These systems have two main features:

- Pneumatic systems use compressed gas such as air or nitrogen to perform work processes.
- Pneumatic systems are open systems, exhausting the compressed air to atmosphere after use.

Hydraulic Systems

These systems also have two main features:

a) Hydraulic systems use liquids such as oil and water to perform work processes.
b) Hydraulic systems are closed systems, recirculating the oil or water after use.

1.3 APPLICATIONS OF PNEUMATICS

Pneumatic systems are used in many applications. New uses for pneumatics are constantly being discovered. In construction, it is indispensible source of power for such tools as air drills, hammers, wrenches, and even air cushion supported structures, not to mention the many vehicles using air suspension , braking and pneumatic tires.

In manufacturing, air is used to power high speed clamping, drilling, grinding , and assembly using pneumatic wrenches and riveting machines. Plant air is also used to power hoists and cushion support to transport loads through the plant.

Many recent advances in air – cushion support are used in the military and commercial marine transport industry.

Some of the Industrial applications of pneumatics are listed in the Table 1.2

Table 1.2: Industrial applications of Pneumatics

Material Handling	Manufacturing	Other applications
Clamping Shifting positioning Orienting Feeding Ejection Braking Bonding Locking Packaging Feeding Sorting stacking	Drilling Turning Milling Sawing Finishing Forming Quality Control Stamping Embossing Filling	Aircraft Cement plants chemical plants Coal mines Cotton mills Dairies Forge shops Machine tools Door or chute control Turning and inverting parts

1.4 PROPERTIES OF AIR

1.4.1 **Composition:** Air is one of the three states of matter. It has characteristics similar to those of liquids in that it has no definite shape but conforms to the shape of its container and readily transmits pressure. Gases differ from liquids in that they have no definite volume. That is , regardless of the size or shape of the containing vessel, a gas will completely fill it. Gases are highly compressible, while liquids are only slightly so. Also, gases are lighter than equal number of liquids, making gases less dense than liquids.

Air is a mechanical mixture of gases containing by volume, approximately 78 % of nitrogen and 21 % of oxygen, and about 1 % of other gases, including argon and carbon dioxide. Water being the most important remaining ingredient as far as pneumatics is concerned. The dilution of the oxygen by nitrogen makes air much less chemically active than pure oxygen but it is still capable of causing spontaneous combustion or explosion , particularly if oil vapor at an elevated temperature is present, as may occur in an air receiver.

Air is colorless, odorless, tasteless, and compressible and has weight. Air has a great affinity with water and unless specifically dried, contains considerable quantities of water vapour, sometimes as much as 1% by weight. Life on earth depends on air for survival and man harness its forces to do useful work. Table 1.3 gives the physical properties of air

Table 1.3: Properties of air

Property	Value
Molecular weight	28.96 kg/kmol
Density of air at 15 and 1 bar	1.21 kg/m^3
Boiling point at 1 bar	191 to -194
Freezing point at 1 bar	-212 to -216
Gas constant	286.9 J/kg K

1.4.2 Free air and Standard air

In pneumatics, the existence of the following two conditions of atmospheric air is well accepted

- **Free air:** Air at the atmospheric condition at the point where the compressor is located is defined as free air. Free air will vary with atmospheric conditions like altitude, pressure and temperature.

- **Standard air**: It is also called normal air. It is defined as the air at sea level conditions (1.01324 bar as per ISO –R554 and 20 ℃ and Relative humidity of 36%). The condition of normal atmosphere is used as a basis for getting average values for compressor delivery volumes, efficiencies and operating characteristics.

1.4.3 **Atmospheric pressure**, **Gauge pressure and Absolute pressure**

Air has mass and exerts a pressure on the surface of the earth. A barometer consisting of an inverted tube close at the top will support a column of mercury at exactly 760 mm at sea level when measured at standard conditions. Pressure above one atmosphere (~ 1 bar) are positive, whereas the pressure below one atmosphere cause a vacuum to be formed. Both positive pressures and vacuum pressures have useful purposes in pneumatics. Vacuum measurement is usually given a mm of mercury and then converted into the holding force for such devices such as suction pads and cylinders with a specified diameter.

Atmospheric pressure: The earth is surrounded by air. Since air has weight it can exert a pressure on the earth's surface. The weight of the column of air on one square meter of earth's surface is known as atmospheric pressure or reference pressure.

The atmospheric pressure varies slightly from day to day. In pneumatic circuit calculations, standard atmospheric pressure is taken as 101.325 kPa.(14.7 psia)(760 mm of Hg). The atmospheric pressure is measured using barometer.

Atmospheric pressure = Density x Acceleration due to gravity x height of barometer column (usually mercury)

$p_{atm} = \rho\, g\, h,$

For a mercury we can use ,

$\rho = 13600 \frac{kg}{m^3},\ \ g = 9.81 \frac{m}{s^2},\ \ h = 760\, mm\ at\ sea\ level$

$$p_{atm} = \rho\, g\, h = 13600\ \times 9.81 \times 0.760 = 101396 \frac{N}{m^2} = 1.013\, atm = 1.013\, bar$$

Gauge Pressure: In pneumatic application, pressure is measured using pressure gage and pressure gauges are calibrated to indicate the pressure above that of the Atmospheric pressure. Gauge pressure refers to pressure indicated by pressure gauge.

Absolute pressure: refers to the true or total pressure. Absolute pressure = Atmospheric pressure + Gauge pressure. Calculations involving formulae associated with the Gas laws must be made with absolute pressure. Figure 1.1 shows the difference between the gauge and absolute pressure. Let's examine the two pressure levels P_1 and P_2.

Relative to a prefect vacuum, they are

p_1= 0.7 bar (absolute) (a pressure less than a atmospheric pressure)

p_2= 2 bar (absolute) (a pressure greater than a atmospheric pressure)

Relative to the atmosphere, they are

p_1= -0.3 bar (Gauge) (Suction) (or vacuum)

p_2 = 1bar (Gauge)

As can be seen from Figure 1.1, the following rule can be used in pressure conversion calculations

Absolute pressure = Gauge pressure +Atmospheric pressure

It should be noted that vacuum or suction pressures exits in certain location of fluid power systems (for example, in the inlet or suction lines of pumps). Therefore, it is important to understand the meaning of pressures below atmospheric pressure. One way to generate a suction pressure is to remove some of fluid from a closed vessel initially containing fluid at atmospheric pressure.

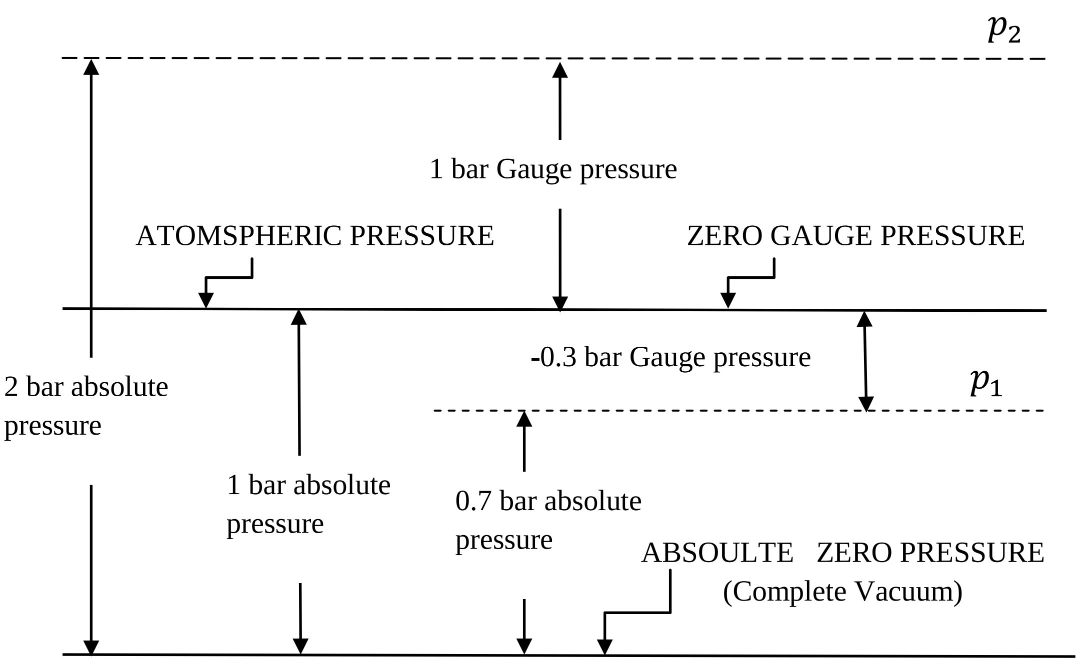


Figure 1.1 Difference between absolute and gauge pressure.

1.4.4 Variation of pressure with altitude

According to Barometric law, the atmospheric pressure decreases exponentially with the increase in altitude. The relation is shown graphically in the Figure 1.2. Variation of pressure with respect to altitude is linear up to 6000 m and pressure drops about 11 kPa per 1000 m change in altitude.

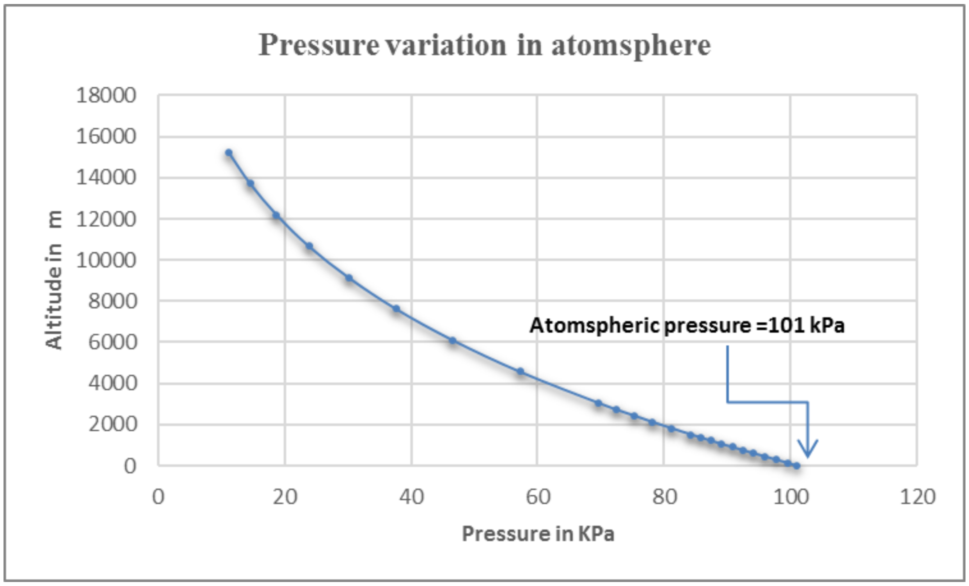


Figure 1.2 **Pressure variations in atmosphere.**

1.5 GAS LAWS

Early experiments were conducted concerning the behavior of air and similar gases. These experiments were conducted by scientists such as Boyle, Charles and Gay-Lussac. The results of their experiments indicated that gases behaviors follow the law known as ideal gas laws.

1.5.1 Boyle's Law

Robert Boyle (1627-1691), an English scientist, was among the first to experiment with the pressure volume relationship of gas at constant temperature.

Statement: If a given mass of a gas is compressed or expanded at a constant temperature, then the absolute pressure is inversely proportional to the volume.

$$Pressure \propto \frac{1}{Volume}, \qquad when\ temperaute = constant$$

$$or\ \ pV = constant, for\ state\ 1\ and\ 2, p_1V_1 = p_2V_2$$

Example 1.1: .50 mm diameter piston of the pneumatic cylinder of Figure 1.3 retracts 130 mm from its present position. $(p_1 = 2bar\ (gauge)V_1 = 300cm^3)$ due to the external load on the rod. If the part at the blind end of the cylinder is blocked, find the new pressure, assuming temperature remains constant.

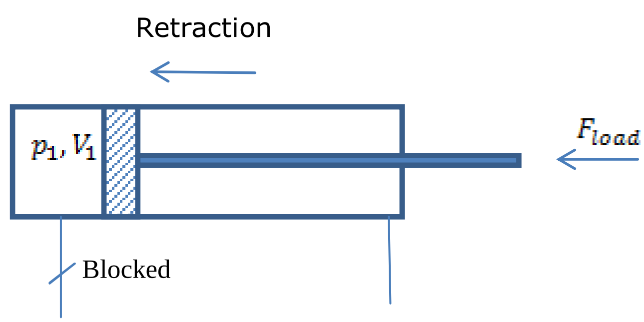


Figure 1.3

Given Data:

D= diameter of piston = 50mm =5cm

Stroke= L = 130 mm =13 cm

$p_1 = 2bar\ (gauge) V_1 = 300cm^3$

Solution

$p_1 = 2bar\ (gauge) + atomspheric\ \ pressure = 3\ bar\ abs.$

Final volume = $V_2 = V_1 - \left(\frac{\pi}{4} D^2 L\right) = 300 -\ (0.7854 \times (5^2) \times 13) = 45cm^3$

Since the temperature is constant, we can use Boyle's Law

$$p_1 V_1 = p_2 V_2$$

$$3 \times 300 = p_2 \times 45$$

Thus, $p_2 = 20\ bar\ (abs) = \mathbf{19\ bar\ (gauge)}$

Example 1.2: . Four cubic m of nitrogen are under a pressure of 7 bar (gauge) the nitrogen is allowed to expand to a volume of 6 cubic m . What is the new gauge pressure ?

Given Data:

$p_1 = 7bar\ (gauge) = 8\ abs,\ \ V_1 = 4\ m^3$

$V_2 = 6\ m^3$

Solution

Since the temperature is constant, we can use Boyle's Law

$$p_1V_1 = p_2V_2$$

$$8 \times 4 = p_2 \times 6$$

Thus, $p_2 = 5.333\ bar\ (abs) = \boldsymbol{4.333\ bar\ (gauge)}$

Example 1.3: . Piston compresses air at atmospheric pressure to 1/7th the volume as illustrated in the Figure 1.4 Assuming constant temperature, what is the gauge pressure of the resulting air?

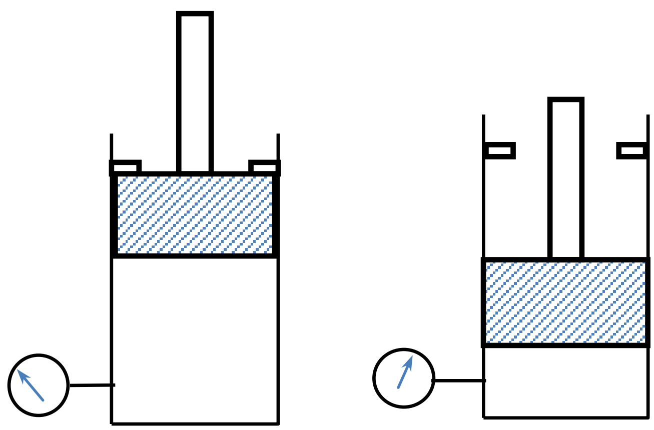


Figure 1.4

Solution

$$\frac{V_2}{V_1} = \frac{1}{7}\ or\ \frac{V_1}{V_7} = 7$$

Since the temperature is constant, we **can use** Boyle's Law

$$p_1V_1 = p_2V_2$$

$$p_2 = p_1 \times \frac{V_1}{V_2}$$

$$p_2 = 1 \times \frac{7}{1} = 7\ bar(absoulte)$$

$$p_2 = 6\ bar(guage)$$

1.5.2. Charles law

Boyle's law assumes conditions of constant temperature. In actual situations this is rarely the case. Temperature changes continually and affects the volume of a given mass of gas.

Jacques Charles (1746 to 1823),a French physicist, provided much of the foundations for modern kinetic theory of gases. Through experiments, he found that all gases expand and contract proportionally to the change in the absolute temperature, providing the pressure remains constant.

Statement: If a given mass of a gas is heated or cooled at a constant pressure, then the volume is directly proportional to the absolute temperature.

$$Volume\ \propto Temperature, \qquad when\ pressure = constant$$

$$or\ \ \frac{V}{T} = constant, for\ state\ 1\ and\ 2, \qquad \frac{V_1}{T_1} = \frac{V_2}{T_2}$$

Example 1. 4: .The cylinder of Figure 1.4 as an initial position where $p_1 = 2bar\ (gauge)V_1 = 300cm^3)$ as controlled by the load on the rod. The air temperature is 30 ℃. The load on the rod is held constant to maintain constant air pressure, but the air temperature has increased to 65℃. Find the new volume of air the blank end of the cylinder.

Given Data:

$p_1 = 2bar\ (gauge)V_1 = 300cm^3$,

Solution

Since the pressure is constant, From Charles law, we have

$$\frac{V_1}{T_1} = \frac{V_2}{T_2}$$

$T_1 = temperature\ in\ Kelvin = 30 + 273 = 303\ K,\ \ T_2 = 65 + 273 = 338\ K$

$$\frac{300}{303} = \frac{V_2}{338}$$

Solving we get, $V_2 = 145\ cm^3 = 145 \times 10^{-6}\ m^3$

Example 1.5: A sample of $H_2(g)$ occupies a volume of 69.37 cm^3 at a pressure of exactly 1 atm when immersed in a mixture of ice and water. When the gas (at the same pressure) is immersed in boiling benzene, its volume expands to 89.71 cm^3. What is the boiling point of benzene?

Given Data:

$p_1 = 1bar\ (gauge) V_1 = 69.37cm^3\ \ V_2 = 89.71cm^3$,

Solution

Since the pressure is constant, From Charles law, we have

$$\frac{V_1}{T_1} = \frac{V_2}{T_2}$$

$T_1 = temperature\ in\ Kelvin = 0 + 273 = 273\ K,$

$$\frac{69.37}{273} = \frac{89.71}{T_2}$$

Solving we get, $T_2 = 353.05\ K$, yields the desired result. (The ice-water mixture must be at 273.15 K, the freezing point of water.)

Example 1.6: A cylinder of gas under a pressure of 125 bar at 70 °F is left out in the sun in tropics and heats up to a temperature of 130 °F . What is the new pressure within the cylinder.

Given Data:

$p_1 = 125bar\ (gauge) T_1 = 70\ °F = 70 + 460 = 530\ K$,

$T_2 = 130\ °F = 130 + 460 = 590\ K$

Solution

Since the pressure is constant, From Charles law, we have

$$\frac{P_1}{T_1} = \frac{P_2}{T_2}$$

$$\frac{126}{590} = \frac{P_2}{530}$$

Solving we get, $P_2 = 140.3\ bar\ (absoulte) = 139.3\ bar(gauge)$

1.5.3 Gay-Lussac's Law

A third gas law may be derived as a corollary to Boyle's and Charles's laws. Suppose we double the thermodynamic temperature of a sample of gas. According to Charles's law, the volume should double. Now, how much pressure would be required at the higher temperature to return the gas to its original volume? According to Boyle's law, we would have to double the pressure to halve the volume. Thus, if the volume of gas is to remain the same, doubling the temperature will require doubling the pressure. This law was first stated by the Frenchman Joseph Gay-Lussac (1778 to 1850).

Statement: At constant pressure, the absolute pressure of an ideal gas will vary directly with the absolute temperature.

$$Pressure \propto Temperature, \qquad when\ Volume = constant$$

$$or\ \frac{p}{T} = constant, for\ state\ 1\ and\ 2, \qquad \frac{p_1}{T_1} = \frac{p_2}{T_2}$$

Example 1.7: .The cylinder of Figure (same as above figure no) has a locked position ($V_1 = constant$). $p_1 = 2bar\ (gauge)\ T_1 = 25°C$. if temperature increases to 70°C What is the new pressure in the blank end.

Given Data:

$$p_1 = 2bar\ (gauge)\ \ V_1 = constant, T_1 = 25°C, T_2 = 70°C\ ,$$

Solution

Since the volume is constant, by applying Gay –Lussac's law, we get,

$$p_1 = 2bar\ (gauge) = 2 + 1 = 3\ bar(absoute)$$

$$T_1 = temperature\ in\ Kelvin = 25 + 273 = 298\ K,\ \ T_2 = 70 + 273 = 343\ K$$

$$\frac{p}{T} = constant\ or\ \frac{p_1}{T_1} = \frac{p_2}{T_2}$$

$$\frac{3}{298} = \frac{p_2}{343}$$

Solving we get, $p_2 = 3.45\ bar(absoulte) = 2.45\ bar(gauge)$

Example 1.8:. A compressed air receiver has a volume of 0.5 m^3 is filled with compressed air at 7 bar (gauge) and at a temperature of 40°C . The temperature then cools to 20°C. What is the final pressure?

Given data$V_1 = V_2 = 0.5m^3,\ p_1 = 7\ bar\ (gauge) = 7 + 1 = 8\ bar(absoulte)$

$T_1 = 20°\text{C} = 20 + 273 = 293\ K$

$T_2 = 40°\text{C} = 40 + 273 = 313\ K$

Since the volume is constant we can apply, Gay-Lussac law

$$\frac{p_1}{T_1} = \frac{p_2}{T_2}$$

$$\frac{8}{298} = \frac{p_2}{313}$$

Solving we get,

Thus, $p_2 = 7.5\ bar\ (abs) = \boldsymbol{6.5\ bar\ (gauge)}$

Example 1.9:.

A container is designed to hold a pressure of 2.5 atm. The volume of the container is 20.0 cm^3, and it is filled with air at room temperature (20°C) and normal atmospheric pressure. Would it be safe to throw the container into a fire where temperatures of 600°C would be reached?

Given data$V_1 = V_2 = 0.5m^3,\ p_1 = 0\ bar\ (gauge) = 0 + 1 = 1\ bar(absoulte)$

$T_1 = 20°\text{C} = 20 + 273 = 293\ K$

$T_2 = 600°\text{C} = 600 + 273 = 873\ K$

Since the volume is constant we can apply, Gay-Lussac law

$\frac{p_1}{T_1} = \frac{p_2}{T_2}$

$\frac{1}{293} = \frac{p_2}{873}$

Solving we get,

Thus, $p_2 = 2.98\ bar\ (abs) = \sim \boldsymbol{2\ bar\ (gauge)}$

This concept works in reverse, as well. For instance, if we subject a gas to lower temperatures than their initial state, the external atmosphere can actually force the container to shrink

1.5.4. General gas equation

For any given mass of gas undergoing changes of pressure, temperature and volume, the general gas equation can be used. By combining Boyle's law and Gay-Lussac's law we get,

$$\frac{p_1 V_1}{T_1} = \frac{p_2\ V_2}{T_2}$$

Example 1.10: .Gas at 80 bar (gauge) and 50 ℃ is contained in the 1290 cm^3cylinder of Figure 1.5. A piston compresses the volume to 1000cm^3 while the gas is heated to 120℃. what is the final pressure of the cylinder.

Given Data:

$p_1 = 80\ bar\ (gauge)\ \ V_1 = 1290\ cm^3\ ,\ \ V_2 = 1000\ cm^3\ \ T_1 = 50℃, T_2 = 120℃\ ,$

Solution

Since the volume is constant

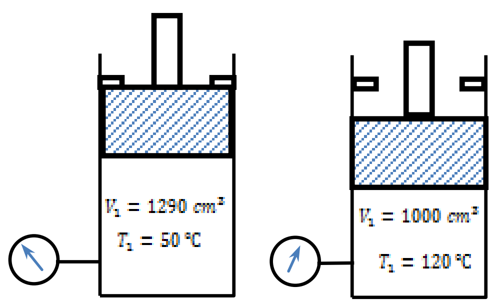


Figure 1.5

$p_1 = 80 bar(gauge) = 81\ bar(absoulte)$

$T_1 = 50℃ = 50 + 273 = 323\ K$

$T_2 = 120℃ = 120 + 273 = 393\ K$

$V_1 = 1290\ cm^3\ ,\ \ V_2 = 1000\ cm^3$

Using General gas law

$$\frac{p_1 V_1}{T_1} = \frac{p_2 \ V_2}{T_2}$$

$$\frac{81 \times 1290}{323} = \frac{p_2 \ \ \times 1000}{393}$$

Solving we get, $p_2 \ \ = 127.13 \ bar \ (absolute) = 126.17 \ bar(gauge)$

Example 1.11: The empirical formula of benzene is CH. When heated to 100°C in a flask whose volume was 247.2 ml, a sample of benzene vaporized and drove all air from the flask. When the benzene was condensed to a liquid, its mass was found to be 0.616 g. The barometric pressure was 742 mmHg. Calculate (a) the molar mass and (b) the molecular formula of benzene

Solution

Molar mass is mass divided by amount of substance. The latter quantity can be obtained from the volume, temperature, and pressure of benzene vapor

Part (a)

$$V = 247.2 \ cm^3 \times \frac{1 \ liter}{10^3 cm^3} = 0.2472 \ liter$$

$$\mathrm{T} = 273 + 100 = 373 \ \mathrm{K}$$

$$\mathrm{P} = 742 \ mm \ \mathrm{Hg} \times \frac{1 \ atm}{760 \ mm \ of \ \ Hg} = 0.976 \ atm$$

Using General gas law

$$n = \frac{pV}{RT} = \frac{0.976 \times 0.2472 \ liter}{0.0820 \ liter \ atm \ mol^{-1} K^{-1} \times 373K} = 7.89 \times 10^{-3} \ mol$$

Solving we get, $M = \frac{m}{N} \ = \frac{0.616 \ g}{7.89 \times 10^{-3} mol} = 78.1 \ g \ mol^{-1}$

Part (b)

The empirical formula CH would imply a molar mass of (12.0 +1.008) g mol-1 or 13.02 g mol-1, The experimentally determined molar mass is 6 times larger.

$$n = \frac{78.1 \ g \ mol^{-1}}{13.02 g \ mol^{-1}} = 6$$

So molecular formula must be C_6H_6

Example 1.12: .Two cubic meter of a gas at 5 bar and 300 K are compressed to a volume of 1 cubic m and then heated to a temperature of 420 K. what is the new gauge pressure.

Given Data:

$p_1 = 5\ bar\ (gauge)\ \ V_1 = 2\ m^3\ ,\ \ V_2 = 1\ m^3\ \ T_1 = 300\ K, T_2 = 420\ K\ ,$

Solution

Using General gas law

$$\frac{p_1 V_1}{T_1} = \frac{p_2\ \ V_2}{T_2}$$

$$\frac{6 \times 2}{300} = \frac{p_2\ \ \times 1}{420}$$

Solving we get, $p_2\ \ = 16.8\ bar\ (absolute) = 15.8\ bar(gauge)$

1.6 BASIC COMPONENTS OF PNEUMATIC SYSTEMS

Pneumatic system carries power by employing compressed gas generally air as a fluid for transmitting the energy from an energy-generating source to an energy – use point to accomplish useful work. Figure 1.6 shows the simple circuit of a pneumatic system with basic components.

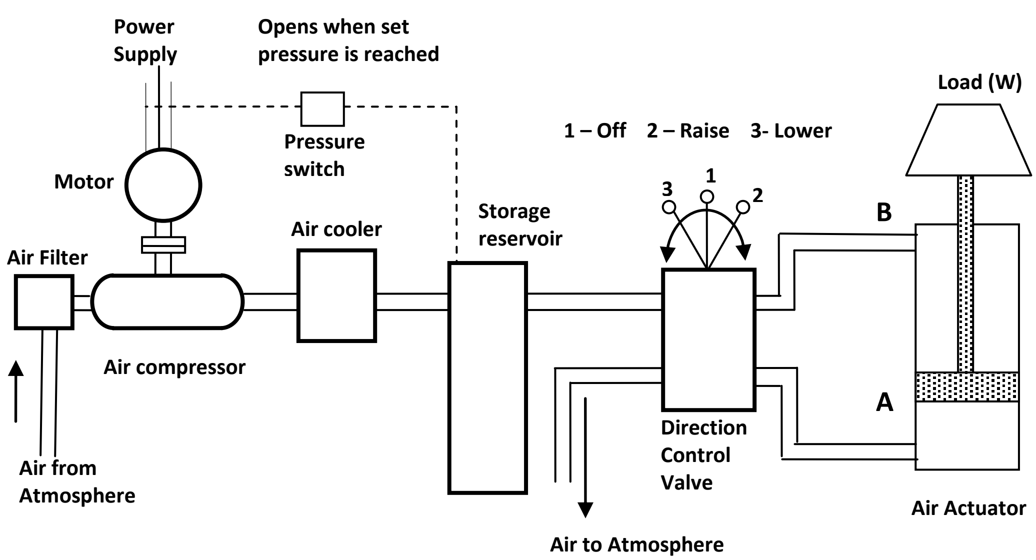


Figure 1.6 Components of Pneumatic System

Functions of components

- Pneumatic actuator converts the fluid power into mechanical power to do useful work
- Compressor is used to compress the fresh air drawn from the atmosphere.
- Storage reservoir is used to store a given volume of compressed air.
- Valves are used to control the direction, flow rate and pressure of compressed air.
- External power supply (Motor) is used to drive the compressor.
- Piping system carries the pressurized air from one location to another.

Air is drawn from the atmosphere through air filter and raised to required pressure by an air compressor. As the pressure rises, the temperature also rises and hence air cooler is provided to cool the air with some preliminary treatment to remove the moisture.

Then the treatment pressurized air needs to get stored to maintain the pressure. With the storage reservoir, a pressure switch is fitted to start and stop the electric motor when pressure falls and reached the required level, respectively.

The cylinder movement is controlled by pneumatic valve. one side of the pneumatic valve is connected to the compressed air and silencers for the exhaust air and the other side of the valve is connected to port A and Port B of the cylinder.

Position of the valve is as follows

1. **Raise:** To lift the weight , the compressed air supply is connected to port A and the port B is connected to the exhaust line, by moving the valve position to the " Raise"

2. **Lower:** To bring the weight down, the compressed air line is connected to port B and port A is connected to exhaust air line , by moving the valve position to the "lower"

3**. Off:** The weight can be stopped at a particular position by moving the valve to position to "Off" position. This disconnects the port A and port B from the pressurized line and the retrieval line, which locks the air in the cylinder.

Advantages of Pneumatic system

- Low inertia effect of pneumatic components due to low density of air.
- Pneumatic Systems are light in weight.
- Operating elements are cheaper and easy to operate
- Power losses are less due to low viscosity of air
- High output to weight ratio

- Pneumatic systems offers a safe power source in explosive environment
- Leakage is less and does not influence the systems. Moreover, leakage is not harmful

Disadvantages of Pneumatic systems

- Suitable only for low pressure and hence low force applications
- Compressed air actuators are economical up to 50 kN only.
- Generation of the compressed air is expensive compared to electricity
- Exhaust air noise is unpleasant and silence has to be used.
- Rigidity of the system is poor
- Weight to pressure ratio is large
- Less precise. It is not possible to achieve uniform speed due to compressibility of air
- Pneumatic systems is vulnerable to dirt and contamination

1.7 COMPARISON BETWEEN HYDRAULIC AND PNEUMATIC SYSTEM.

Usually hydraulic and pneumatic systems and equipment do not compete. They are so dissimilar that there are few problems of selection between then which cannot be readily be resolved, all factors considered, with a clear preponderance in favour of one or the other. But what of the case where one of the two resources of power is not at hand , and compressed air is used to do something for which oil might be better adopted, and vice versa. Certainly, availability is one of the important factors of selection by this may be outweighed by other factors. In numerous instances, for example, air is preferred to meet certain unalterable conditions. That is, hot spots where there is an open furnace or other potential ignition hazard or in operations where motion is required at extremely high speeds. It is often found more efficient to use a combined circuit in which oil is used in one part and air in another on same machine or process. Comparison between hydraulics and pneumatics is given in **Table 1.4**

Table 1.4 Comparison between Hydraulic and Pneumatic systems

Sl. No	Hydraulic system	Pneumatic system
1	It employs a pressurized liquid as fluid	it employs a compressed gas usually air as a fluid
2	Oil hydraulics system operates at pressures upto 700 bar.	Pneumatics systems usually operate at 5 to 10 bar.
3	Generally designed for closed systems	Pneumatic systems are usually designed as open system
4	System get slow down of leakage occurs	Leakage does not affect the system much more
5	Valve operations are difficult	Easy to operate the valves

6	Heavier in weight	Light in weight
7	Pumps are used to provide pressurized liquids	Compressors are used to provide compressed gas
8	System is unsafe to fire hazards	System is free from fire hazards
9	Automatic lubrication is provided	Special arrangements for lubrication needed.

1.8 COMPARISON OF DIFFERENT POWER SYSTEMS

There are three basic methods of transmitting power electrical, mechanical, and fluid power. Most applications actually use a combination of the three methods to obtain the most efficient overall system. To properly determine which method to use, it is important to know the salient features of each type. For example, fluid systems can transmit power more economically over greater distances than can are mechanical types. However, fluid systems are restricted to shorter distances than are electrical systems.

1.12.1 Electrical Power transmission

Electrical energy in the form of current is transmitted and distributed through wires and cables. The energy medium is controlled using switches, pushbuttons, relays, contactors, timers, sensors, pressure switches etc. Final actuator in this system is electric motor which converts electrical energy into mechanical energy in the form of rotary motion to perform useful work. Electrical systems are suitable for power transmission over long distances. The limitations of electric power include magnetic saturation, which may limit the torque capacity of the motor and material limitation, which may limit the speed and heat dissipation problems. Switching contacts may arc or corrode. Electric arc is a fire hazard in explosive environments. The response time of the electro mechanical solenoid is too slow for today’s controllers. The switching time of control elements is usually greater than 10 milliseconds. Electrical systems need to provide for heat dissipation as most systems which generate heat by pressure or friction.

1.12.2 Mechanical Power transmission

Mechanical power is transmitted by employing a variety of kinematic mechanisms such as belts, chains, pulleys, sprockets, gear trains, bar linkages and cams. They are suitable for the transmission of motion and force over relatively short distances. The disadvantages of mechanical power transmission include lubrication problems, limited speed and torque control capabilities, uneven force distribution, and relatively large space requirement

1.12.3 Hydraulic Power transmission

In hydraulic transmission of energy, a pump is used to raise the pressure of oil and energy stored in oil is transmitted through pipes and hoses to perform useful work. They are suitable for power transmission over intermediate distances; they can be employed over greater distances than mechanical types but not as electrical systems

Hydraulic systems are mechanically stiff, and can be designed to give fast operative and move very heavy loads. They can easily generate linear motion using liner actuators (also called cylinders). Speed control is easy and precise motion of the actuator is possible.

The disadvantages of hydraulic system include fluid leakage, containments and fire hazards with flammable hydraulic fluids.

1.12.4 Pneumatic Power transmission

All gases are readily compressible and it is this property which differentiates them most from liquids as a power transmission medium. In pneumatic transmission of energy, a compressor is used as the power source to raise the pressure of the air to the required level quite slowly. They are suitable for power transmission over intermediate distances. Pneumatic systems use simple equipment has small transmission lines, and do not present a fire hazard.

The disadvantages of pneumatic system include a high fluid compressibility and a small power to size ratio of components. Pneumatic systems are unsuitable for uniform motion. Operating pressure of pneumatics is around 6 to 8 bar. And hence are capable of generating only medium forces. The switching time of control elements is usually greater than 5 milli seconds and the speed of the control signal is 10 to 50 m/s. Table 1.5 give the comparison of all the systems.

Table 1.5 : Comparison of Electrical, mechanical, pneumatic and hydraulic transmission systems

Property	**Electrical/Mechanical**	**Pneumatic**	**Hydraulic**
Energy	IC engines, electrical energy is used to drive motors	Electrical energy is used to drive compressor and other equipments	I C Engines Electric Motor Air Turbine are used to drive hydraulic pumps.
Medium	. There is no medium, Energy is transferred through Levers, Gears, Shafts	Compressed air/gas in Pipes and hoses	Pressurized liquid in Pipes and hoses
Energy storage	Batteries	Reservoir, air tank	Accumulators
Regulations	Variable frequency drives	Pneumatic valves	Hydraulic valves
Transmitters	Transmitted through mechanical components like levers, gears, cams, screw cts	Transmitted through pneumatic cylinders , rotary drives and rotary actuators	Transmitted through hydraulic cylinders, and hydraulic rotary actuators.
Distribution system efficiency	Good	Limited (say up to 1000m)	Good (say up to 100 m)
Operating speed	Low	Limited (up to 1.5 m/s)	Limited (up to 0.5 m/s)

Positioning accuracy	Precision in terms of few micron can be achieved	. Precision in terms of few mm (usually 0.1 mm) can be achieved	Precision in terms of few micron can be achieved
Stability	Good stability is possible using mechanical elements	. Low stability is due to High compressibility of air	Good stability is possible due to low compressibility
Forces	Mechanical elements break down if overloaded. Poor overloading capacity	Protected against overload with system pressure of 6 to 8 bar, forces up to 50 kN can be generated.	Protected against overload with high system pressure of 600 bar, very large forces can be generated.
Cost of energy	Lowest	Highest	Medium
Linear actuators	Short stroke length using mechanical/electrical transmission elements	Is possible using pneumatic actuators (cylinders), it can produce medium force.	Is possible using hydraulic actuators(cylinders), and it can produce heavy force
Rotary actuators	AC , DC, Servo motors and steeper motors can be used	Pneumatic rotary actuators can be used	Hydraulic motors and vane motors can be used
Controllable force	Possible with solenoid and DC motors, Needs cooling and hence complicated	medium force can be controlled easily	High force can be controlled
Work environment	Danger because of electric shock	Noise	Dangerous, unsightly and fire hazardous because of leakage.

Objective Type Questions

1. When the power requirement is ------- like in forging presses, sheet metal press oil hydraulic system is preferred

2. Pneumatic systems have --------output to weight ratio, whereas electrical systems have ------- output to weight ratio

3. Positioning accuracy of pneumatic actuator is -------------- compared to hydraulic systems

4. Compressed air actuators are economical up to ----- force only.

5. Low inertia effect of pneumatic components due to ----- density of air.

State True or False

1. Pneumatic systems are used for high pressure and low speed applications

2. Pneumatics is very useful in hazardous environment

3. Compressed air is expensive

4. According to Barometric law, the atmospheric pressure increases exponentially with the increase in altitude

5. If a given mass of a gas is compressed or expanded at a constant temperature, then the absolute pressure is directly proportional to the volume

Review Questions

1. Define Pneumatics

2. List the broad rules followed in the selection of a working medium.

3. List the advantages and disadvantages of compressed air.
4. Why air is used as fluid medium in pneumatic systems

5. List seven reasons for considering pneumatics instead of hydraulic system.

6. Name five characteristics of pneumatic systems

7. List twenty applications of pneumatics

8. Differentiate between Free air and standard air

9. Differentiate between Absolute pressure and atmosphere pressure

10. Discuss various pressure ranges used in pneumatic applications

11. State the following perfect gas laws

a) Boyle's Law b) Charles Law c) Gay-Lussac law d) General gas law

12. Name six basic components used in a pneumatic systems.

13. With a simple sketch, explain the functions and working of basic components required for a pneumatic system.

14. List five advantages and five disadvantages of pneumatics

15. Differentiate between hydraulics and pneumatics

16. Compare and contrast between hydraulic, pneumatic and electrical power transmission systems.

Answers

Fill in the Blanks

1. High
2. High/low
3. Low
4. 50 kN
5. Low

State True or False

1. False
2. True
3. True
4. False
5. False

Lecture 34

PREPARATION OF COMPRESSED AIR.

Learning Objectives

Upon completion of this chapter, Student should be able to

- Explain the various stages of air preparation
- Describe the working of various compressors
- List the advantages and disadvantages of various compressors
- Carry out thermodynamic analysis of compressors
- Compare various types of compressors
- List various ways to control compressor
- Understand the selection criteria for compressor
- List various hazardous of compressed air.

1.1 AIR PREPARATION

Pneumatic control systems operate on a supply of compressed air, which must be made available in sufficient quantity and at a pressure to suit the capacity of the system. The operational reliability and service life of a pneumatic system depend to a large extent on the preparation of the compressed air. Impurities in the compressed air such as scale, rust and dust as well as the liquid constituents in the air which deposit as condensate can cause a great deal of damage in pneumatic systems. These contaminants accelerate wear on sliding surfaces and sealing elements, adversely affecting the functioning and service life of pneumatic components. As a result of switching the compressors on and off, pressure fluctuations occur which have an unfavourable effect on the functioning of the system. In order to eliminate these effects, compressed air preparation should be given utmost importance. There are four distinct stages of air preparation they are:

Stage 1 : This consist of air intake system

Stage 2: This stage consist of compressors, with drives controls, inter-cooling, compressor cooling, waste heat recovery and air inlet filtration

Stage 3: This stage includes Conditioning equipment, consisting of air receivers, after coolers, separators, traps (also frequency called drain traps or drains) , filters and air dryers

Stage 4: This stage consist of air distribution subsystems, including main trunk lines, drops to specific usage, valving, additional filters and traps(drains) , air hoses, possible supplement air conditioning equipment, connectors, often pressure regulators and lubricator.

Stage 1 : An Intake filter removes larger particles which can damage the air compressor.

a) Location: The intake for a compressor will located either outdoors or indoors, whichever provides the better air quality. Elevation of the compressor relative to sea level is required to determine the atmospheric pressure and density of intake air. Air quality is judged by its temperature, humidity and cleanliness. We must ensure that air intage is free of moisture or pollution.

b) **Intake Temperature:** The density of air varies inversely with its temperature : an increase in delivery of approximately 1 percent is gained for -20 ℃ reduction of intake temperature.

c) **Intake pipe material**: The inside of intake piping must be smooth and not subject to rusting or oxidation. Rust that flakes off will enter and damage the compressor.. Acceptable intake air piping materials include plastic, cooper, stainless steel, aluminium or galvanized steel. On metallic piping, mechanical couplings will be used. Welded joint must be avoided since weld beads can break free, enter and damage the compressor.

d) **Critical pipe length:** resonance of intake piping will reciprocating air compressor is prevented by avoiding certain pipe lengths. These are called critical pipe lengths, and are a function of the air temperature and the speed of the compressor in revolutions per minute. Critical pipe lengths must be verified with equipment manufacturers.

e) **Intake air filter:** The selection of filter type is based on whether air compressor to be used is lubricated or non lubricated, and on the quality of ambient air.

- Viscous impingement filters have an efficiency of 85 to 90 percent of particle size larger than 10 microns. This type of filter is acceptable for lubricated reciprocating compressor operating under normal conditions
- Oil bath filters have an efficiency of 96 to 98 percent of particle sized larger than 10 microns. This type of filter is more expensive, and for the most part no longer recommended by compressor manufacturers, but may be considered for lubricated reciprocating compressor operating under heavy duty conditions.

- Dry filters have an efficiency of 99 percent of particles larger than 10 microns. Because of their high filtration efficiency, these filters are the best selection for rotary and reciprocating compressors. They must be used for non-lubricated compressors and whenever air must be kept oil free.

- Two stage dry filters , to provide 99 percent efficiency of particles larger than 0.3 micron, will be used for centrifugal units

- With all types of filters, a means of monitoring the air pressure drop through the element must be provided, which indicates element contaminations.

Stage 2: In this stage air is compressed using compressor. This book is not meant to be a comprehensive analysis of all types of air compression system that can be designed. Instead, it will concentrate on those most often found in industry and on thermodynamic analysis in those systems. It will explore positive displacement types in great detail and dynamic compressor in brief.

Stage 3: In this stage outlet temperature at the compressor is reduced, solid contaminants usually large than 100 micron are removed, and air is dried to reduce to its humidity. The units used in the primary stage are after cooler, main line filter and dryer.

Stage 4: In this stage moisture and fine dirt particles are removed. In this stage pressure is regulated to suit individual machine's requirement and introduces the fine mist of oil to the compressed air to aid lubrication. The units used in secondary air treatment are filter , regulator and lubricator (Called FRL or service units)

Figure 1.1 shows all four stages of air preparation. Figure 1.2 illustrates a typical compressed air system.

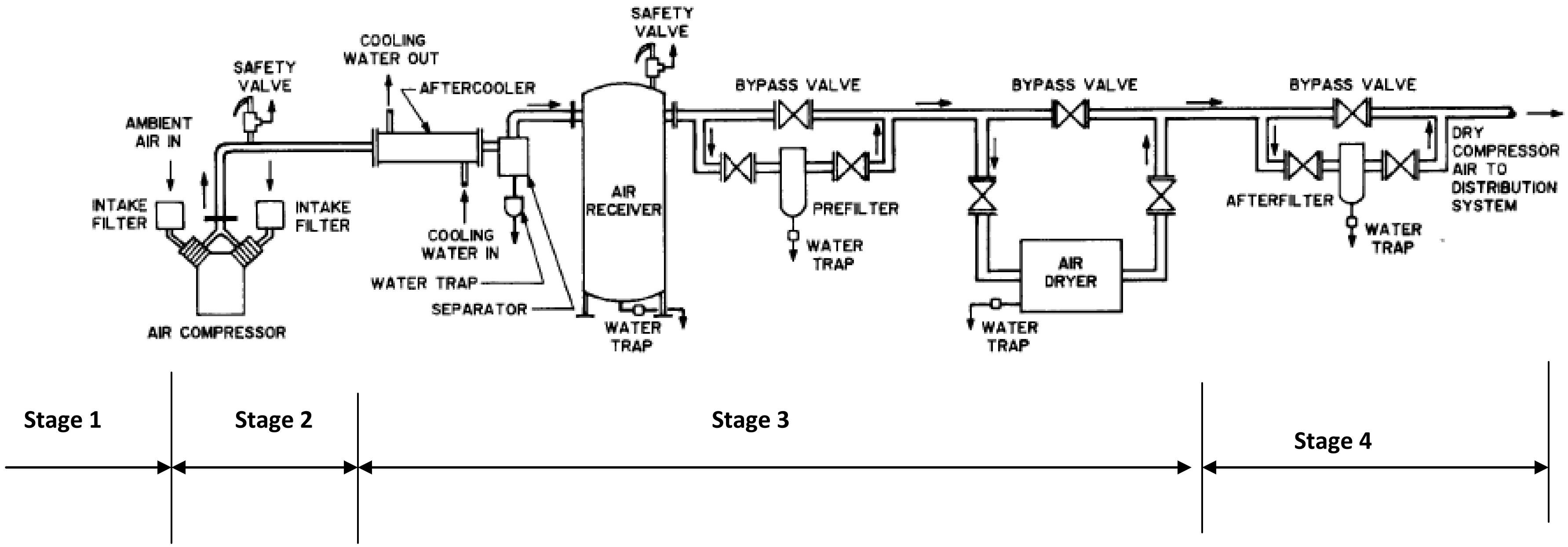


Figure 1.1 Four stages of air preparation

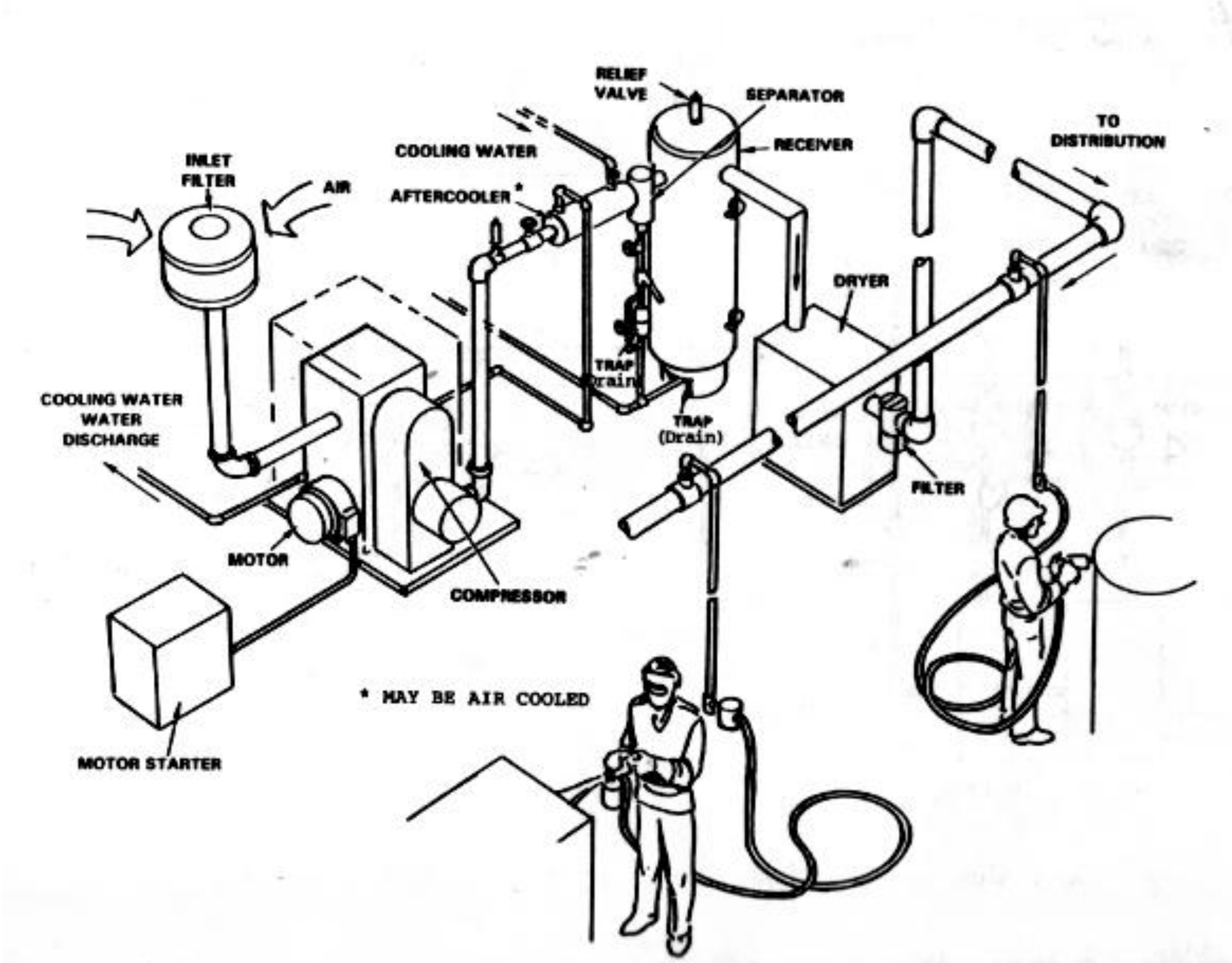


Figure 1.2 An Industrial compressed air system.

1.2 AIR COMPRESSORS: HISTORY AND ITS CLASSIFICATION

The first air compressor were human lungs; by blowing on cinders man started his fires. Then with birth of metallurgy man began to melt metal and high temperatures were needed. A more powerful compressor was required.

One of the earliest recorded uses of compressed gas (air) dates back to 3rd century B.C. This early use of compressed air was the "water organ." The invention of the "water organ" is commonly credited to Ctesibius of Alexandria. Ctesibius also developed the positive displacement cylinder and piston to move water. The water organ consisted of a water pump, a chamber partly filled with air and water, a row of pipes on top (organ pipes) of various diameters and lengths plus connecting tubing and valves. By pumping water into the water/air chamber the air becomes compressed. This concept was further improved by Hero of Alexandria (also noted for describing the principles of expanding steam to convert steam power to shaft power).

The first mechanical compressor, the hand-operated bellows, emerged in 1500 B.C. In the 1850s, while trying to find a replacement for the water wheel at their family's woollen mill, Philander and Francis Roots devised what has come to be known as the Roots blower. Their design consisted of a pair of figure-eight impellers rotating in opposite directions. While some Europeans were simultaneously experimenting with this design, the Roots brothers perfected the design and put it into large-scale production.

In 1808 John Dumball envisioned a multi-stage axial compressor. Unfortunately his idea consisted only of moving blades without stationary airfoils to turn the flow into each succeeding stage. Not until 1872 did Dr. Franz Stolze combine the ideas of John Barber and John Dumball to develop the first axial compressor driven by an axial turbine. Due to a lack of funds, he did not build his machine until 1900. Dr. Stolze's design consisted of a multi-stage axial flow compressor, a single combustion chamber, a multistage axial turbine, and a regenerator utilizing exhaust gases to heat the compressor discharge gas.

A Compressor is a machine that compresses the air or another type of gas from a low inlet pressure (usually atmospheric pressure) to a higher desired pressure level. Compressor increases the pressure of the air by reducing its volume. Work required for increasing pressure of air is available from the prime mover driving the compressor. Generally, electric motor, internal combustion engine or steam engine, turbine etc. are used as prime movers. Compressors are similar to fans and blowers but differ in terms of pressure ratios. Fan is said to have pressure ratio up to 1.1 and blowers have pressure ratio between 1.1 to 4 while compressors have pressure ratios more than 4.

Compressors can be classified in the following different ways.

(a) **Based on principle of operation:** Based on the principle of operation compressors can be classified as.

(i) Positive displacement compressor.

(ii) Non-positive displacement compressors.

In positive displacement compressors the compression is realized by displacement of solid boundary and preventing fluid by solid boundary from flowing back in the direction of pressure gradient. Due to solid wall displacement these are capable of providing quite large pressure ratios. Positive displacement compressors can be further classified based on the type of mechanism used for compression. These can be

(i) Reciprocating type positive displacement compressors

(ii) Rotary type positive displacement compressors.

Reciprocating compressors generally, employ piston-cylinder arrangement where displacement of piston in cylinder causes rise in pressure. Reciprocating compressors are capable of giving large pressure ratios but the mass handling capacity is limited or small. Reciprocating compressors may also be single acting compressor or double acting compressor. Single acting compressor has one delivery stroke per revolution while in double acting there are two delivery strokes per revolution of crank shaft. Rotary compressors employing positive displacement have a rotary part whose boundary causes positive displacement of fluid and thereby compression. Rotary compressors of this type are available in the names as given below;

(i) Roots blower

(ii) Vane type compressors

Rotary compressors of above type are capable of running at higher speed and can handle large mass flow rate than reciprocating compressors of positive displacement type.

Non-positive displacement compressors also called as steady flow compressors use dynamic action of solid boundary for realizing pressure rise. Here fluid is not contained in definite volume and subsequent volume reduction does not occur as in case of positive displacement compressors. Non-positive displacement compressor may be of 'axial flow type' or 'centrifugal type' depending upon type of flow in compressor.

(b) **Based on number of stages:** Compressors may also be classified on the basis of number of stages. Generally, the number of stages depends upon the maximum delivery pressure. Compressors can be single stage or multistage. Normally maximum compression ratio of 5 is realized in single stage compressors. For compression ratio more than 5 the multistage compressors are used.

Type values of maximum delivery pressures generally available from different type of compressor are,

(i) Single stage Compressor, for delivery pressure upto 5 bar.

(ii) Two stage Compressor, for delivery pressure between 5 to 35 bar

(iii) Three stage Compressor, for delivery pressure between 35 to 85 bar.

(iv) Four stage compressor, for delivery pressure more than 85 bar

(c) **Based on Capacity of compressors**: Compressors can also be classified depending upon the capacity of Compressor or air delivered per unit time. Typical values of capacity for different compressors are given as;

(i) Low capacity compressors, having air delivery capacity of 0.15 m^3/s or less

(ii) Medium capacity compressors, having air delivery capacity between 0.15 to 5 m^3/s.

(iii) High capacity compressors, having air delivery capacity more than 5 m^3/s

(d) **Based on highest pressure developed:** Depending upon the maximum pressure available from compressor they can be classified as low pressure, medium pressure, high pressure and super high pressure compressors. Typical values of maximum pressure developed for different compressors are as under:

(i) Low pressure compressor, having maximum pressure upto 1 bar

(ii) Medium pressure compressor, having maximum pressure from 1 bar to 8 bar

(iii) High pressure compressor, having maximum pressure from 8 to 10 bar

(iv) Super high pressure compressor, having maximum pressure more than 10 bar.

Detailed classification is given in the Figure 1.3. Air compressors are generally positive displacement units and either of reciprocating piston type or the rotary screw or rotary vane types. These three types are explained in detail.

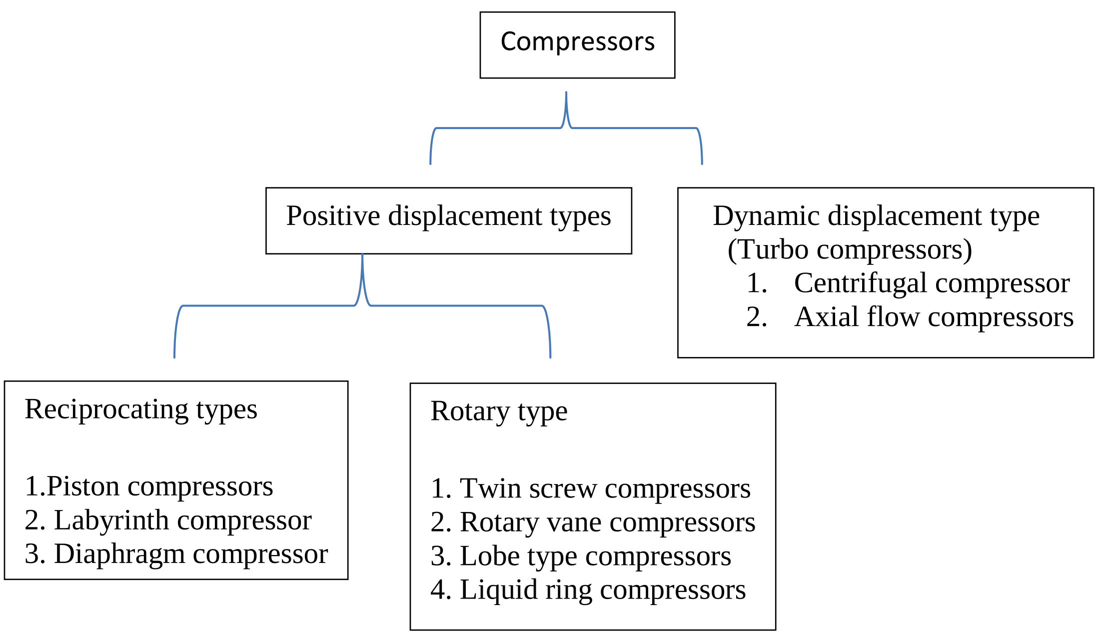


Figure 1.3 Classification of Compressors

Piston type of compressors are used commonly in Industries. Therfore only detailed discussion on piston type of compressor is presented in this chapter.

1.2.1 RECIPROCATING COMPRESSORS

Reciprocating compressors have been the most widely used for industrial plant air systems. The two major types are single acting and double acting, both of which are available as one or two stage compressors. The Single acting cylinder performs compression on one side of the piston during one direction of the power stroke. Two stage compressions reach the final output pressure in two separate compression cycles, or stages, in series.

The double acting compressor is configured to provide a compression stroke as the piston moves in either direction. This is accomplished by mounting a cross head on the crank arm which is then connected to a double acting piston by a piston rod. Distance pieces connect the cylinder to the crankcase. They are sealed to prevent mixing of crank shaft lubricant with the air, but vented so as to prevent pressure built up.

1.2.1.1 PISTON COMPRESSORS

Piston type compressors are the oldest and most commonly used compressor in the pneumatic industry because of its flexibility, high pressure capability, ability to rapidly dissipate heat of compression and oil free. They are built for either stationary or portable services.

A. SINGLE CYLINDER COMPRESSOR

Piston compressors are available as single or double acting, oil lubricated or oil free with different number of cylinders in different configurations. With the exception of really small compressors with vertical cylinders, the V configuration is the most common for small compressors. On double acting, large compressors the L type with vertical low pressure cylinder and horizontal high pressure cylinder, offer immense benefits and is why this the most common design. The construction and working of a piston type reciprocating compressor is very much similar to that of an internal combustion engine**.**

a) Construction: Piston type compressor consists of cylinder, cylinder head, and piston with piston rings, inlet and outlet spring loaded valves, connecting rod, crank crankshaft and bearings.

b) Operation

Compression is accomplished by the reciprocating movement of a piston within a cylinder. This motion alternately fills the cylinder and then compresses the air. A connecting rod transforms the rotary motion of the crankshaft into the reciprocating motion of piston in the cylinder. Depending on the application, the rotating crank (or eccentric) is driven at constant speed by a suitable prime mover (usually electric motor). Schematic diagram of single cylinder compressor is shown in Figure 1.4

Inlet stroke: -suction or inlet stroke begins with piston at top dead centre (a position providing a minimum or clearance volume). During the downward stroke, piston motion reduces the pressure inside the cylinder below the atmospheric pressure. The inlet valve then opens against the pressures of its spring and allows air to flow into the cylinder. The air is drawn into the cylinder until the piston reaches to a maximum volume position (bottom dead centre).The discharge valve remains closed during this stroke

Outlet stroke: During compression stroke piston moves in the opposite direction (Bottom dead centre to top dead centre), decreasing the volume of the air. As the piston starts moving upwards, the inlet valve is closed and pressure starts to increase continuously until the pressure inside the cylinder is above the pressure of the delivery side which is connected to the receiver. Then the outlet valve opens and air is delivered during the remaining upward motion of the piston to the receiver.

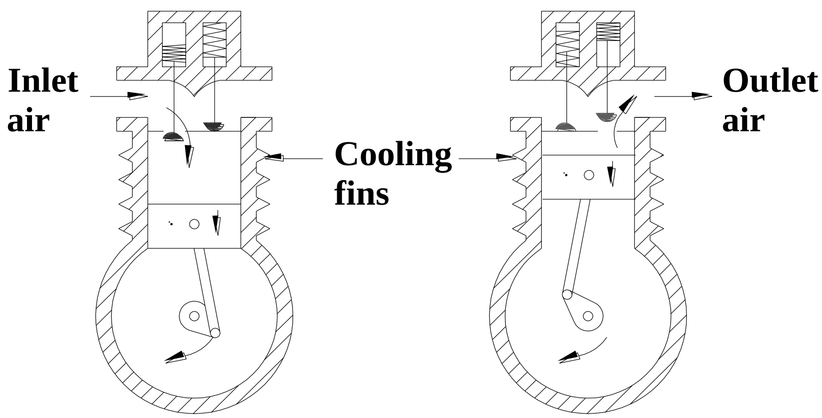


Figure 1.4 Single cylinder compressors

B. ANALYSIS OF SINGLE CYLINDER SINGLE STAGE AIR COMPRESSOR

A typical indicator diagram for reciprocating compressor with three different types of compression is shown in the Figure 1.5. Clearance volume is neglected.

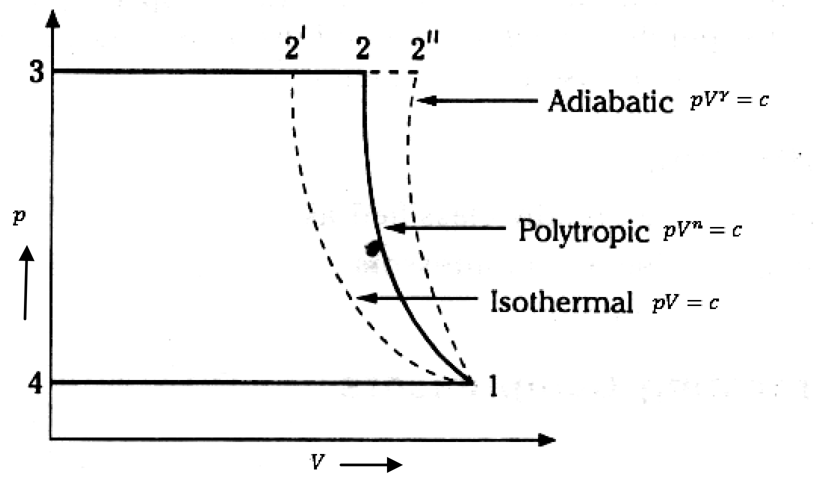


Figure 1.5 Types of compression

Constant pressure line 4-1 represents the suction stroke. The air is then compressed adiabatically (process line 12”,) and is then forced out of the cylinder at constant pressure (process 2”3). Area 12”34 represents the work. If the compression is carried out isothermally, then it follows the curve 12’ which has less slope than both isentropic and polytrophic processes. This work done that is area 12’34 in isothermal process is considerably less than that due to adiabatic compression. Thus compressor will have higher efficiency if compression follows isothermal process. It is not possible in practice as to achieve isothermal process, as the compressor must run very slowly. In practice compressors run at high speeds which results in polytropic process. The cold water spray and multi stage compression are used for approximating to isothermal compression while still running the compressor at high speeds.

C.WORK DONE IN A SINGLE STAGE COMPRESSOR NEGLECTING CLEARANCE.

Figure 1.6 shows the PV diagram of the air in the cylinder of an air compressor. Constant pressure line **ab** represents the suction stroke. The air is then compressed adiabatically (process line **bc**,) and is then forced out of the cylinder at constant pressure (process **cd**). Area abcd represents the work.

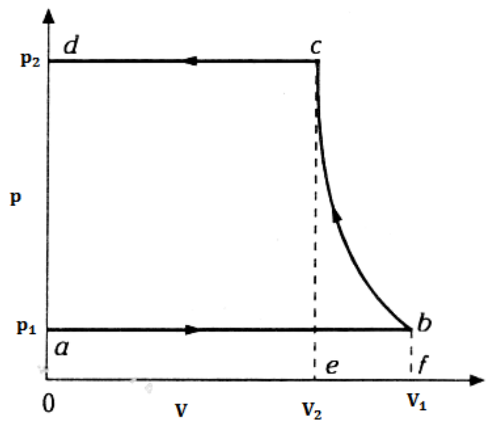


Figure 1.6 PV diagram

There are three types of compression processes possible in compressor they are

Isothermal compression. Compression of air takes place at constant temperature

$$\text{work done during compression} = p_1 V_1 \ln\left(\frac{p_2}{p_1}\right) \text{---------} (1.1)$$

Where p_1 = inlet pressure , p_2 = outlet pressure

Adiabatic compression. There is no flow of heat energy into or out of the gas during expansion or compression.

$$\text{work done during compression} = \frac{\gamma}{\gamma-1}(p_2 V_2 - p_1 V_1) \text{-----} (1.2)$$

$$workdone\ during\ compression = \frac{\gamma\, p_1 V_1}{\gamma - 1}\left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right]$$

γ *is the ratio of specific heat* $= 1.4$ *for air*

$$c_p = 1.005 \frac{kJ}{kg} k, c_v = 0.718 \frac{kJ}{kg} k$$

Polytrophic compression This process lies between Isothermal and adiabatic. In pneumatics, most compression/expansions are neither adiabatic (Very fast) nor isothermal (Very slow). It is polytrophic

$$workdone\ during\ compression = \frac{n}{n-1}(p_2 V_2 - p_1 V_1)$$

$$workdone\ during\ compression\ = \frac{n\, p_1 V_1}{n-1}\left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right]$$

$n\ is\ the\ polytropic\ = 1.4\ for\ air$

Efficiency of compressor In a reciprocating compressor the work is minimum when compression follows the isothermal process. The ratio of isothermal work done to the actual work done is called isothermal efficiency.

$$\eta_{isothermal} = \frac{Isothermal\ work}{actual\ work}$$

D.**WORK DONE IN A SINGLE STAGE COMPRESSOR CONSIDERING CLEARANCE.**

In practical design of compressors, some clearance is required between the cylinder and piston to prevent hitting of piston to crown of the cylinder. Figure 1.7 shows a PV diagram of single stage compressor with clearance.

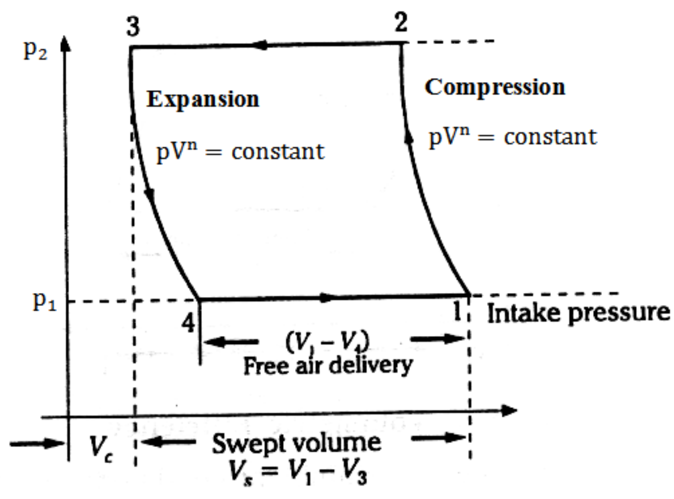


Figure 1.7 PV diagram with clearance

Thus when the compressed air is delivered during the delivery stroke, some amount of air corresponding to clearance volume V_3 at a pressure p_2 will be left over in the cylinder. During the next suction stroke this air expands back to initial pressure p_1 and volume V_4. Thus before the fresh air enters the cylinder some air corresponding to volume V_4 will be already there in the cylinder. This the volume inhaled during the suction stroke will be $V_1 - V_4$ which is less the swept volume V_s

The work done on the air delivered is not affected by the clearance volume as the work required to compress the clearance volume is theoretically regained during its expansion from $V_3\ to\ V_4$. Thus the work done is given by

$$workdone\ during\ compression\ = \frac{n\, p_1(V_1 - V_4)}{n-1}\left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right]$$

E. **Volumetric efficiency**

Free air delivery It is the amount of atmospheric air that can be sucked by the compressor at suction or inlet condition of compressor at one atmospheric pressure, 20 ℃, 100 percent dry air and compressor motor running at 100 % of the rated value. FAD is an important purchasing parameter and it measures the capacity of a compressor in terms of air flow it can handle. FAD is used to compare different compressors. It is important to note that induced mass per cycle must equal the delivered mass per cycle as per law of conservation of mass, although the induced and delivered volumes will be different.

$$\eta_{volumetric} = \frac{Actual\ volume\ of\ air\ taken\ referred\ to\ free\ air\ conditions}{swept\ volume\ of\ the\ compressor}$$

$$\eta_{volumetric} = \frac{p_1(T_a)}{p_a T_1}\left[1 + \mathrm{k} - \mathrm{k}\left(\frac{p_2}{p_1}\right)^{\frac{1}{n}}\right]$$

Where $\mathrm{k} = \left(\frac{V_c}{V_s}\right) = \text{clearance ratio}$

Subscript "a" refers to free air or ambient conditions and subscript "1" refer to the condition before compression.

F. Analysis of Air capacity rating of compressors

Air compressors are rated in terms of m^3/min of free air, defined as air at actual atmospheric conditions. Where standard atmospheric condition are 101000 Pa (absolute) and 20 ℃

In Industry we still use British system for air rating of compressors. Air compressors are rated in terms of CFM of free air, defined as air at actual atmospheric conditions. CFM of free air is called SCFM when the compressor inlet air is at standard atmospheric condition of 17.7 (psia) 1 bar and 68F

$Q_1 and\ Q_2 = volume\ flow\ rate\ of\ air\ at\ the\ compress\ inlet\ and\ out\ let\ \ (m^3/min)$
$p_1 and\ p_2 = absoulte\ pressure\ of\ air\ at\ the\ compressor\ inlet\ and\ outlest\ \ (kPa(abs)$
$T_1 and\ T_2 = Absolute\ temperature\ of\ air\ at\ the\ compressor\ \ at\ inlet\ and\ out\ let\ \ (k)$
Using general gas law

$$\mathrm{Q}_1 = \mathrm{Q}_2 \left(\frac{p_2}{p_1}\right)\left(\frac{T_1}{T_2}\right)$$

Example 1.1: A compressor delivers 4 m^3 of the free air per minute at a pressure of 7 bar gauge. Assuming that the compression follows the law $p\,V^{1.3} = Constant$, determine the theoretical work done.

Given data

$n\ is\ the\ polytropic\ = 1.3\ for\ air$

$V_1 = 4\,m^3, p_1 = 1\ bar\ (absolute)$

$p_2 = 7\ bar\ (gauge) = 8\ bar\ (absoulte)$

Solution

$$p_1 V_1^{1.3} = p_2 V_2^{1.3}$$

$$V_2 = V_1 \left(\frac{p_1}{p_2}\right)^{1.3}$$

$$V_2 = 4\left(\frac{1}{8}\right)^{1.3} = 0.807\ m^3/min$$

$$workdone\ during\ compression\ = \frac{n}{n-1}(p_2V_2 - p_1V_1)$$

$$workdone\ during\ compression\ = \frac{1.3}{0.3}(8 \times 0.807 - 1 \times 3) \times 10^5$$

Solving we get Work done = 14.98$\times 10^5$ N.m/minute = 0.2496 $\times 10^5$N.m/s = 0.25$\times 10^5$ Watts =25 kW

Example 1.2: A single stage air compressor running at 80 RPM, compress air from a pressure of 1 bar and temperature of 15 °C to a pressure of 5 bar (see Figure 1.8) The clearance volume is 5 % of swept volume which is 0.42 m^3Assuming that the compression and expansion to follow the law $pV^{1.3}$ = Constant,determine the power required to drive the compressor

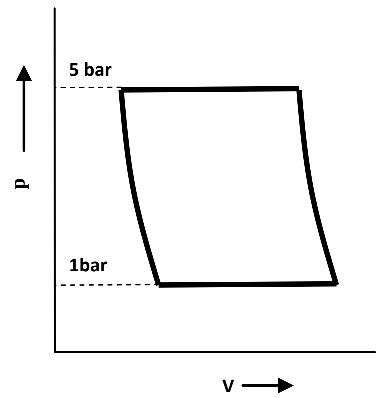


Figure 1.8

Given data

$n\ is\ the\ polytropic\ = 1.3\ for\ air$

$\mathrm{N} = 80\ \mathrm{RPM}\ , p_2 = 5 bar\ (absolute)$

$p_1 = 1\ bar\ (absoulte)\ T_1 = 15\ °\mathrm{C}$

Solution

$$\mathrm{k} = \left(\frac{V_c}{V_s}\right) = \frac{5}{100} = 0.05$$

$V_s\ = 0.42\ m^3$

Volumetric efficiency referred to the suction conditions.

$$\eta_{volumetric} = \left[1 + \mathrm{k} - \mathrm{k}\left(\frac{p_2}{p_1}\right)^{\frac{1}{\mathrm{n}}}\right] = \left[1 + 0.05 - 0.05\left(\frac{5}{1}\right)^{\frac{1}{1.4}}\right] = 0.8775$$

$$\eta_{volumetric} = \frac{Actual\ volume\ of\ air\ taken\ referred\ to\ free\ air\ conditions}{swept\ volume\ of\ the\ compressor}$$

$$0.8775 = \frac{V_1}{0.42}$$

$$V_1 = 0.3685\ \mathrm{m}^3/\mathrm{cycle}$$

$$Mass\ of\ air = \frac{p_1 V_1}{RT_1} = \frac{0.3685 \times 1 \times 100}{0.287 \times 288} = 0.445\ kg/cycle$$

$$Mass\ of\ air = \frac{0.445 \times 80}{60} = 0.5933\ kg/sec$$

$$T_2 = T_1\left(\frac{p_2}{p_1}\right)^{\frac{\mathrm{n}-1}{\mathrm{n}}} = 288(5)^{\frac{0.3}{1.3}} = 417.53\ \mathrm{K}$$

$$power = \frac{n}{n-1}\mathrm{mR}(T_2 - T_1)$$

$$power = \frac{1.3}{1.3-1} \times 0.5933 \times 0.287 \times (417.53 - 288) = 95.5\ \mathrm{kW}$$

C. **MULTI STAGE PISTON COMPRESSOR.**

As per general gas laws, if the pressure increases temperature also increases. For example : if the exit pressure of compressor is 5 bar in a single acting compressor, the compressor air temperature can rise to over 200 ℃ and the motor power needed to drive the compressor rises. Therefore single stage compressors are not used for high pressures. Multistage compressors are used when high pressures are required, because better cooling between stages can effectively increase the efficiency and reduce the input power requirements.

Single stage machines compress the air to pressure of about 6 bars and in exceptional cases to 10 bars, two stage machines normally discharge pressure up to 15 bars. Discharge pressures in the range of 250 bars can be obtained with high pressure reciprocating compressor of three and four stages.

In single stage compressor, entire compression of air takes place in single stroke of the piston. In multi stage compressor, compression takes in stages. For maximum compressor efficiency, it is desirable to cool air after one stage using inter- stage cooler. In two stage compressor, initial compression takes place in the low pressure cylinder. Air from this stage (low pressure cylinder) is passed through the inter cooler to reduce the temperature. Then the cooled air is compressed in the high pressure cylinder.

Working:

Figure 1.11 shows the two stage (inline type) reciprocating air compressor. When the prime mover connected to crank shaft rotates, crank rotates and the piston in the first stage reciprocates. It sucks the air through the suction filter and inlet valve. The air, compressed to a certain degree passes from the left cylinder to right cylinder through the intermediate cooler. The compression ratio in the first stage is determined by the degree of cooling required.

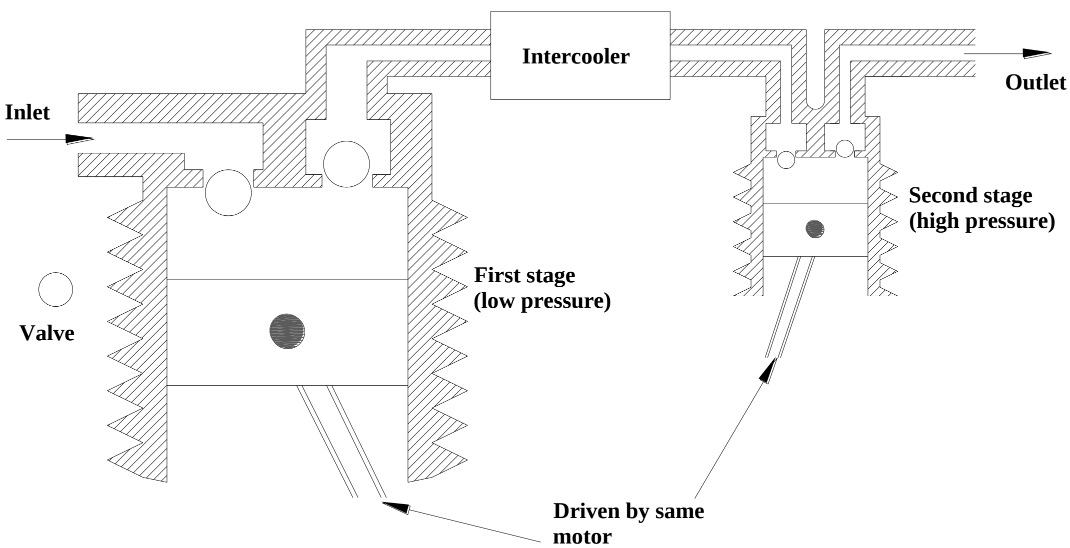


Figure 1.11 Multi stage piston air compressor with intercooler

Figure 1.12 shows various parts of three stage (V type) reciprocating air compressor with receiver (air tank). The pressure switch is connected to the electric motor. When the desired pressure in the air tank is reached it stops the motor and hence the compressor. The safety valve opens when the pressure in air tank exceeds the set safe pressure.

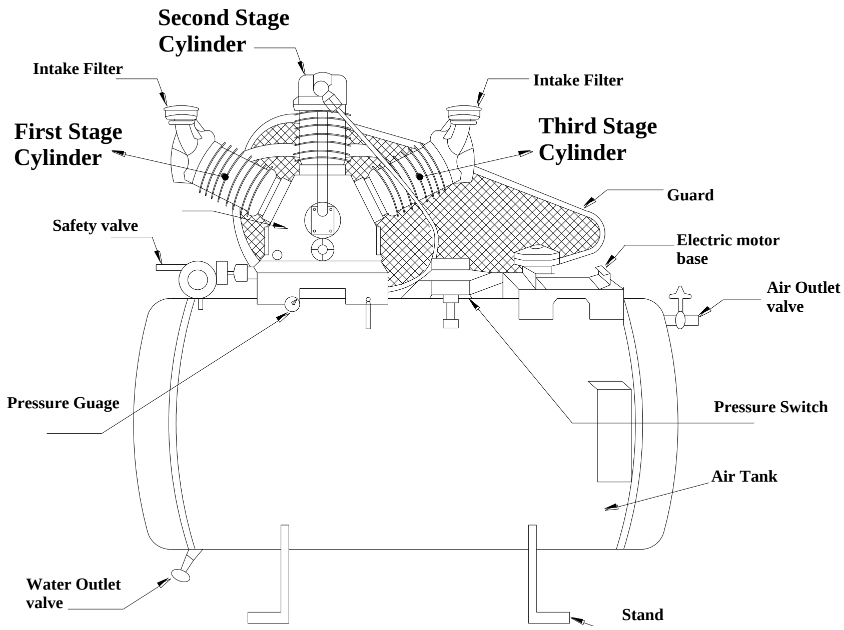


Figure 1.12 Various parts of three stage compressor

The drain valve drains the condensate produced at the condenser and the receiver. Cylinders and intercoolers are either air cooled (with fins) or water cooled (with water jackets in the cylinder). Air cooled compressor are used for low pressure applications and water cooled compressors are used for high pressure applications.

Range: Used of pressures up to 4-30 bar and low delivery volumes (< 10000 m^3/h). For pressures exceeding 30 bar multi stage compressors are required. The multi stage compressors are available with pressure up to 250-350 bar.

Advantages of piston type compressor

1. Piston type compressors are available in wide range of capacity and pressure
2. Very high air pressure (250 bar) and air volume flow rate is possible with multi-staging.

3. Better mechanical balancing is possible by multistage compressor by proper cylinder arrangement.
4. High overall efficiency compared to other compressor

Disadvantages of piston type compressor

1. Reciprocating piston compressors generate inertia forces that shake the machine. Therefore , a rigid frame, fixed to solid foundation is often required
2. Reciprocating piston machines deliver a pulsating flow of air. Properly sized pulsation damping chambers or receiver tanks are required.
3. They are suited for small volumes of air at high pressures.

B. **ANALYSIS OF MULTI SINGLE STAGE AIR COMPRESSOR**

The volumetric efficiency of a reciprocating compressor is a function of a clearance ratio, the pressure ratio and index of expansion. As the pressure ratio is increased, the volumetric efficiency of a compressor having a fixed clearance decreases and finally a stage may be reached when the volumetric efficiency may be zero as see from Figure 1.13

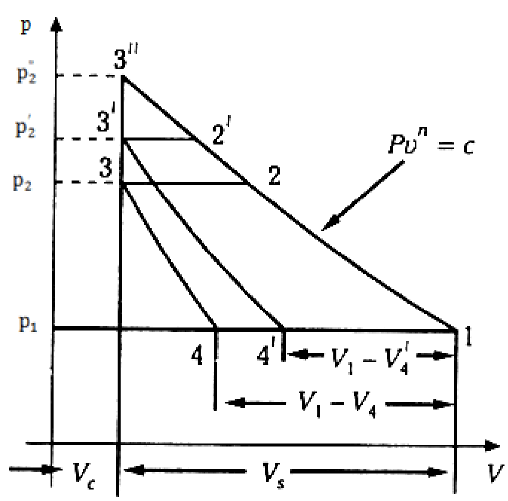


Figure 1.13 Multi stage compressions

It is seen that for a given intake pressure, the volume of air taken into the compressor cylinder decreases with increase in delivery pressure. At some delivery pressure the compression line intersects the line of clearance volume (point 3”) indicating that there is no delivery of air.

At this stage compression and re-expansion of same air takes place over and over again without any delivery of compressed air. As a result it is seen that the maximum pressure ratio attainable with a reciprocating compressor is limited by the clearance volume of the compressor. As clearance volume cannot be reduced beyond a certain limit the only alternative is to resort to multi-staging.

Work done in a two stage air compressor

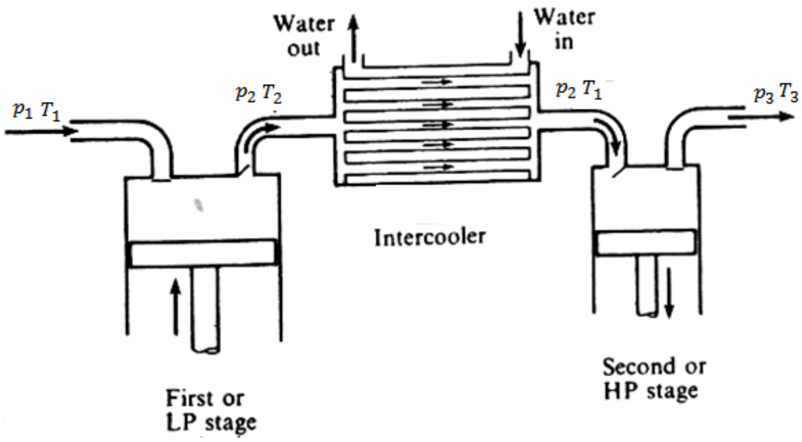


Figure 1.14 Multi stage compressions

Schematic diagram of Two stage air compressor with intercooler is shown in Figure 1.14 The air is first compressed in the LP cylinder to intermediate pressure p2. It is then passed through an intercooler where air is cooled at constant pressure before it is compressed in the HP cylinder. If the air is cooled back to initial temperature, then the inter cooling is said to be perfect.

The PV diagram for two stage compressor is shown in the Figure 1.15

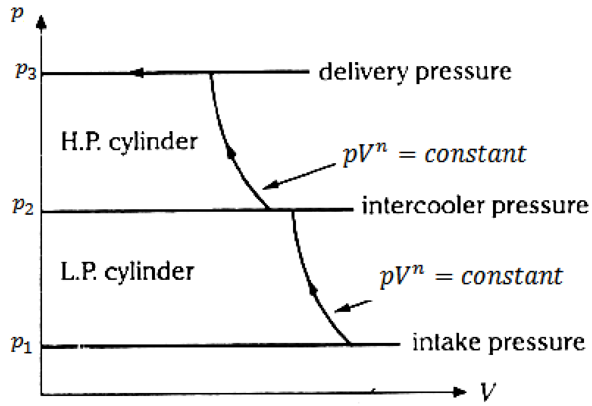


Figure 1.15 PV Diagram for two stage air compressor

Work done per cycle in LP cylinder

$$W_{LP} = \frac{n\,p_1V_1}{n-1}\left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right]$$

Work done per cycle in HP cylinder

$$W_{HP} = \frac{n\,p_2V_2}{n-1}\left[\left(\frac{p_3}{p_2}\right)^{\frac{n-1}{n}} - 1\right]$$

If the intercooling is perfect then $p_1 V_1 = p_2 V_2$

$$W = \frac{n\,p_1V_1}{n-1}\left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} + \left(\frac{p_3}{p_2}\right)^{\frac{n-1}{n}} - 2\right]$$

Condition for minimum power in two stage compressor with perfect intercooling is given by

$$p_2 = \sqrt{p_1 p_3}$$

For a N stage compressor with perfect intercooling , compressing air from p_1 to p_{N+1}

$$W = \frac{Nn\,p_1V_1}{n-1}\left[\left(\frac{p_{N+1}}{p_1}\right)^{\frac{n-1}{nN}} - 1\right]$$

Example 1.8: A two stage air compressor takes in air at a rate of 0.2 m^3/s Intake pressure is 1 bar and 16℃. Final pressure is 7 bar the intermediate pressure is ideal with perfect intercooling. (See Figure 1.16) The compression takes place according to law $\mathrm{p\,V^{1.25}} = \mathrm{Constant}$. The compressor runs at 600 RPM. Neglecting clearance determine a)The intermediate pressure b) Volume of each cylinder c) cylinder power

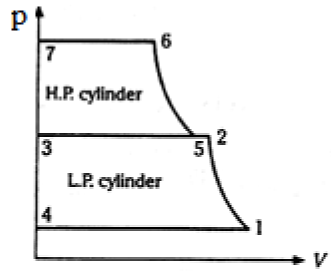


Figure 1.16

Given data

$n\ is\ the\ polytropic\ = 1.25\ for\ air$

$$m = 1\,\frac{kg}{min} = \frac{1}{60} = 0.0166\ kg/sec$$

$$V_1 = 0.2\,\frac{m^3}{sec}\quad T_1 = 16℃$$

$p_1 = 1\ bar\ , p_3 = 7\ bar$

Solution

Part a

For perfect intercooling

$$p_2 = \sqrt{\mathrm{p_1 p_3}} = \sqrt{1 \times 7} = 2.645\ \mathrm{bar}$$

Part b

$$V_1 = Volume\ of\ LP\ cylinder = 0.2\,\frac{m^3}{sec}\quad T_1 = 16℃$$

$$V_1 = \frac{0.2}{600/60}\ \frac{m^3}{cycle} = 0.02\ \frac{m^3}{cycle}$$

$$p_1\ V_1^{1.25} = p_2\ V_2^{1.25}$$

$$V_2 = V_1\left(\frac{p_2}{p_1}\right)^{\frac{1}{n}} = 0.02\left(\frac{1}{2.645}\right)^{\frac{1}{1.25}}$$

Volume of HP cylinder $= 0.0092\ \frac{m^3}{cycle}$

Part c

Minimum power required

$$W = \frac{Nn\, p_1 V_1}{n-1}\left[\left(\frac{p_{N+1}}{p_1}\right)^{\frac{n-1}{nN}} - 1\right]$$

$$W = \frac{2 \times 1.25 \times 100 \times 0.2}{1.25 - 1}\left[\left(\frac{7}{1}\right)^{\frac{1.25-1}{2\times1.25}} - 1\right]$$

$$W = 43\ kW$$

$$W = 42.58\ \mathrm{kW}$$

1.3 **COMPARISON OF DIFFERENT COMPRESSOR**

Flow rate, efficiency and the pressure rise within the compressor are the three most parameters used in defining the performance of a compressor and in its selection. Positive displacement compressors are generally suitable for small flow rates while centrifugal and axial compressors are more commonly applied for medium and large flow applications respectively. The advantages of centrifugal compressors are that they are reliable, compact and robust , have better resistance to foreign object damage and are less affected by performance degradation due to fouling. Positive displacement machines have wider operating domain when compared to other compressor types. Centrifugal compressors are most commonly applied in petrochemical or process industries in the flow rates ranging from 30 m^3/min to 3000 m^3/min. Typical comparison is given in the Table 1.2

Table 1.2 comparison of different compressors

Item	Reciprocating	Rotary vane	Rotary screw	Centrifugal
Efficiency at full load	High	Medium-high	High	High
Efficiency at part load	High due to staging	Poor: below 60% of full load	Poor: below 60% of full load	Poor: below 60% of full load
Efficiency at no load(power as % of full load)	High (10%-25%)	Medium (30%-40%)	High Poor (255-60%)	High-medium (20%-30%)
Noise level	Noisy	Quiet	Quiet it enclosed	Quiet
Size	Large	Compact	Compact	Compact
Oil carry over	Moderate	Low-medium	Low	Low
Vibration	High	Less	Less	Less
Maintenance	Many wearing parts	Few wearing parts	Very few wearing parts	Sensitive to dust in air
Capacity	Low-high	Low-medium	Low-high	Medium-high
Pressure	Medium- very high	Low-medium	Medium-high	Medium-high

1.4 CONTROL OF COMPRESSOR

1.4.1. Compressor drives

Alternating current (AC) electric motor are by far the most common drivers used for industrial plant air compression. Large electric motors are available in two basic types: Induction motor and synchronous motor. Nearly all industrial motors are three phase ac powered. Both the induction and synchronous motors rely upon the production of a revolving magnetic field (RMF) in the field winding.

Induction motors are used in 90 percent of industrial applications and are designed in two types: squirrel cage and wire wound rotor. The primary differences are the starting torque, current and amount of slip. Squirrel cage motors are commonly used.

Normally, gas turbine, diesel or Otto cycle engine power is not economic for stationary, continuous service compressor installation, except in special circumstances.

1.4.2 Control of compressors

There is a growing variety of control systems available for compressed air installations. These most often concern electric driver controls and compressor controls. Compressor controls are described below.

Plant air compressor systems normally are designed to operate at a fixed pressure and to deliver a variable volume. The Compressor is sized to deliver the maximum capacity and a control system is employed to reduce the compressor output to match the system demand.

Compressor may incorporate several different control systems to match the compressor volume and pressure to the demand. All of these controls monitor the system pressure as instantaneous indicator of the status of the match between the compressor output and system demand. Usually the control system will recognize and be designed to deliver air pressures between a design minimum and a design maximum damped system pressure. The damping is required to eliminate the effect of pressure pulses produced by most compressors. Pressure differentials of 0.1 to 0.5 bar between the minimum and maximum are specified in practice, the actual differential being a function of user requirements. This differential is known as the control range.

Energy consumption represents 80 percent of total cost for compressed air. therefore we must choose regulations systems carefully. There are two main groups of such regulation systems

1. **Continuous capacity regulation:** This method involves continuous control of drive motor or valve according to variation in pressure as shown in Figure 1.34. The result is normally a small pressure variations (0.1 to 0.5 bar), depending on the regulation system's amplification and its speed.

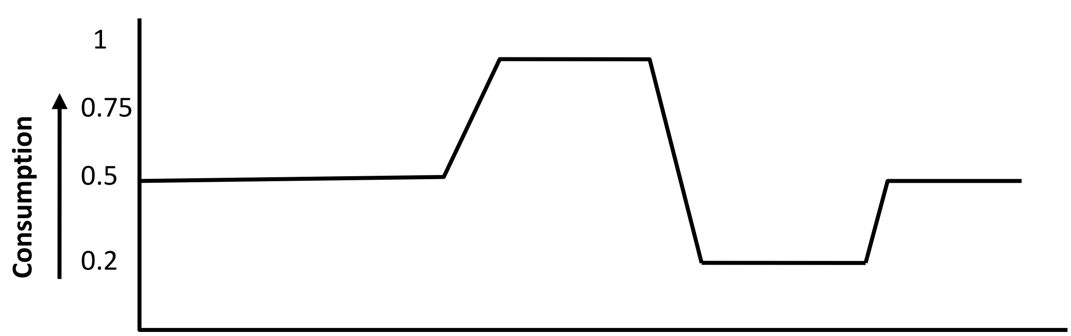


Figure 1.34: Continuous capacity regulation

1. **Load/unload regulation:** This method involves the acceptance of variation in pressure between two values as shown in Figure 1.35. This takes place by completely stopping the flow at the higher pressure (off loading) and resume the flow rate (loading) when the pressure has dropped to the lowest valve. Pressure variation depend on the permitted number of load/unload cycles per time unit, but normally lie within the range of 0.3 to 1 bar.

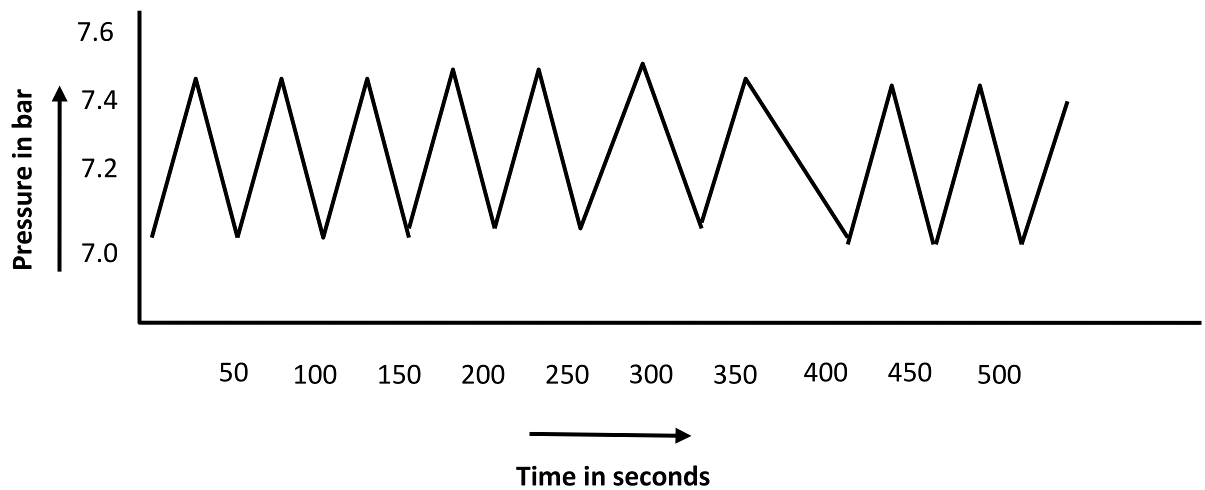


Figure 1.35: Load/unload regulation

1.4.2.1 Regulation principles for displacement compressors

a) Start/Stop and Load/unload controls

The simplest control mechanism turns the compressor on or off in response to system pressure. Schematic diagram of start /stop and unload/load control is shown in Figure 1.36 When the present system high pressure is reached, the compressed is turned off. When the system pressure falls to the preset minimum, the compressor is turned on. Compressors less than 5-10 kW are often controlled by completely stopping the electric motor when the pressure reaches an upper limit valve and restarting it when the pressure passes the lower limit value. The method demands a large system volume or large pressure difference between the start and stop pressure, to minimise the load on the electric motor. This is an effective regulation method under the condition that the number of starts per time unit is kept low.

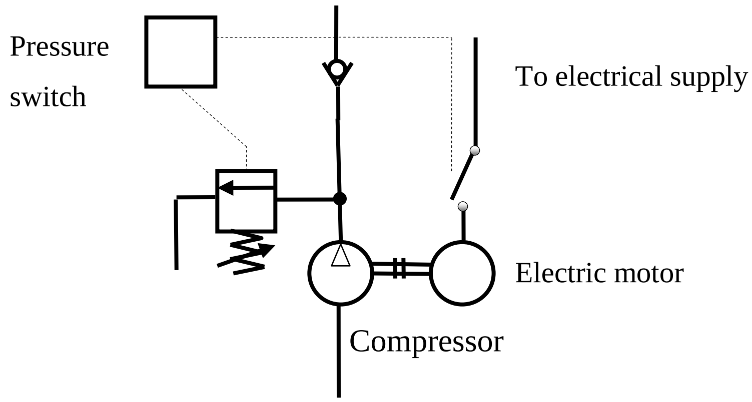


Figure 1.36: Load/unload regulation

b) Pressure Relief valve

The method used Pressure relief valve (PRV). This valve releases excess pressure into the atmosphere when the preset pressure is reached. The preset pressure can be set by adjusting the spring tension of the spring. Now a day, servo valve is used. The pressure can be easily controlled and the valve can also act as off-loading when starting a compressor under pressure. Schematic diagram of control of compressor using PRV is shown in Figure 1.37

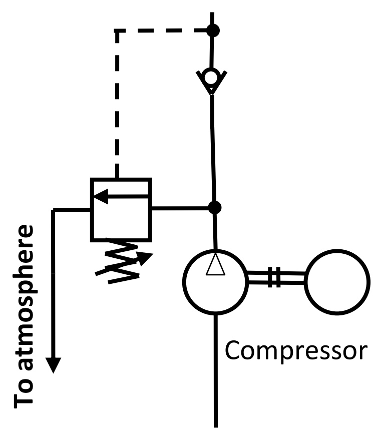


Figure 1.37: Regulation using pressure relief valve

c)**By pass regulation**

In this method, pressure relieved air is cooled and returned to the compressor intake. This method is often used on process compressors where gas is unsuitable or too valuable to release into the atmosphere. Schematic diagram of control of compressor using bypass regulation is shown in Figure 1.38

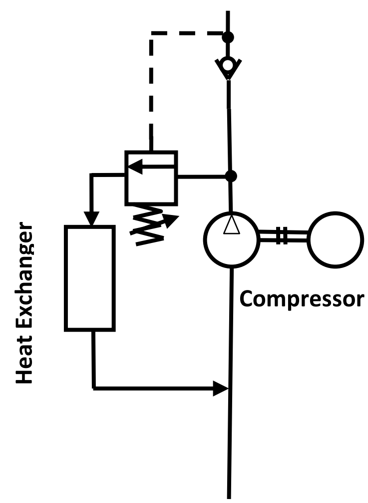


Figure 1.38: Bypass regulation

d) **Throttling the intake**

Throttling is an easy method to reduce the flow. Schematic diagram of control of compressor using throttle intake regulation is shown in Figure 1.39. By increasing the pressure ratio across the compressor, depending on the induced under pressure in the intake, the method is however limited to a small regulation range. Liquid injected compressors, which have a large permitted pressure ratio, can however be regulated down to 10 % of the maximum capacity. This method makes relatively high energy demands, due to the high pressure ratio.

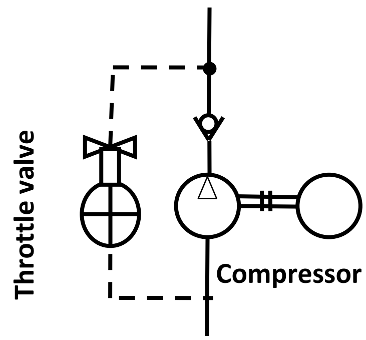


Figure 1.39: Throttle intake control regulation

e) **Pressure relief with throttled intake.**

Schematic diagram of control of compressor using throttle intake and pressure regulation is shown in Figure 1.40.The most common regulation method currently used that unites a maximum regulation range(0-100%) with low energy consumption, only 15-20% of full load power with an off-loaded compressor (zero flow). The intake valve is closed, but with a small opening remaining, at the same time as a blow off valve opens and relieves the outgoing air from the compressor

The compressor element then works with a vacuum in the intake and low counter pressure. It is important that the pressure relief is carried out quickly and that is relieved volume is small to avoid unnecessary losses during the transition from loaded to unloaded. The system demands a system volume (air receiver) , the size of which is determined by the acceptable difference between loading and off loading pressure and by the permitted number of unloading cycles per hour.

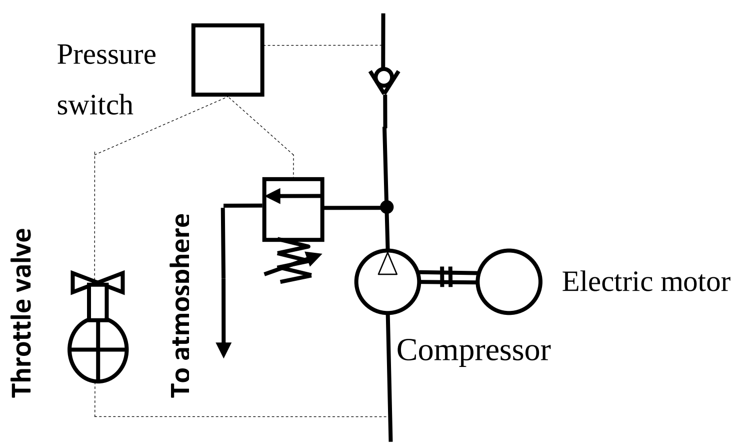


Figure 1.40: Throttle intake and pressure relief control regulation

f) **Speed regulation**

A combustion engine, turbine or frequency controlled electric motor controls the compressor's speed and thereby the flow. It is an efficient method to attain an equal outgoing pressure and low energy consumption.

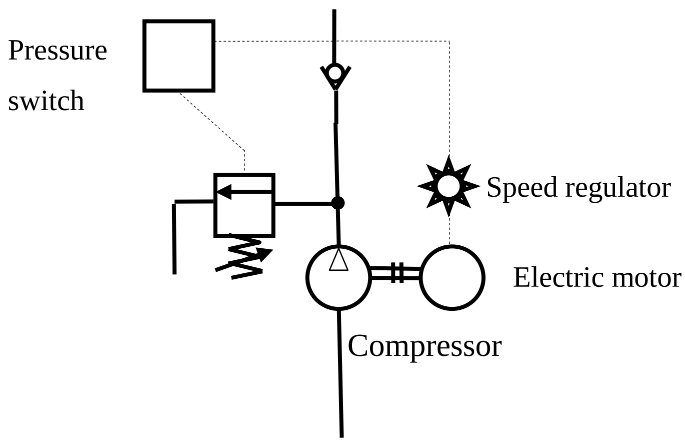


Figure 1.41: Speed regulation

Schematic diagram of speed control of compressor is shown in Figure 1.41.The regulation range varies with the type of compressor, but is greatest for liquid injected compressor. Frequently speed regulations and pressure relief are combined, with or without a throttled intake at low degrees of loading.

g) **Variable discharge port**

The capacity of screw compressors can be regulated by moving the position of the discharge port in the housing, in the screw's lengthways direction, towards the intake. However, the method demands high power consumption with sub-loads and is relative unusual.

g) **Suction valve unloading**

Piston compressors can be effectively relieved by mechanically forcing the intake valves to the open position. Air is then pumped out and in under the position of the piston, with minimal energy losses as result often lower than 10% of the loaded shaft power. On double acting compressors there is generally multi-stage off loading, where one cylinder at a time is balance to better adapt the capacity to demand. An odd method used on process compressor is to allow the valve to be open during a part of the piston stroke and thereby receive a continuous flow control.

Objective Type Questions

1. -----------------types of reciprocating compressors are most commonly used compressors**.**

1. Compare to positive displacement type compressor, dynamic compressor are much -------in size and produce much ----- vibration.

3. If the air compressor has a duty of -------, and the air compressor will be running for 10 minutes, then it should run for a combined maximum of 6 minutes ON and 4 minutes OFF.

4. It is not possible in practice as to achieve isothermal process, as the compressor must run very --------. In practice compressors run at high speeds which results in -------- process

5. In ------------------ control regulation, pressure relieved air is cooled and returned to the compressor intake

6.In a single acting reciprocating compressor, the suction, compression and delivery of air takes place in ___________ of the piston

7. Intercooling in mutli stage compressor is done to -----------the work of compression

8. The ratio of work done per cycle to the stroke volume of the compressor is known as ---------

9. The volume of air delivered by the compressor is called ………………..

10. In a centrifugal compressor, an increase in speed at a given pressure ratio causes increase in flow and --------- in efficiency

11.A large clearance Volume in a reciprocating compressor results in ----------------volume flow rate

12.When the temperature of air leaving the intercooler, in a two stage compression with intercooler, is ___________ the original atmospheric air temperature, then the intercooling is known as perfect or complete intercooling

State True or False

1. A dynamic compressors works with a constant pressure.

2. Positive displacement compressor is selected for larger volume of gas and higher pressure ratios. Dynamic compressor is selected for lower volume of gas fluid and smaller pressure ratios.

3. Compressor will have higher efficiency if compression follows isothermal process.

4. Running on the same operating speed, the two lobes blower can deliver large air flow and volume than conventional three lobes type

5. Continuous capacity regulation is the most common type of control and regulations used in Compressor.

6. The total heat rejected in a reciprocating air compressor is equal to the sum of the heat rejected during polytropic compression per kg of air and heat rejected in the intercooler per kg of air

7. In a four stage compressor, if the pressure at the first and third stage are 1 bar and 16 bar, then the delivery pressure at the fourth stage will be 64 bar

8. The volumetric efficiency for reciprocating air compressors is about 60 to 70 %

9. The minimum work required for a two stage reciprocating air compressor is double the work required for each stage

10. The clearance volume in the compressor is kept minimum because it effects on volumetric efficiency.

11. Work done by a two-stage reciprocating air compressor per cycle is equal to the workdone in LP. cylinder and H.P. cylinder

12. The actual volume of air delivered by a compressor, When reduced to the normal temperature and pressure conditions is called compressor capacity

13. An axial compressor gives optimum performance at high speeds and large volume flows

Review Questions

1. Explain the different stages of preparation of compressed air.

2. Explain the effect of type of compression in reciprocating air compressor

3. Define isothermal efficiency and derive an expression for the same

4 Explain the effect of clearance volume in single cylinder single stage compressor

5. Define volumetric efficiency of an air compressor and derive an expression for the same

6. Explain the effect of increasing delivery pressure on the volume of air delivered

7. Derive an expression for the intermediate pressure which gives minimum power in a two stage compressor with perfect inter cooling

8. What is the advantage of multi stage compressor

9. Differentiate between positive and dynamic displacement compressor

10. Name three types of positive displacement compressor

11. Explain the working principle of screw compressor. What are its advantages and disadvantages?

12. What is a diaphragm compressor? List its types and advantages

13. Explain on/off regulation of compressor with a neat sketch

14. Explain pressure relief and by pass regulation of compressor control with neat sketches.

15. List all factors to be considered in selection of a compressor for a give application.

Answers

Fill in the Blanks

1. Piston
2. smaller/less
3. 60/40
4. Slowly/polytropic
5. Bypass
6. Two strokes
7. minimise
8. Mean effective pressure.
9. Compressor capacity
10. decrease
11. reduced
12. equal to

State True or False

1. True
2. False
3. True
4. False
5. False
6. True
7. True
8. False.
9. True
10. True
11. True
12 False
13. True

Lecture 35

COOLING AND DRYING OF COMPRESSED AIR.

Learning Objectives

Upon completion of this chapter, Student should be able to

- Explain the importance of secondary treatment of compressed air
- List the various functions of an after cooler
- Describe the functions of air and water coolers
- Explain the theory of air dryers
- Describe various types of air dryers used in pneumatic Industry
- List the various functions of Air receiver
- Differentiate Air receiver and Air accumulators
- Size the Air receiver and compressor

1.1 IMPORTANCE OF SECONDARY TREATMENT

Compressed air is an essential power source that is widely used throughout industry. This safe, powerful and reliable utility is the most important part of production process. However, compressed air will contain water, dirt, wear particles and even degraded lubricating oil which all mix together to form an unwanted condensate. This condensate often acidic, rapidly wears tools and pneumatic machinery, blocks valves and orifices causing high maintenance and costly air leaks. It also corrodes piping systems and can bring production process to an extremely expensive standstill.

The quality of air required throughout a typical compressed air system can vary. It is highly recommended that the compressed air is treated prior to entry into the distribution system as well as at each usage point or application. This approach to system design provides the most cost effective solution to system purification as it not only removes the contamination already in the distribution system, it ensures that only the most critical areas receive air treated to the highest level. In many instances the compressed air system will be supplying air to more than one application and although the purification equipment specified in the compressor room would remain unchanged, the point of use protection will vary depending upon the air quality requirements of each application. In many cases this action alone is not enough, as modern production systems and processes demand an even higher level of air quality.

Where required, “point of use” filtration, refrigeration or desiccant air dryers can provide the correct air quality, without the need for drying the complete compressed air installation, which can be both costly and totally unnecessary.

The three desirable qualities of compressed air are:

a) **Compressed air must be cleaned**: Dirty or contaminated air may cause seals to wear. Dirt can wedge into clearance between moving parts or block air passages which may cause faulty operation or component failure
b) **Compressed air must be cooled:** Heat is undesirable due to the fact that if this hot air is passed directly into the pipes, they would elongate die to the heat. Contraction would occur when compressor is shut down. This recurring process would cause the joint to leak and reduce efficiently. Therefore, flexibility must be built in. Instead of discharging air from the compressor outlet directly into the air receiver for storage, the air is passed through the after cooler
c) **Compressed air must be dried**: water in compressed air lines can wash away lubricant from pneumatic tool parts causing rapid wear increased air consumption, and increased maintenance costs. Moisture can freeze at the exhaust of tools and interfere with their efficient operation. In applications like paint spraying, air agitation of liquid etc. moisture can cause rejections and scrap work. There are several ways to remove moisture from compressed air before it can damage. Generally, the least expensive is after cooling.

1.2 AFTER COOLER

Air compression process may be designed to either to be adiabatic or to involve heat transfer, depending on the purpose for which the gas is compressed. If the compressed air is to be used promptly in engine, adiabatic compression may be required. In much application, however, air is stored in a tank for use as needed. The air in the tank loses heat to the surroundings and reaches room temperature when finally used. In this case the overall effect of compression and storage is to increase pressure of gas without change of temperature. In can be shown that if the gas is cooled during compression, instead of after the process, the work required will be less than for adiabatic compression. A further advantage of cooling is the reduction of volume and the consequent reduction of pipe line losses. For this reason, since cooling during compression is not very effective, after coolers are often used to cool the gas leaving the compressor.

The compressed air discharged from an air compressor is hot. Compressed air at these temperatures contains large quantities of water in vapour form. After coolers are heat exchangers for cooling the discharge from a air compressor. They use either air or water and are an effective means of removing moisture from compressed air.

After coolers control the amount of water vapour in a compressed air system by condensing the water vapour into liquid form. In a distribution or process manufacturing system, liquid water can cause significant damage to the equipment that uses compressed air. An after cooler is necessary to ensure the proper functionality of pneumatic or air handling devices that are a part of process manufacturing systems

About 75 % of the moisture can be removed using after cooler. A moisture separator installed at the discharge of the after cooler removes most of the liquid moisture and solids from the compressed air. Utilizing centrifugal force, moisture and solids collect at the bottom of the moisture separator. An automatic drain should be used to remove the moisture and solids

1.2.1 Functions of compressed air after coolers

- Cool air discharged from air compressors via the heat exchanger
- Reduce risk of fire (Hot compressed air pipes can be a source of ignition)
- Reduce compressed air moisture level
- Increase system capacity
- Protect downstream equipment from excessive heat

Coolers are usually sized with a CTD (Cold Temperature Difference) of 2.7°C, 5.5°C, 8.3°C, or 11°C. This means that the compressed air temperature at the outlet of the after cooler will be equal to the cooling medium temperature plus the CTD when sized at the specified inlet air temperature and flow.

1.2.2 Types of after coolers.

There are two basic types of air after coolers:

1. Air-cooled
2. Water-cooled.

Compressor manufacturers may include after coolers within the compressor package. In general these compressors are referred to as integral after coolers. A stand-alone or freestanding after cooler is a separate unit installed downstream of the compressor.

1.2.2.1 Air-Cooled After cooler

Air-Cooled After coolers provide economical cooling by using ambient air to cool the hot compressed air from an air compressor. They cool the hot compressed air leaving the compressor at a temperature of approximately 100°C to 150°c to the desired inlet temperature of an air dryer which is approximately 35°c to 50°c. As the compressed air cools, about 75% of the water vapour present condenses into liquid water which should be immediately removed from the system with a separator. Air-Cooled After coolers can be sized to cool the hot compressed air to within -15°c to -5°F of the ambient air temperature. They are available in capacities from 500 LPM to100000LPM

1.2.2.2 Water-Cooled Pipe Line after cooler

The most common style for compressed air service is a Shell and Tube Heat Exchanger. The pipe line after cooler consists of a shell with a bundle of tubes fitted inside. Typically the compressed air flows through the tubes in one direction as water flows on the shell side in the opposite direction. Heat from the compressed air is transferred to the water. Water vapor forms as the compressed air cools. The moisture is removed by the moisture separator and drain valve. The tube bundles can be fixed or removable. Fixed tube bundles cost less but are more difficult to maintain than bundles that can be removed for cleaning or service.

The disadvantages of a water-cooled after cooler include high water usage and difficult heat recovery. Advantages to using a water-cooled after cooler include better heat transfer and no required electricity

1.3 DRYING OF COMPRESSED AIR

Function: is to lower the dew point of the compressed air by removing the moisture from it. For simple applications, to remove excess humidity, we need simple after cooler, an air receiver, and a filter with condensate traps. However, to get high quality compressed air additional means of dehydration must be provided using dryer.

1.3.1 TYPES OF AIR DRYERS

Generally four basic types of air dryers are used in Industries.

1. Absorption type dryer
2. Adsorption type dryer
3. Refrigeration dryer
4. Membrane dryer

1.3.1.1 Absorption type dryer

Absorption drying is a purely chemical process. The moisture in the compressed air forms a compound with drying agent like phosphoric pentaoxide in the tank. This causes the drying agent to break down. It is then discharged in the form of a fluid at the base of the tank. Schematic diagram of absorption dryer is shown in Figure 1.1

Oil vapour and oil particles are also separated in the absorption dryer. Large quantities of oil have an effect on the efficiency of the dryer. Therefore it is advisable to include a fine filter in front of the dryer.

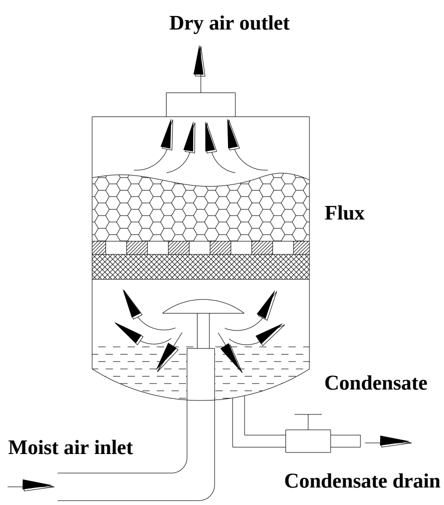


Figure 1.1 Absorption dryer

Advantages of Absorption dryer

1. Simple to install
2. Low mechanical wear because there is no moving parts
3. No external energy requirement

Disadvantages of Absorption dryer

1 Maintenance cost is high

2. Low efficiency

3. Consumable cost is high

1.3.1.2 Adsorption type dryer

Adsorption is a physical process of moisture removal on the porous surface of certain granular materials. Gaseous molecules are attracted to certain solid surfaces by van der walls forces and this causes the adsorption. The degree of attraction or adsorption depends on properties of gaseous molecules and desiccant. Most commonly used desiccants re activated alumina, molecular sieves and silica gel.

Figure 1.2 shows the various parts of adsorption dryer. Wet incoming compressed air after passing through a pre-filter is directed to the adsorption chamber containing the desiccant. Water vapour in the compressed air is absorbed by the desiccant. Thereafter dry air is allowed to pass to the application through the after filter.

Adsorption dryers usually have two desiccant filled chambers with interconnecting piping and switching valves. The valves permit removal of the collected moisture from one chamber while the other chamber is used to purify the compressed air. The twin tower design facilitates simultaneous compressed air drying and saturated desiccant regeneration for non-stop production. A contaminated desiccant bed can be regenerated by either elevating its temperature or by decreasing its pressure and purging.

The capacity of the desiccant bed is limited owing to abrasion and contamination of the adsorption medium by oil and other substances. Under normal conditions, it is required to replace the drying agent once in 2 to 3 years.

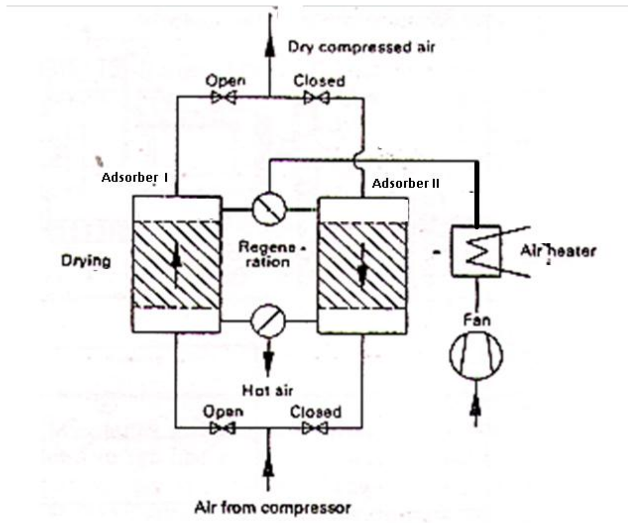


Figure 1.2 Adsorption type dryer

1.3.1.3 Refrigerated dryer

The layout of a typical refrigerated air dryer is shown in **Figure 1.3** It is composed of a heat exchanger (stage1) and a refrigerating unit (Stage2) to reduce the temperature of the compressed air. The incoming warm and humid air is first passed through the air –to-air heat exchanger, and then through the refrigerating unit to reduce the temperature of the compressed to as low as +2℃ . This drying method is based on the principle that if the compressed air is cooled to a temperature below the dew point, condensation talks place and water is precipitated. Almost all the water and oil particles get condensed, and collected in the water traps provided at appropriate points. The cooled compressed air is then filtered to remove from it the suspended solid particles and most of the oil mist. The pressure dew point of 2℃ is possible with this type, which is sufficient enough for the smooth operation of the most of the industrial and process applications.

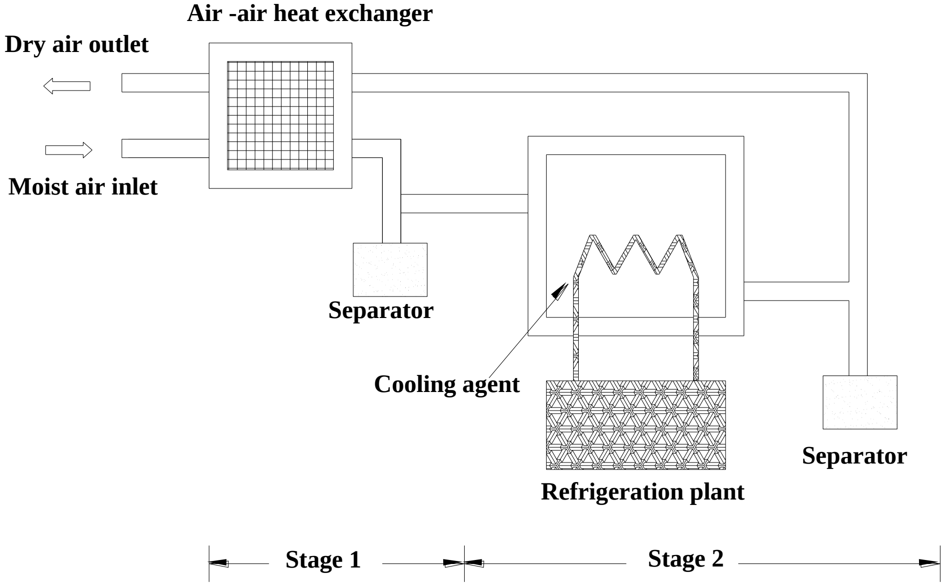


Figure 1.3 **Refrigerated dryer**

In most cases, the type of dryer needed is determined by the pressure dew point required in the systems. **Table 1.1** gives the summary of advantages and disadvantages of the dryers.

Table 1.1 Comparison of three types of dryer

Type	Advantages	Disadvantages
Absorption	Pressure dew point + 16 °C	Inlet temperature must not exceed 30 °C
	Low capital cost	Drying agents are consumables and therefore must be regularly replenished. Highly corrosive chemical are used. They are not environmental friendly
Refrigeration	+ 3 °C pressure dew point	Output dew point will vary with approach temperature at the inlet and cleanliness of heat exchanger
	Input temperature can be as high as 16 °C	
Adsorption	Achievable pressure dew point of -40 °C	High capital cost

		High operating cost
		Use of micro filters adds cost to prevent the residue from chemicals.

1.3.1.4 Membrane dryer

Membrane dryers are yet another type of dryer to remove moisture from compressed air. It consists of three stages.

Stage 1: Contains a filter which removes the water and contaminants down to 5 micron.

Stage 2: High efficiency coalescing filter removes oil and sub micron particles down to 0.01 micron

Stage 3: membrane module removes the remaining moisture in the vapour form

In this of dryer, pre-cleaned compressed air is passed through a bundle of hollow fibres in the membrane module. The hollow fibres constitute a membrane layer specially designed to attract the water vapour inside. This water vapour diffuses through the very thin selective layer until it reaches the outside of the membrane due to partial pressure difference between inside and outside of the membrane. The permeated water vapour is then swept away by a small amount of dry air fed back along the length of the membrane fibre through a purging valve.

Advantages of Membrane dryer

1. Membrane dryers typically maintain a pressure dew point of 0 ℃
2. Membrane dryers are simple and compact
3. Dryers run almost noiseless
4. There is no need for regeneration because membranes never gets saturated
5. They do not require electric supply
6. Low operating cost

3.3.2 Theory of drying

Air taken from the atmosphere contains moisture which in normal circumstances is precipitated as condensate within the system through which it passes, namely in the compressor after cooler, the air receiver, collecting points in the piping and the filter of the service unit ahead of the air consumers.

Compressed air will always contain as much water as it is capable of absorbing at the lowest temperature assumed by the air on its passage from the compressor to the consumption point. Typical values of water vapour saturation capacity of an air at selected atmospheric temperatures is given in Table 1.2

Table 1.2 Water vapour saturation capacity of air at selected atmospheric temperatures

Temperature ℃	-10	0	5	10	15	20	30	50	70	90
Water vapour, g/m^3	2.1	4.9	7	9.5	13	17	30	83	198	424

For example, at 20℃ one cubic meter of compressed air will still contain 17 grams of moisture. Existing in the form of water vapour, this moisture is not separated in the filter of the service unit either; rather it is entrained by the air into all the downstream control and operating components. Due to the centrifugal action taking place in the filter and increased flow velocity resulting there from, the air stream is cooled a certain amount relative to the ambient temperature, which does cause a small portion of condensate to be separated in the filter.

Since the control and operating elements of the pneumatic systems will normally be at ambient temperature, no further condensate will be precipitated at these points. Therefore, the remaining moisture passes out with the external air released to the atmosphere. As observed above, the low residual moisture content of the air will hardly constitute a potential hazard to standard pneumatic control and operating components.

In the case of specialised applications and used of compressed air, however, as in spray painting, intricate low-pressure control systems, the chemical and pharmaceutical industries, food industry, pneumatic instruments and pneumatic conveying, the situation is different. Everywhere that the compressed air comes into direct contact with the process medium in such instances, simple conditioning of the air by the means described usually will not be sufficient. It is then necessary to provide additional means of dehydrating and filtering the compressed air.

1.3.2.1 Humidity and relative humidity

Water vapour is constantly evaporating from lakes, rivers and seas and is absorbed by the atmosphere and carried across vase distances by winds, finally being deposited in the form of rain, mist, etc. Atmospheric air therefore, is nature's way of transmitting large quantity of water vapour all over the earth.

Relative humidity(RH): it is ratio of amount of water present in a given quantity of air, to the maximum possible amount which it can contain under the same conditions of pressure and temperature and ratio is usually expressed as percentage. The amount of moisture condensing out of compressed air is a function of the relative humidity of the intake air and temperature.

$$\text{Relative humidity of air} = \frac{\text{Absoulte humidity}}{\text{humidity at saturatrion}} \times 100$$

In other words, relative humidity is the amount of water vapour present in a given volume of air, whereas the humidity at saturation is the total amount of water vapour which that the same volume of air can absorb at the given temperature. **Table 1.3** gives the mass of water in kg per 100 m^3 of free saturated air. **Table 1.4** gives the amount of condensate in g/m^3 of air at various temperature and Relative humidity

Table 1.3 Mass of water in kg per 100 m^3 of free saturated air.

Temperature °C	Gauge pressure (bar)				
	0	2	4	6	8
0	0.48	0.17	0.10	0.07	.05
20	1.73	0.576	0.346	0.247	0.192
40	5.10	1.7	1.02	.728	.567
60	12.95	4.32	2.59	1.85	1.44

80	29.04	9.68	5.81	4.15	3.23
100	56.00	19.33	11.76	8.40	6.53
120	0.00	36.73	22.04	15.74	12.24
140	0.00	0	37.74	29.96	20.97

Mass of water in in kg per 100 m^3 of free saturated air

Table **1.4** Amount of condensate in g/m^3 of air at various temperature and RH

Temp ℃	Percentage relative humidity (RH)									
	10	20	30	40	50	60	70	80	90	100
-12	0.179	0.354	0.533	0.709	0.888	1.066	1.2503	1.421	1.600	1.780
0	0.483	0.965	1.451	1.933	2.426	2.906	3.387	3.867	4.348	4.830
10	0.934	1.865	2.790	3.730	4.670	5.606	6.520	7.460	8.400	9.337
15	1.313	2.617	3.816	5.063	6.386	7.795	9.029	10.158	11.631	17.957
21	1.826	3.661	5.469	7.300	9.131	11.946	12.793	14.600	16.431	18.698
27	2.494	4.980	7.516	10.069	12.521	15.012	17.582	20.024	22.757	25.634
32	3.307	6.614	10.061	13.548	16.835	20.276	23.486	27.0044	30.451	33.721
35	3.936	7.872	11.808	15.744	19.681	23.617	27.554	31.489	35.426	39.248
37.8	4.531	9.039	13.571	18.102	22.611	27.141	31.673	36.489	39.900	45.248
43.3	6.018	12.037	18.056	24.075	30.094	36.112	42.131	48.150	54.160	60.371
49	7.895	15.790	23.686	31.580	39.476	47.370	55.267	63.166	78.910	78.928
55	10.161	20.322	30.583	40.650	50.805	61.200	71.127	81.3	91.504	101.656

The condensate, as the precipitated water in the compressed air line is termed, causes the damage if it is not removed properly. Corrosion in pipes and tubes, corrosion in control and working elements and corrosion in machine parts. If the condensate gets into the pneumatic equipment, proper functioning may be prevented. Solid particles such as dust, rust, and scale can also have an adverse effect on the function of the various items of pneumatic equipment.

Oil residues from the compressor can produce together with the compressed air a mixture of oil mist and air (gas mixture) which can cause explosions at higher temperatures (above 353k)

Problem 1.1 : A compressor delivers 400 m^3 of free air per hour at a pressure of 6 bar gauge and a temperature of 40℃ . Atmospheric air at compressor intake has a relative humidity of 80 % and a temperature of 20℃. Determine the amount of water that can be extracted from the compressor plant per hour.

Solution: Refer to Table 1.3

At 20 ℃ and zero bar gauge pressure, 100 m^3 of free saturated air contains 1.73 kg of water. From the definition of RH

$$\text{Relative humidity} = \frac{\text{Amount of water actually present in air}}{\text{Amount of water present in saturated air}} \times 100$$

$$80 = \frac{\text{Amount of water actually present in air}}{1.73} \times 100$$

Amount of water actually present in air = 1.384 kg

Since 400 m^3 is delivered, water content of air entering the compressor = 1.384 × 4 =5.536 kg

From the Table 1.3, corresponding to 40 ℃, and 6 bar compressor output pressure, amount of water per 100 m^3 of free saturated air is given by 0.728

Since 400 m^3 is delivered, water content of air leaving the compressor = 0.728 × 4

=2.912 kg

Therefore the amount of water extracted from the compressor plant per hour is

5.536-2.912 = 2.62 kg

1.3.2.2 Dew point temperature

The temperature at which air is fully saturated with moisture (that is 100 % humidity) is the dew point. Simply put, dew point is the temperature where condensation begins. Cooling below dew point will cause condensation of the water vapour. Lower the dew point, the less moisture the air is able to absorb or hold. For example 1 m3 of air has 17 grams of water at 20 ℃ and at -10℃

water vapour it is 2.1 grams. The capacity of holding water in air is a function of volume and temperature it does not depend on pressure. But still it is necessary to consider the working pressure of the systems when comparing different facility for the dehydration of air. This brings in the term pressure dew point.

a) Pressure dew point: Temperature representing the dew point at the respective operating pressure of the dryer is known as pressure dew point. The air is a compressible gas and the dew point temperature changes with the pressure. More precisely it is not the pressure but volume that matters. When the gas is pressurized, the volume is reduced and the air has less capacity to hold moisture. We can say with the increasing pressure and reduced volume, the dew point temperature also increases. In drying air by refrigeration, pressure dew point defines the lowest air temperature attainable in the dryer at the operating pressure of the system.

b) Atmospheric dew point:

As discussed above, Compression and expansion of air affects its dew point. Generally speaking, compression increases dew point, and expansion (i.e. de-compression) lowers dew point. For example, consider compressed air leaving a dryer at 15 bar with a pressure dew point of -40 ºC @ 15bar. If the pressure is eventually reduced to 7.5bar, the pressure dew point will fall to -45 ºC @ 7.5 psig. If the air is further expanded to 5 bar, the pressure dew point becomes -60º C @ 0.4 bar. For this reason, the phrase pressure dew point (PDP) is commonly used. This term usually refers to the dew point of the compressed air at full line pressure. Conversely the phrase atmospheric dew point refers to what the dew point would be if fully depressurized to atmospheric conditions. Figure 1.4 shows the relationship between atmospheric dew point versus pressure dew point for various pressures ranging from 0 MPa to 1.5 MPa

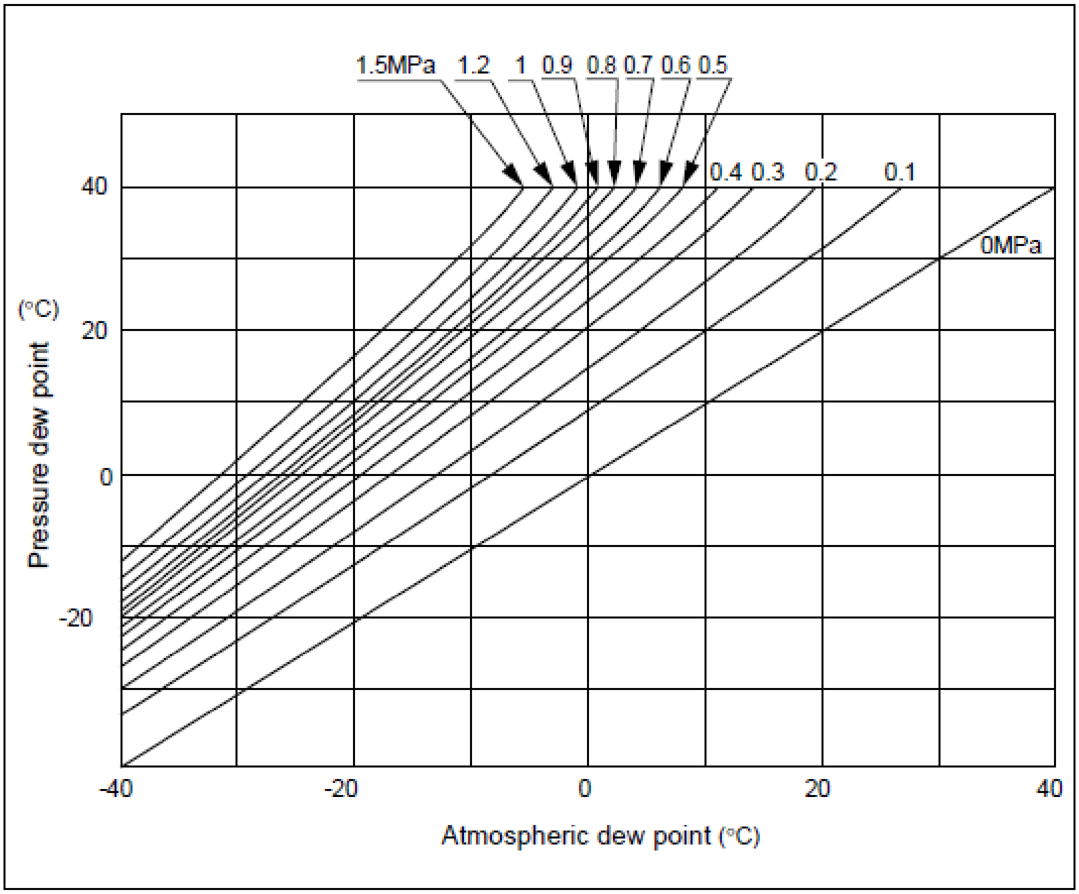


Figure 1.4 Atmospheric dew point versus Pressure dew point

Problem 1.2 : Find the atmospheric dew point where pressure is 0.7 MPa and pressure due point is 5 °C

Solution

Refer to Figure 1.4, Corresponding to 0.7 MPa and pressure dew point of 5 °C we get atmospheric dew point as -20 °C

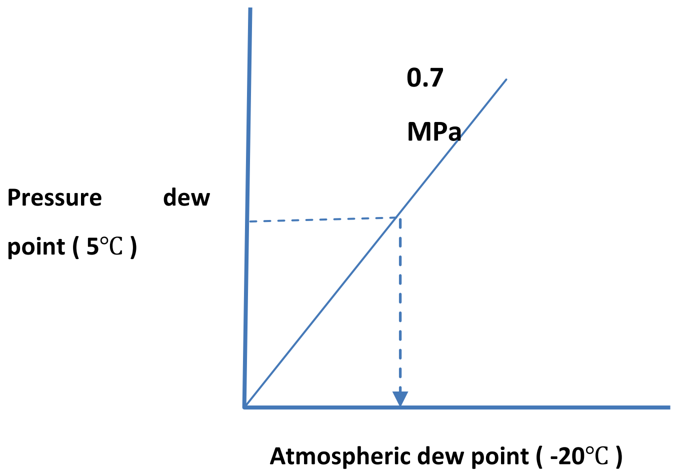


Figure 1.5

Problem 1.3 : Find the amount of condensate per shift in a plant, if air is compressed to 6 bar (gauge) receiver pressure. Assume the initial condition of air at sea level at 21 ℃ and 60 % Relative humidity (RH). Assume 35 ℃ temperature of compressed air in the receiver. Flow rate of the compressor is given as 20 m^3/min

Solution:

Refer to Table 1.4

Given that the initial condition of air at sea level at 21 ℃ and 60 % Relative humidity (RH).

1 m^3 of air has 11.946 grams of the water vapour

Compression ratio reveals how many m^3 of ambient air is to be used to produce 1 m^3 of compressed air at that pressure.

$$\text{compression ratio} = \frac{6+1}{1} = 7$$

It means 7 m^3 of free air is required. From the Table 1.4 each m^3 of air contains 60 % RH and 21℃ , 11.946 grams of moisture.

Hence the amount of water in 6.9 m^3 is

$7 \times 11.946 = 83.622$ grams of water

Let us assume 35 °C temperature of compressed air in the receiver.

From the table, 1 m^3 of air at 35 ℃ and 100 % RH amount of water is 39.248 grams.

Therefore, amount of water condensed and separated

= 83.622-39.248 =43.374 grams

Therefore, there will be $\frac{43.364}{7} = 6.2$ grams of water for each m^3 of ambient air

Flow rate of the compressor is given as 20 m^3/min

Amount of water for 20 m^3/min = $6.2 \times 20 = 124$ grams in one minute

Amount of water for one hour = $124 \times 60 = 7440$ grams

Amount of water in 8 hours shift = $7440 \times 8 = 59520$ grams ≅ 60 kg.

Problem 1.4 : Find the amount of condensate in one hour if 22 kW compressor operates under the following condition a) Air at 60% relative humidity and 30℃ ambient temperature is pressurised to 7 kg/cm^2(7 bar). It is then cooled to 25 ℃. Compressor output is 3 Nm3/min at 7 kg/cm^2(7 bar)

Solution

Refer the nomogram given in the **Figure 1.6**, locate point 1 which corresponds to inlet temperature of the compressor and erect a perpendicular line to meet 60%RH line. And then draw the horizontal line to cut 7 bar pressure line. We get pressure dew point temperature as 60℃.($1 \rightarrow 2 \rightarrow 3 \rightarrow 4 \rightarrow 5$)Since the air is cooled to 25 erect a vertical line to cut 7 bar pressure line. ($6 \rightarrow 7 \rightarrow 8 \rightarrow 3$) From the nomogram water liquid collected is 20.7-3.2 = 17.5 g/Nm^3

Which amounts to 17.5x3x60=3150 grams per hour. More than 3 litres of water is produced each hour.

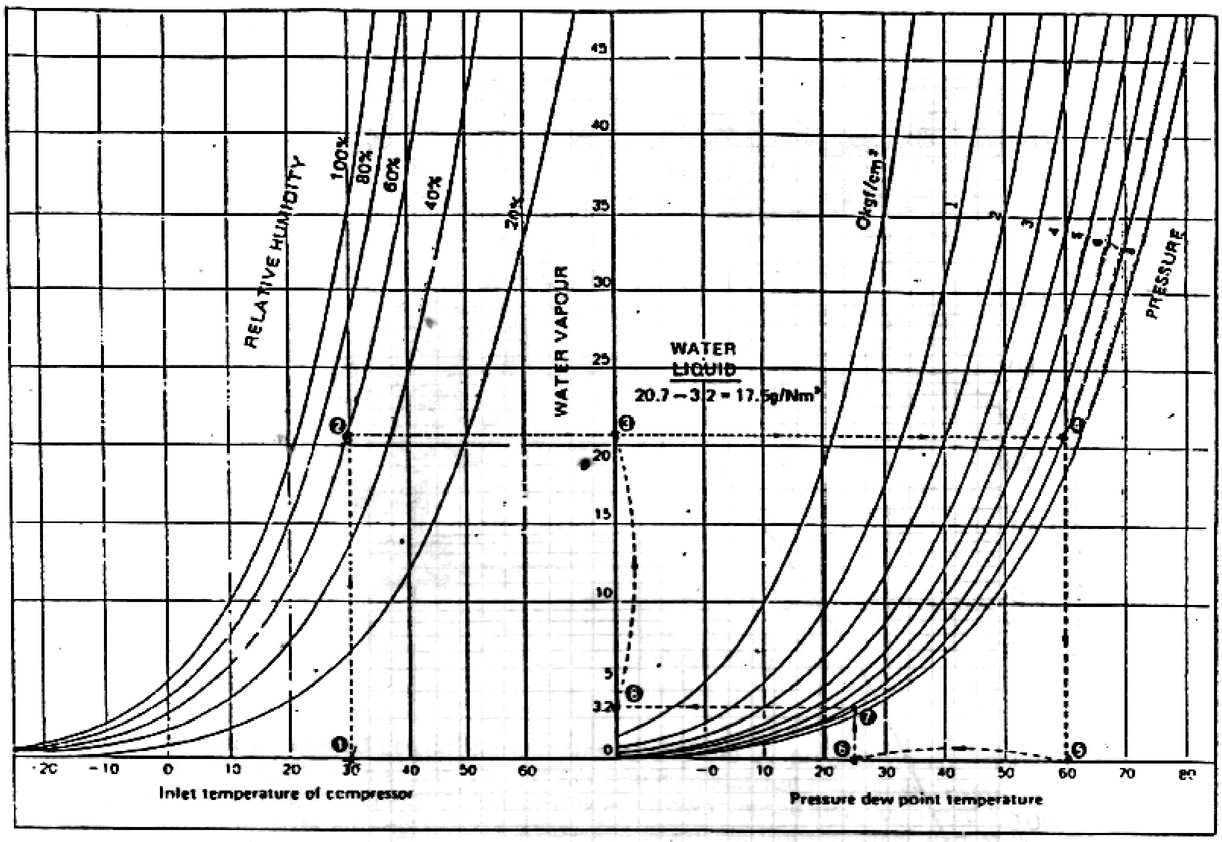


Figure 1.6: Pressure dew point nomogram.

Problem 1.5 : Using nomogram, Find the amount of water vapour saturation capacity of an air at 20°C and compare with Table 1.2

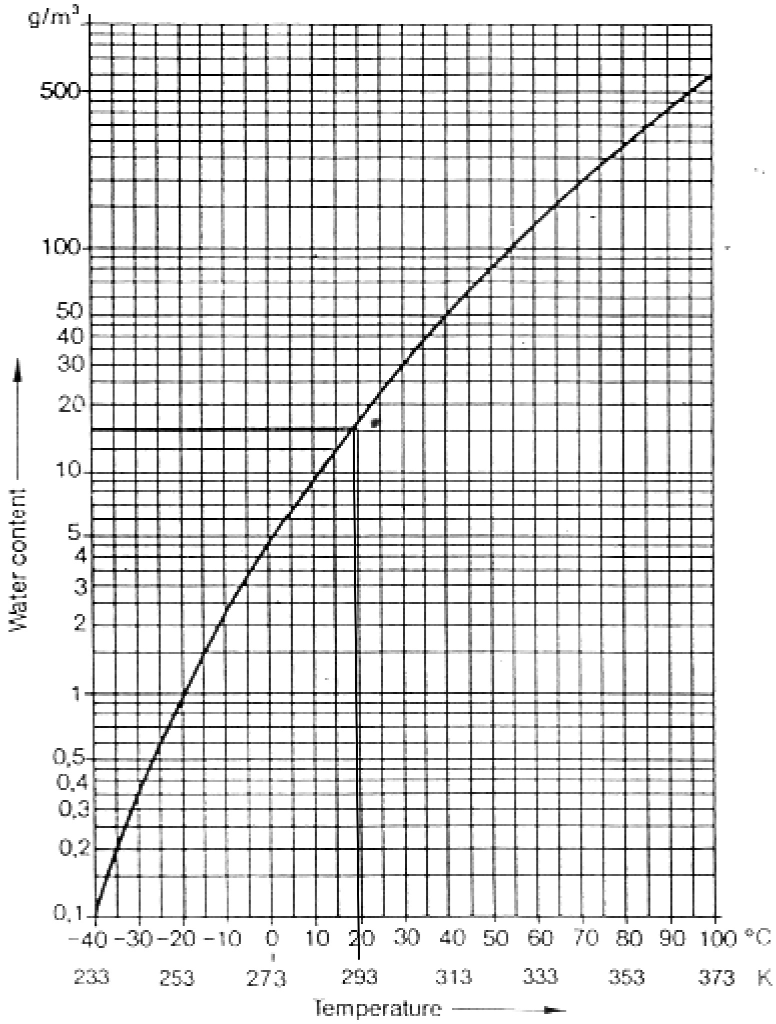


Figure 1.7: Dew point chart (saturation chart)

For nomogram, at 20°C one cubic meter of compressed air will still contain 16 grams of moisture. It closely matches with typical values of water vapour saturation capacity of an air at 20°C given in Table 1.2

1.4 STORAGE OF COMPRESSED AIR

1.4.1 AIR RECEIVER

Receivers perform several functions in compressed air systems. Firstly, they provide a larger system capacity, which increases the cycle time of compressor control systems. This makes less difficult the elimination of unstable and overcorrecting control cycles.

The receiver also dampens pulsations from reciprocating compressors, acts as a reservoir to prevent excessively temporary pressure drop during sudden short-term demand, and can be used to smooth air flow through dryers, separators and other air conditioning equipment. Because the air entering the receiver is reduced in velocity and cooled, some of the moisture may condensate and fall to the bottom of the receiver where it can be removed by a valve or preferably, a trap. Such a receiver can reduce further the amount of moisture which must be removed by a subsequent drying stage. The receiver always equipped with a pressure relief valve.

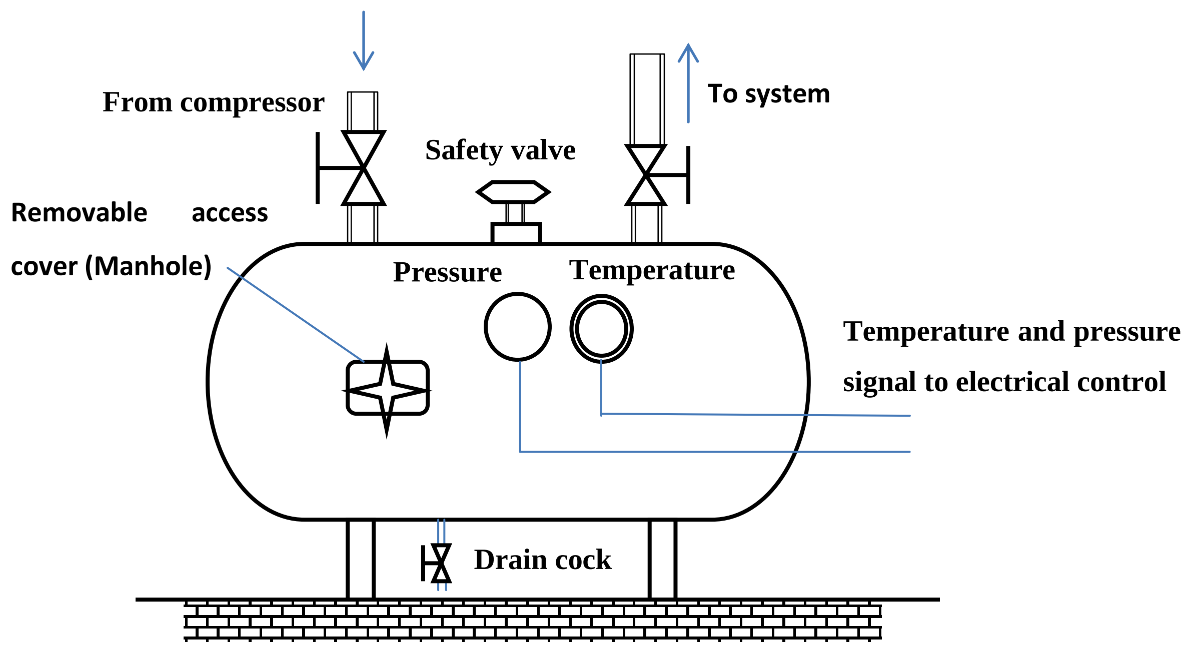


Figure 1.8 Components of Air receiver

Figure 1.8 shows essential features of a receiver. They are usually of cylindrical construction fabricated out of steel. Receiver is basically a pressure vessel and attracts the safety of testing requirement as per factory act. A safety valve is necessary to release the excess pressure of air stored. A pressure gauge and temperature gauges are provided and usually coupled with pressure switches to control on-off of the compressor and for remote alarms.

A drain cock allows removal of condensed water. Access via a manhole allows cleaning. Obviously, removal of manhole cover is hazardous with a pressurized receiver and safety routines must be defined and followed to prevent accidents.

Installation of accumulator within a pneumatic system will depend on the specific air consumers and will only be necessary when large volume of air are consumed in short periods, that is intermittent peak load.

Figure 1.9 shows diagram of compressed air installation with air receiver and accumulator.

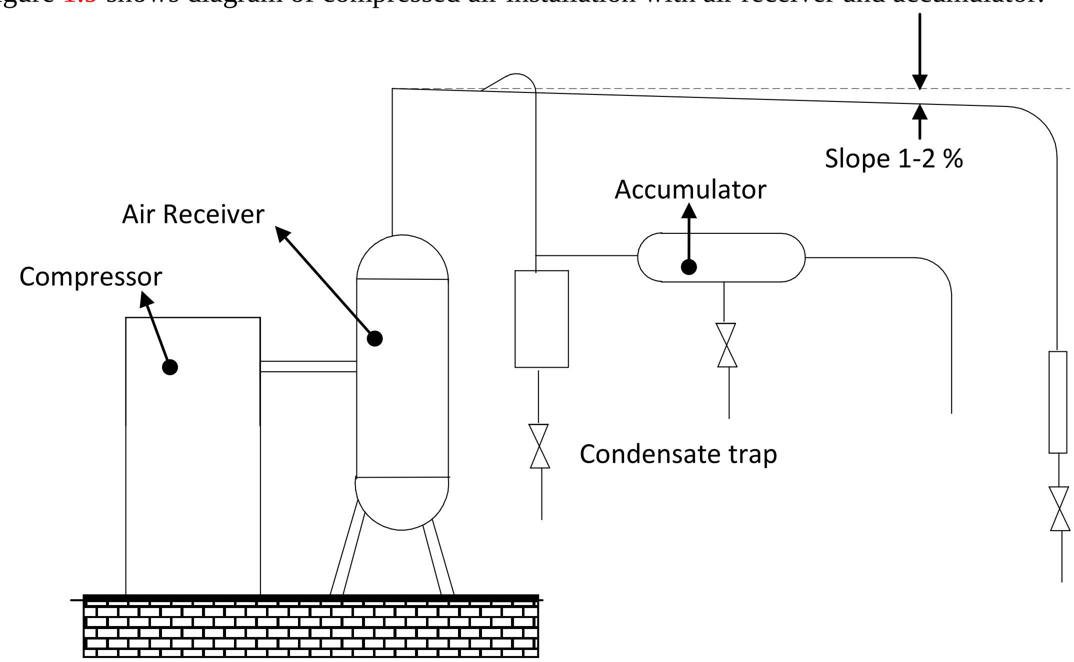


Figure 1.9 Diagram of compressed air distribution system with branch line with accumulator

1.4.2 Analysis sizing of Air Receivers

The receiver size in cubic feet normally should be atleast 12 seconds of compressor capacity in small units, to not less than 8 seconds of compressor capacity for larger units. Empirical rule suggest that the receiver volume should be 1/10 of FAD per minute to 1/6 FAD per minute for small compressor.

Sometimes small compressors are mounted upon a receiver and the receivers serve as a basic mounting frame for the assembly of the compressors and their accessories. Under these and some other circumstances, the only pressure relief valve frequently located upon the receiver. When this mounting procedure is employed, care must be taken to assure that no other valve is between the compressor and the receiver, and that support members or frames are not welded into the receiver tank seam welds.

1.4.2.1 Empirical relations to size the compressor

The sizing of air receivers requires taking into account parameters such as system pressure and flow rate requirements, compressor output capability, and the type of duty of operations. Basically a receiver is an air reservoir. The receiver must be capable of handing transient demand.

Case 1 : No air is supplied to the receiver during the time interval in which air is being drawn off

In such cases we can find the capacity of receiver as

$$V_r = \frac{101\ t\ [Q_r]}{[p_{max} - p_{min}]}$$

Case 2 : Air is supplied to the receiver during the time interval in which air is being drawn off

Then, The air receiver size can be determined by using the following empirical equation

$$V_r = \frac{101\ t\ [Q_r - Q_c]}{[p_{max} - p_{min}]}$$

V_r = receiver size (m^3)

t = time that receiver can supply required amount of air , (min)

Q_r = consumption rate of pneumatic system (standard m^3/min)

Q_c = outflow rate of pneumatic system (standard m^3/min)

p_{max} = maximum pressure level in receiver (kPa)

p_{min} = maximum pressure level in receiver (kPa)

Case 3 : Compressor with offloading/loading regulation The following relation applies when dimensioning the receiver's volume. Note that this relation only applies for compressors with off loading/loading regulation

$$V_r = \frac{0.25\ T_r\ Q_{fad}}{f_{max}[p_{max} - p_{min}]T_c}$$

V_r = receiver size (liter)

T_r = temperatur of compressed air in receiver , (K)

Q_{fad} = compressor fad (l/second)

T_c = Maximum ambient temperature , compressor inlet temperature (K)

p_{max} = maximum pressure level in receiver (bar

p_{min} = maximum pressure level in receiver (bar)

f_{max} = maximum loading frequency , usually 1cycle every 30 seconds

1.4.2.2 Requirement on Air receiver according to Factory act 1973 section 39

Air receiver are classified as pressure vessels and become subject to periodical inspection and test under the factory acts. Every owner of an air receiver must be acquainted with the requirement of factory act relating to receiver, and a summary of these rules is given below-

- Every air receiver and its fittings shall be of sound construction
- Each air receiver under pressure shall be protected by a suitable safety valve adjusted to prevent the pressure exceeding the maximum working pressure
- Safety valve should be constructed so as to permit the air to escape without increasing the pressure beyond 10 percent above the blow off pressure with the compressor running at full capacity
- It must be fitted with an accurate pressure gauge
- I t must be fitted with a suitable appliances for draining the receiver
- It must be provided with a manhole , hand hole or other means to allow thorough cleaning of the interior
- Where there is more than one receiver in the factory bear a distinguishing mark which shall easily be visible
- Every receiver shall be thoroughly cleaned and examined at least once every twenty four months by a competent authority.
- All welded receivers must be hydraulically tested twice the working pressure
- Each receiver must be permanently marked with the following

a) Maker's identification number
b) Date of test
c) Specification number
d) Hydraulic test pressure
e) Maximum working pressure

1.4.2.3 Analysis of power requirement to drive compressor

Another important design consideration is to determine the power required to drive an air compressor to meet system pressure and flow rate requirements. The following empirical relationship can be used to find the theoretical power requirement.

$$Theoretical\ power = \frac{p_1 Q}{17.1}\left(\left(\frac{p_2}{p_1}\right)^{0.286} - 1\right)$$

$p_1 = Inlet\ pressure\ of\ compressor\ \ (kPa)$

$p_2 = Outlet\ pressure\ of\ compressor\ (kPa)$

$Q = flow\ rate\ (m^3/min)$

To find the actual power, the theoretical power is divided by the overall compressor efficiency.

$$Actual\ \ power = \frac{Theoretical\ power}{overall\ efficiency}$$

$$Actual\ power = \frac{\frac{p_1 Q}{17.1}\left(\left(\frac{\mathrm{p}_2}{\mathrm{p}_1}\right)^{0.286} - 1\right)}{Overall\ efficiency}$$

Problem 1.4 : Air is used at a rate of 1 m^3/min from a receiver at 40℃ and 1000 kPa (gauge). If the atmosphere pressure is 101 kPa (abs) and the atmospheric temperature is 20 ℃. How many m^3/min of free air (standard m^3/min) must the compressor provide?

Solution

$\mathrm{p}_2 = 1000\ \mathrm{kPa(gauge)} = 1101\ \mathrm{KPa(absoulte)}$

$\mathrm{p}_1 = 101\ \mathrm{KPa(absoulte)}$

$\mathrm{T}_2 = 40℃ = 40 + 273 = 313\ \mathrm{K}$

$\mathrm{T}_1 = 20℃ = 20 + 273 = 293\ \mathrm{K}$

$$V_2 = 1\frac{m^3}{min},$$

Using General gas law

$$\frac{p_1 V_1}{T_1} = \frac{p_2 \ \ V_2}{T_2}$$

$$\frac{101 \times V_1}{293} = \frac{1101 \times 1}{313}$$

Solving we get $V_1 = 10.20$ standard $\frac{m^3}{min}$

Problem 1.5 : a. Calculate the required size of the receiver that must supply air to pneumatic system consuming $0.850\ m^3/min$ for 10 minutes between 828 kPa and 690 kPa before the compressor resumes operation **b**. what size is required if the compressor is running and delivering at $0.170 m^3/min$

Solution:

The air receiver size can be determined by using the following equation

$$V_r = \frac{101\ t\ [Q_r - Q_c]}{[p_{max} - p_{min}]}$$

Part a

V_r = receiver size (m^3)

t = time that receiver can supply required amount of air , (min) = 10 min

$$Q_r = \text{consumption rate of pneumatic system} \left(\text{standard} \frac{m^3}{min}\right) = 0.850\ m^3/min$$

$$Q_c = \text{outflow rate of compressor} \left(\text{standard} \frac{m^3}{min}\right) = 0\ m^3/min$$

p_{max} = maximum pressure level in receiver (kPa) = 828 kPa

p_{min} = maximum pressure level in receiver (kPa) = 690 kPa

$$V_r = \frac{101 \times \ 10\ [0.850 - 0]}{[828 - 690]}$$

Solving we get $V_r = 6.22\ m^3$

Part b

The required size of the compressor when the compressor is running and delivering air at $0.170 m^3/min$

$$\mathrm{V_r} = \frac{101 \times\ 10\,[0.850 - 0.170]}{[828 - 690]}$$

Solving we get $\mathrm{V_r} = 4.977\ \mathrm{m^3} \cong 5\mathrm{m^3}$

Objective Type Questions

1. About ------ of the moisture can be removed using after cooler

2. Adsorption is a --------- process of moisture removal on the porous surface of certain granular materials.

3. Absorption drying is a purely ------- process. The moisture in the compressed air forms a compound with drying agent like phosphoric pentaoxide in the tank

4. -------------- is to lower the dew point of the compressed air by removing the moisture from it

5. Temperature representing the dew point at the respective operating pressure of the dryer is known as ---------- dew point.

State True or False

1. In adsorption process gaseous molecules are attracted to certain solid surfaces by van der walls forces

2.. High Pressure dew point values indicate small amounts of water vapour in the compressed air.

3. It is important to remember that atmospheric dew point can be compared with PDP when comparing different dryers.

4. Installation of accumulator within a pneumatic system will depend on the specific air consumers and will only be necessary for intermittent peak load.

5. Membrane dryers do not require electric supply

Review Questions

1. State the adverse effects of moisture content compressed air

2. What is the function of an after cooler?

3. What are the different methods of drying compressed air

4. Explain the working of absorption dryer with a neat sketch

5. Explain the working of adsorption dryer with a neat sketch

6. Explain the working of regenerative dryer with a neat sketch

7. Why two drying chambers are used in adsorption dryer

8. Explain the functional and constructional features of a refrigerated dryer

9. List the advantages and disadvantages of absorption dryer

10. List the advantages and disadvantages of membrane dryer

11. List the advantages and disadvantages of refrigeration dryer

12. List five functions of Air receiver

13. Why must receiver tank be drained periodically?

14. Name three fittings used on all air receiver

15. What are the important maintenance activities to be carried out on air receiver

16. What are thumb rules we follow in selecting air receiver

Answers

Fill in the Blanks

1. 70-85%
2. physical
3. chemical
4. After cooler
5. Pressure

State True or False

1. True
2. False
3. False
4. True
5. True

Lecture 36

CONDITIONING AND DISTRIBUTION OF COMPRESSED AIR.

Learning Objectives

Upon completion of this chapter, Student should be able to

- Explain the importance of fluid conditioners
- List the major functions of filter and selection criteria
- List application of the seven qualities of air as per ISO
- Describe the function of various types of air regulators
- List the function of FRL unit
- Explain the various compressed air net work system
- List the factors to be considered in selection of components for pneumatic network
- Size the pipe diameter, pressure drop for a given pneumatic network

1.1 FLUID CONDITIONERS

Operating instructions issued for pneumatic components almost always contain a note recommending the installation of an air filter, pressure regulator and lubricator upstream of the component. This is to ensure that only air which has been suitably conditioned will reach the consumer.

Air filter, pressure regulator and lubricator are now built as packaged combination known as service units. Other than the impurities that might be entrained with the intake air and delivered by the compressor air might pick up contaminants such as dust, scale or rust particles in the distribution main leading to the take-off point. Provided that air main has been properly installed, the major part of these impurities will collect in the condensate drain tanks. Minute particles remain suspended in the air stream however and would damage the working parts of pneumatic components by their abrasive action were they are removed beforehand. Furthermore the air flow in the main pulsated, due, for one thing, to the compressor running intermittently as controlled by pressure in air receiver. The consumer, on the other hand, need to work with a uniform air pressure. Finally, lubrication is required for the moving parts of the pneumatic equipment.

The atmospheric air that is compressed in the compressor is obviously not clean because the atmospheric air contains many contaminants line dirt , smoke water vapour etc. this contaminated air lead to excessive wear and failure of pneumatic components. Therefore fluid conditioners are used to supply clean dry and contamination free compressed air.

The purpose of the fluid conditioners is to make the compressed air more acceptable and suitable fluid medium for the pneumatic system as well as the operating personal. The following five fluid conditioners are used in pneumatic systems

1. Air Filters

2. Air Regulators

3. Air Lubricator

4.1.1 AIR FILTERS

The purpose of the air filter is to clean the compressed air of all impurities and any condensate it contains.

a) Function of air filters

- To remove all foreign matter and allow dry and clean air flow without restriction to regulator and then to the lubricator
- To condensate and remove water from the air
- To arrest fine particles and all solid contaminants from air

Filters are available in wide range starting from a fine mesh wire cloth (which strains heavy foreign particles) to elements made of synthetic material (which removes very small particles)

Usually in line filter elements can remove contaminants in the 5-50 micron range.

b) Source of contamination.

Contaminants in a compressed air system can generally be attributed to the following:

- ***The quality of air being drawn into the compressor***: Air compressors draw in a large volume of air from the surrounding atmosphere containing large numbers of airborne contaminants.
- ***The type and operation of the air compressor***: The air compressor itself can also add contamination, from wear particles to coolants and lubricants.
- ***Compressed air storage devices and distribution systems***: The air receiver and system piping are designed to store and distribute the compressed air. As a consequence, they will also store the large amounts of contaminants drawn into the system.

Additionally, piping and air receivers will also cool the moist compressed air forming condensate which causes damage and corrosion.

c) Types of contamination in a compressed air system

- ***Atmospheric dirt:*** Atmospheric air in an industrial environment typically contains 140 million per m^3 of dirt particles. 80% of these particles are less than 2 microns in size and are too small to be captured by the compressor intake filter, therefore passing directly into the compressed air system
- ***Water vapour, condensed water and water aerosols:*** Atmospheric air contains water vapour (water in a gaseous form). The ability of compressed air to hold water vapour is dependent upon it's temperature. The higher the temperature, the more water vapour that can be held by the air. During compression, the air temperature is increased significantly, which allows it to easily retain the incoming moisture. After the compression stage, air is normally cooled to a usable temperature. This reduces the airs ability to retain water vapour, resulting in a proportion of the water vapour being condensed into liquid water which is removed by a condensate drain fitted to the compressor after-cooler. The air leaving the after-cooler is now 100% saturated with water vapour and any further cooling of the air will result in more water vapour condensing into liquid water. Condensation occurs at various stages throughout the

system as the air is cooled further by the air receiver, piping and the expansion of valves, cylinders, tools and machinery. The condensed water and water aerosols cause corrosion to the storage and distribution system, damage production equipment and the end product. It also reduces production efficiency and increases maintenance costs. Water in any form must be removed to enable the system to run correctly and efficiently.

- ***Rust and pipe scale:*** Rust and pipe scale can be found in air receivers and the piping of "wet systems" (systems without adequate purification equipment) or systems which were operated "wet" prior to purification being installed. Over time, this contamination breaks away to cause damage or blockage in production which can also contaminate final product and processes.
- ***Micro-organisms:*** Bacteria and viruses will also be drawn into the compressed air system through the compressor intake and warm, moist air provides an ideal environment for the growth of micro-organisms. If only a few micro-organisms were to enter a clean environment, a sterile process or production system, enormous damage could be caused that not only diminishes product quality, but may even render a product entirely unfit for use and subject to recall.
- ***Liquid oil and oil aerosols:*** Most air compressors use oil in the compression stage for sealing, lubrication and cooling. During operation, lubricating oil is carried over into the compressed air system as liquid oil and aerosols. This oil mixes with water vapour in the air and is often very acidic, causing damage to the compressed air storage and distribution system, production equipment and final product.
- ***Oil vapour:*** In addition to dirt and water vapour, atmospheric air also contains oil in the form of unburned hydrocarbons. The unburned hydrocarbons drawn into the compressor intake as well as vaporized oil from the compression stage of a lubricated compressor will carry over into a compressed air system where it can cool and condense, causing the same contamination issues as liquid oil.

d) Factor affecting selection of filters

While selecting the filters, the following factors should be taken into account.

- Size of particles to be filtered from the system
- Capacity of the filter
- Accessibility and maintainability
- Life of the filter
- Ability to drain the condensate

e) Construction

The construction of typical cartridge type filter along with graphical symbols is shown in Figure 1.1. It consists of filter cartridge, Deflector, bowl, water drain valve. Filter bowl is usually made of plastic and transparent. For pressure more than 10 bar, bowl may be made of brass.

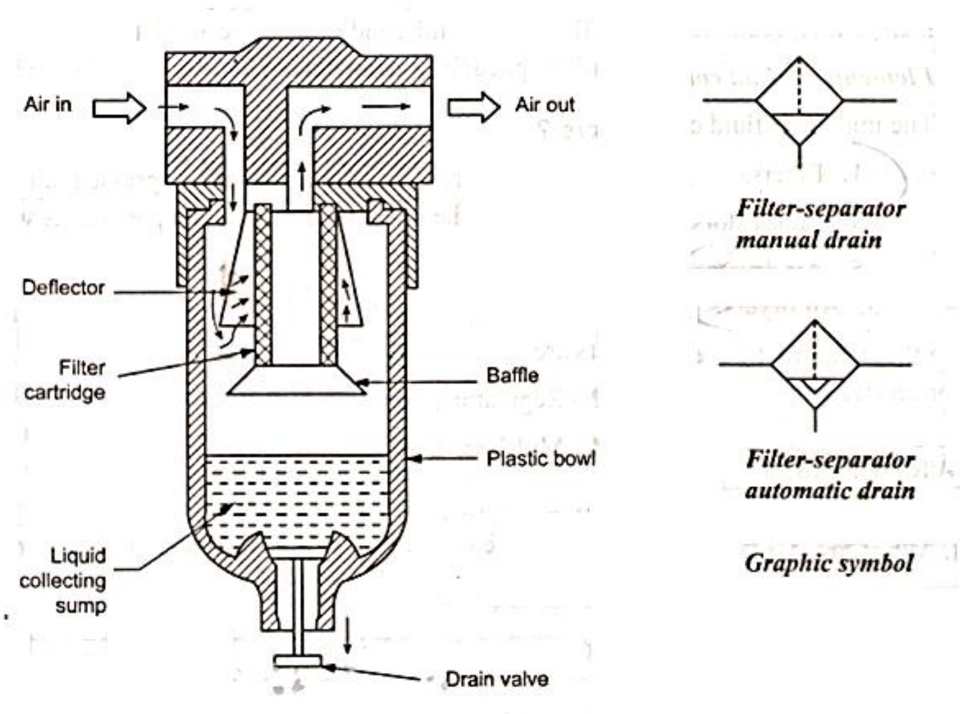


Figure 1.1 Construction of a Air filter.

f) Operation

Air enters the inlet port of the air filter through angled louvers. This causes the air to spin as it enters the bowl. The centrifugal action of the rotating air causes the larger pieces of dirt and water particles to be thrown against the inner wall of the filter bowl. These contaminants then flow down into the bottom of the filter bowl.

A baffle prevents turbulent air from splashing water on to the filter element. The air, which has been pre-cleaned in this way, then passes through the filter element, where the fine dirt particles are filtered out. The size of the dirt particles which can be filtered out depends on mesh size of filter element (usually 5-50 microns). The compressed air then exits through the outer port.

The pressure difference between inlet and outlet will indicate the degree to which the filter element is clogged. Commercially available filters have many additional features like automatic drain facility, coalescing type filter element, service life indicator etc.

g) Seven Quality levels of air required in production systems.

Figure 1.2 illustrates different levels of purity for various applications. Air from a compressor passes through an after cooler with an auto drain to remove condensate. As the air cools further in the air receiver more condensate is removed by an auto drain, installed on the bottom. Additional drains may be fitted to all low points on the pipeline. The system divides into three main parts:- Branches (1 and 2) provide air direct from the air receiver. Branches (3 – 6) use air conditioned by a refrigerated type of dryer. Branch 7 incorporates an additional dryer of the adsorption type. Standard filters in sub branches 1 and 2, equipped with auto drains remove condensate; sub-branch 2 being higher purity because of the micro filter. Sub branches 3 – 5, use refrigerated dry air, thus, branch 3 requires no auto drain, branch 4 needs no pre filtering and branch 5 gives an improved level of air purity using a micro filter and sub micro filter, the moisture having been removed by a refrigerated type of air dryer. Sub branch 6 incorporates an odour removal filter. An adsorption type dryer eliminates all risk of condensation at low temperatures in sub branch 7. Typical applications are listed in **Table 1.1**

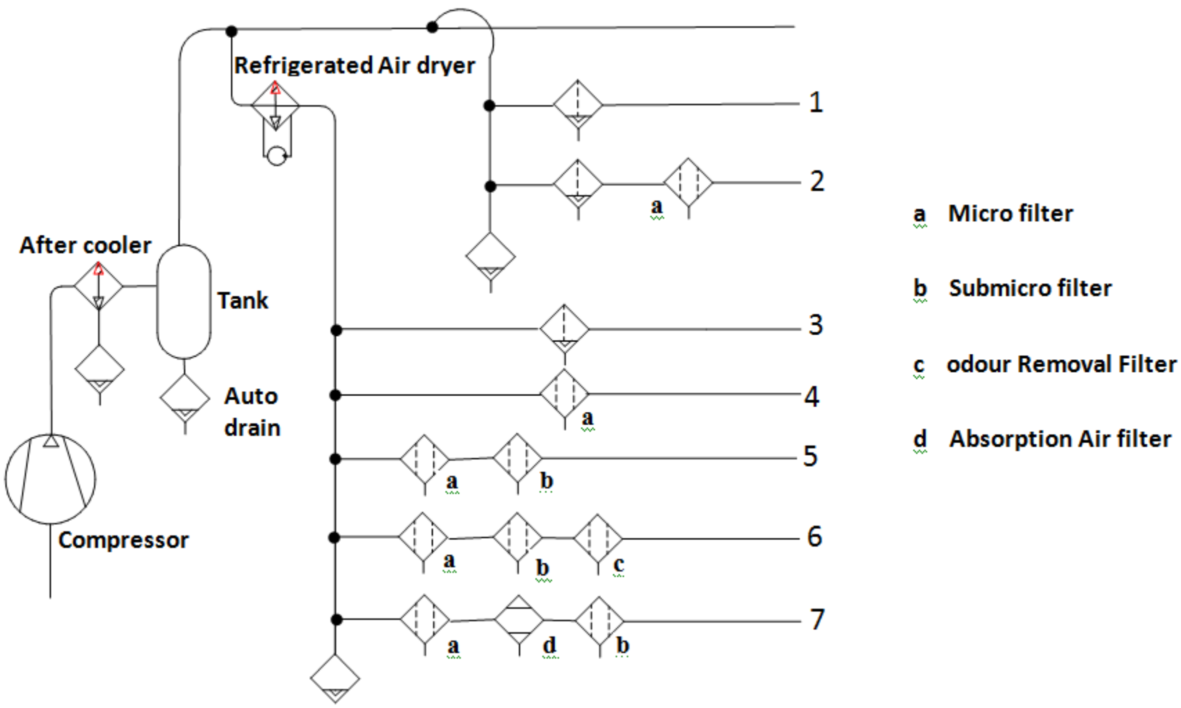


Figure 1.2 Schematic Definition of 7 Degrees of Filtration

Table 1.1 Definition and typical applications of the seven qualities of air.

No.	Removal of	Application	Typical examples
1	Dust particles > 5 micron Liquid oil to 99% Saturated humidity to 96%	Where some solid impurities humidify and oil can be accepted	Workshop air for clamping blowing and simple pneumatic drives
2	Dust particles > 0.3 micron Oil mist to 99.9% Saturated humidity to 99%	Where removal of dust and oil dominates, but a certain amount of condensation can be risked	General industrial equipment, pneumatic controls and drives, seamless metallic joints, air tools and air motors
3	Humidity to atmosphere dew point of -17°C	Where removal of humidity is imperative	Similar to (1) as the air is dry additional spray painting

	Dew point Saturated humidity to 99%	but traces of fine dust and oil are acceptable	
4	Dust particles > 0.3 micron Oil mist to 99.9% Humidity up to an atmospheric dew point of -17℃	Where no humidity, fine dust and oil vapour are acceptable	Process control, measuring equipment , high quality spray painting, cooling of foundry and injection moulding dies
5	Dust particles > 0.3 micron Oil mist to 99.9999% Humidity up to an atmospheric dew point of -17℃	Where pure air , practically free from any impurity is required	Pneumatic precision measuring devices, electrostatic spray painting, cleaning and drying of electronic assemblies
6	As in (5) with odour removal	Where absolutely pure air, as under (5) but odour free	Pharmacy , food industry for packaging , air transport, brewing and breathing air
7	All impurities as in (6) but with atmospheric dew point of greater than -30℃	Where risk of condensation during expansion and low temperature must be avoided,	Drying electronic components, storage of pharmaceuticals, marine measuring equipment, air transport of powder.

h) Main Line Filter

A large capacity filter should be installed after the air receiver to remove contamination, oil vapours from the compressor and water from the air. This filter must have a minimum pressure drop and the capability to remove oil vapour from the compressor in order to avoid emulsification with condensation in the line. It has no deflector, which requires a certain minimum pressure drop to function properly A built-in or an attached auto drain will ensure a regular discharge of accumulated water.

The filter is generally a quick change cartridge type. Figure 1.3 shows schematic diagram of Main line filter.

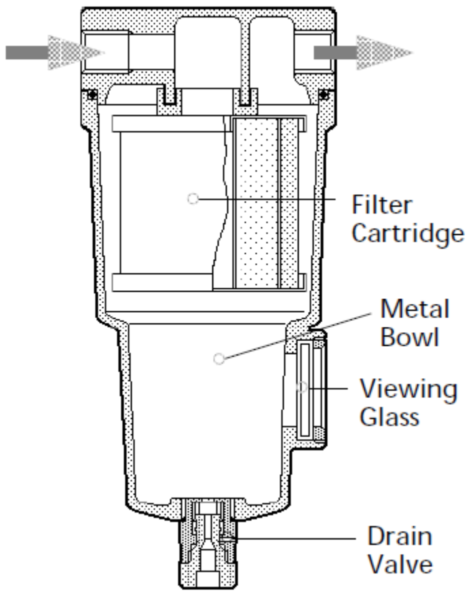


Figure 1.3 Typical main line filter

1.1.2 AIR REGULATOR

a) Function: The function of the air pressure regulator is to maintain working pressure virtually constant regardless of fluctuations of the line pressure and air consumption. When the pressure is too low, it results in poor efficiencies and when the pressure is too high, energy is wasted and equipment's performance decay faster.

In pneumatic system, pressure fluctuations occur due to variation in supply pressure or load pressure. It is therefore essential to regulate the pressure to match the requirement of load regardless of variation in supply pressure or load pressure.

b) Where to regulate

Generally pressure is regulated in pneumatic system at two places.

- At the receiver tank
- In the load circuits

Pressure regulation at the receiver tank is required as a safety measure for the system. In the load circuits, pressure regulator is used to regulate the pressure for downstream components such as valves and actuators.

c) Types of Pressure regulator

There are two types of Pressure regulators

i) Diaphragm type regulator

ii) Piston type regulator

Diaphragm type regulator is commonly used in Industrial pneumatic system. There are two types of diaphragm type regulator

i) Non- reliving or non-venting type.

ii) Relieving or venting type

Relieving or venting type is commonly used and is explained below.

1.1.2.1 Relieving or Venting Type Pressure regulator

A Relieving type pressure regulator is shown in **Figure 1.4**, Outlet pressure is sensed by a diaphragm preloaded with a adjustable pressure setting spring. The compressed air , which flows through a controlled cross section at the valve seat, acts on the other side of the diaphragm. The diaphragm has large surface area exposed to secondary (outlet) pressure and is quite sensitive to its fluctuations. The movement of diaphragm regulates the pressure.

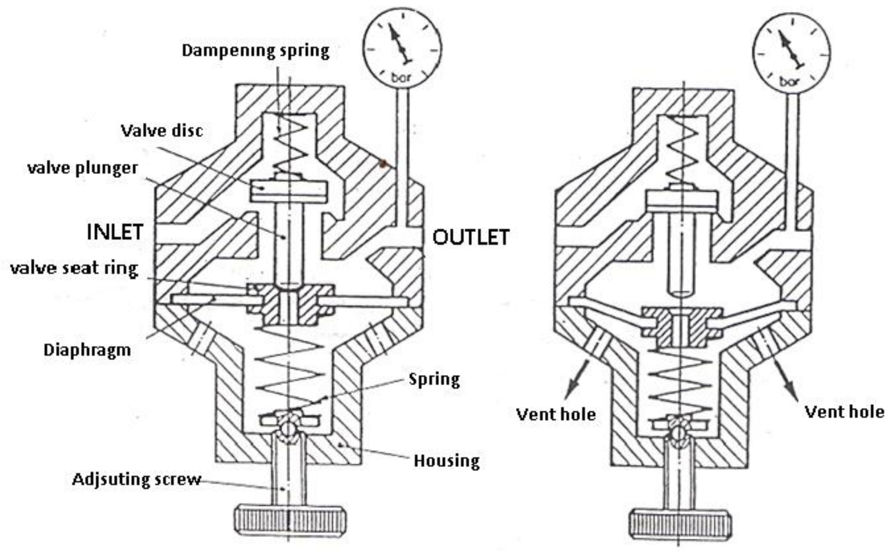


Figure 1.4 **venting type pressure regulator**

If the outlet pressure is low: whenever the more compressed air is consumed on secondary side or load side, then load pressure reduces. Therefore less force acts on diaphragm. The opposing higher spring force pushes the diaphragm in such a way as to move the valve disc more and permitting more air to flow to secondary side and thus increasing the pressure again.

If the outlet pressure is high: whenever the less compressed air is consumed on secondary side or load side, then load pressure increases. Therefore more force acts on diaphragm. The opposing higher spring force pulls down the diaphragm in such a way as to move the valve disc less and permitting air to flow to vent hole and thus decreasing the pressure again

1.1.2.2 Non-Relieving or Non-Venting Type Pressure regulator

In this case compressed air cannot escape to the atmosphere in the event of high backpressure acting on the diaphragm, as there is no exit path provided in the diaphragm for the trapped air. **Figure 1.5** shows the non –relieving venting type pressure regulator.

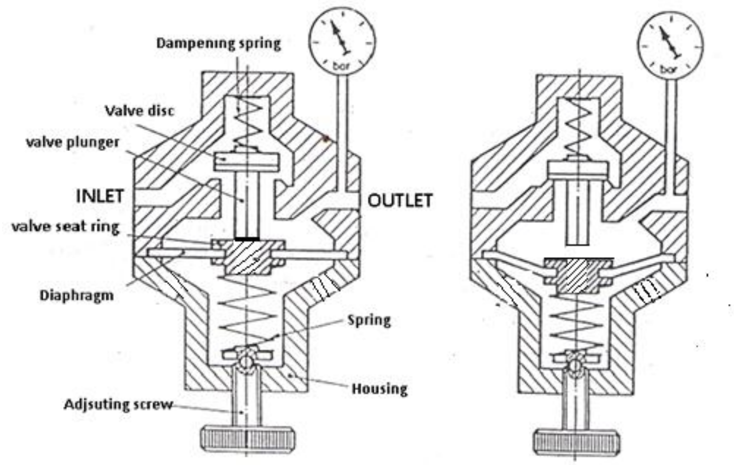


Figure 1.5 **Non-venting type pressure regulators**

1.1.3 AIR LUBRICATOR

Function: The function of air lubricator is to add a controlled amount of oil with air to ensure proper lubrication of internal moving parts of pneumatic components. Lubricants are used to

- To reduce the wear of the moving parts
- Reduce the frictional losses
- Protect the equipment form corrosion

The lubricator adds the lubricating oil in the form of fine mist to reduce the friction and wear of moving parts of pneumatic components such as valves, packing used in air actuators

Excessive lubrication is undesirable. Excessive lubrication may results in

- malfunctioning of components,
- seizing and sticking of components after prolonged downtime
- environmental pollution

Operation: The operation is similar to the principle of the carburettor. Schematic diagram is shown in Figure 1.6. As air enters the lubricator its velocity is increased by a venture ring. The pressure at the venture ring will be lower than the atmospheric pressure and the pressure on the oil is atmospheric. Due to this pressure difference between the upper chamber and lower chamber, oil will be drawn up in a riser tube. Oil droplets mix with the incoming air and form a fine mist. The needle valve is used adjust the pressure differential between across the oil jet and hence the oil flow rate. The air – oil mixture is forced to swirl as it leaves the central cylinder so that large particles of oil is goes back to bowl and only the mist goes to outlet.

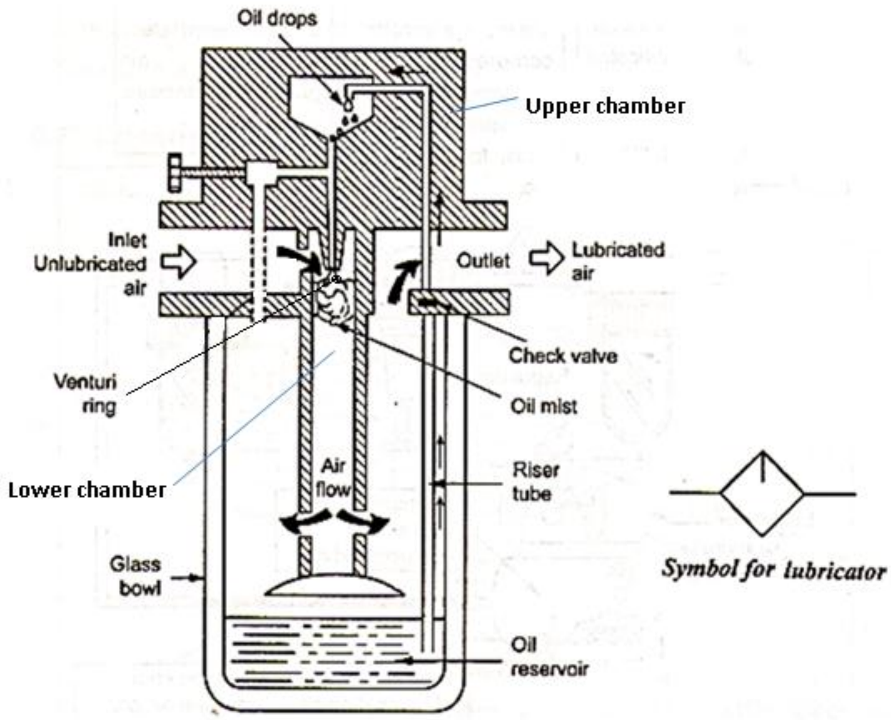


Figure 1.6 **Air lubricator**

The lubricator starts to operate only when there is sufficient flow of air. If too little air is drawn off, the flow velocity at the nozzle is not sufficient to produce an adequate vacuum and hence to draw oil out of the vessel. Only thin mineral oil may be used in pneumatic system lubricator. Viscosity ratings are normally 10-50 Centistokes or SAE 10 . Table 1.2 give the normally used oil. The list is purely alphabetic and not in order of preference.

Table 1.2 Typical oils used in air lubricator

Suitable oil grades/Trade name	**Viscosity at 20 ℃**
ARAL OEL TU 500	23.6 cSt
Avia Avilub RSL 3	34 cSt
BP Energol HL 40	27 cSt
ESSO SPINESSO 34, Nutto H5, H10	23 cSt
Mobil Vac HLP 9, Velocite oil no 6	25.3 cSt
Shell TELLUS OEL 15, OL 10	22 cSt
TEXACO Rando oil AAA	25 cSt
VALVOLINE RITZOL R-60	26 cSt
Vedol Andarin 38	20.5 cSt
Aral , Vitamol, GF10, DE10, CM5, CM10	21 cSt

1.1.4 Filter Regulator Lubricator Unit (FRL Unit) /Service Unit

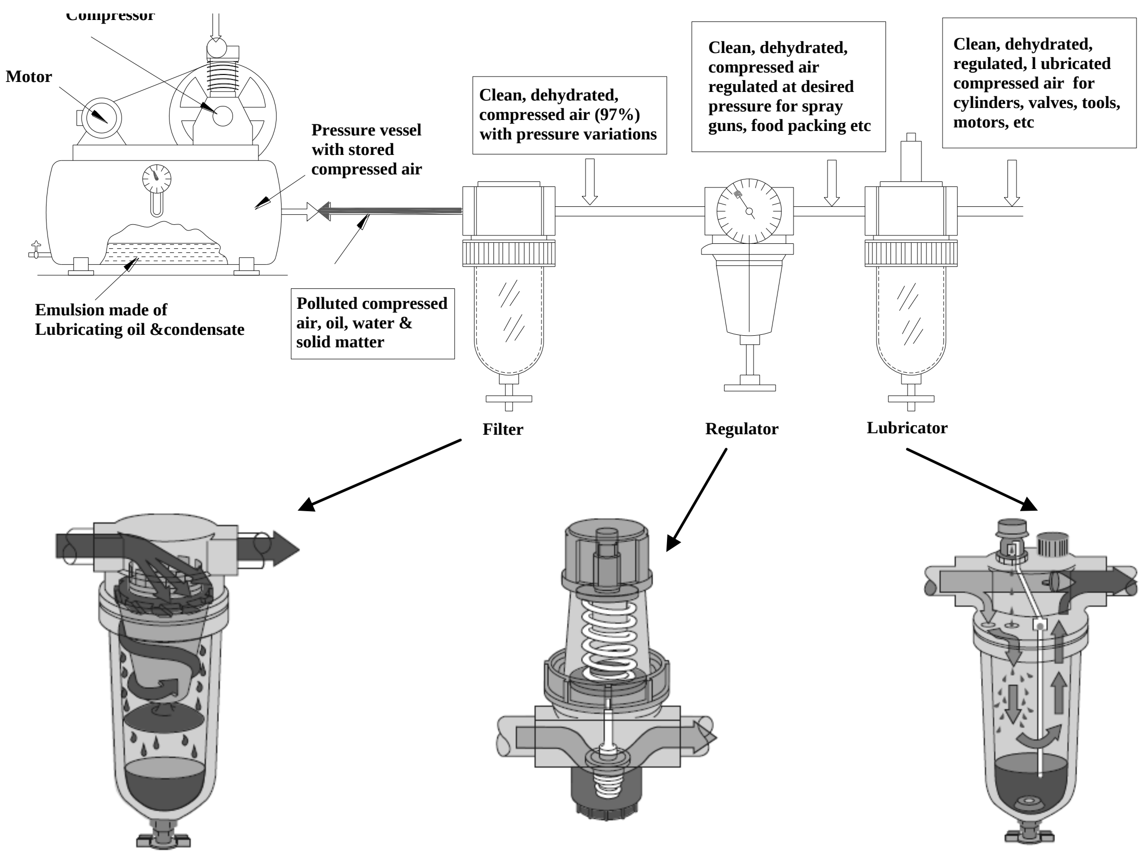


Figure 1.7 Installation of FRL unit

In most pneumatic systems, the compressed air is first filtered and then regulated to the specific pressure and made to pass through a lubricator for lubricating the oil. Thus usually a filter, regulator and lubricator are placed in the inlet line to each air circuit. They may be installed as separate units, but more often they are used in the form of a combined unit. Figure 1.6 shows the schematic arrangement of installation of Filter, Regulator and Lubricator unit .

The combination of filter, regulator and lubricator is called FRL unit or service unit. Figure 1.7 (a) gives the three dimensional view of FRL unit. Figure 1.7(b) gives detailed symbol of FRL unit. Figure 1.7(c) gives simplified symbol of FRL unit.

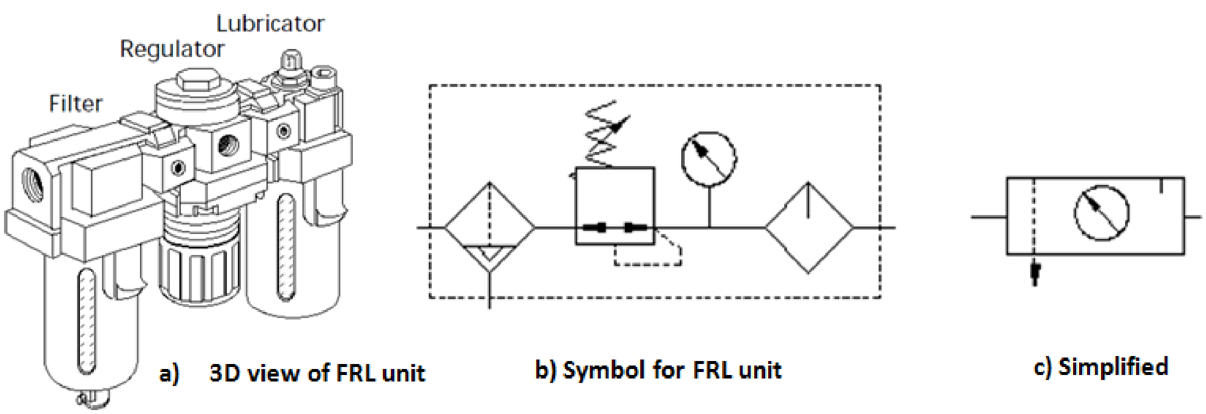


Figure 1.7 a) Three dimensional view of FRL unit b) detained symbol c) simplified symbol of FRL

1.2 AIR DISTRIBUTION SYSTEM

The main objective of air distribution system is to provide a distribution channel for compressed air without any leak and keep the pressure drop within permissible limits. The air distribution system consists of conductors and fittings which interconnect various components of a pneumatic system. Figure 1.8 shows a typical air distribution system. It consists of compressor, water cooled after cooler, Air receiver, dryer and ring main system. The air main takes the shape of a ring. Air from main header is drawn by sub headers. Sub headers may have its own accumulators.

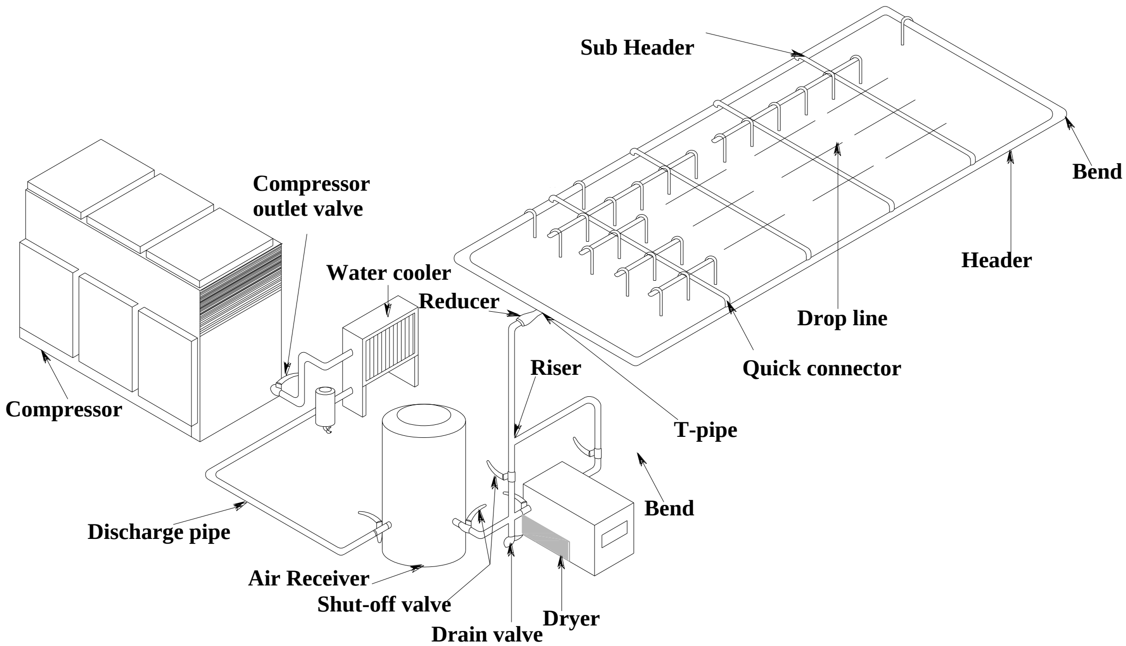


Figure 1.8 : Typical Air distribution system (Ring type)

The air distribution should take into account the following parameters

1. Choice of fluid conductor
2. Flow resistance
3. Correct sizing of pipes
4. Correct sizing of fittings.
5. Pipe layout

Objective Type Questions

1. Air filter, pressure regulator and lubricator are now built as packaged combination known as FRL unit or ---------- units.
2. The pressure difference between inlet and outlet will indicate the degree to which the air filter element is ----------

3. In pneumatic system, pressure fluctuations occur due to variation in ---------- pressure or load pressure.
4. The main objective of air distribution system is to provide a distribution channel for compressed air without any leak and keep the --------- within permissible limits.
5. Equivalent pipe length is length of straight pipe of the same -------- size giving the same pressure drop

State True or False

1. Usually in line filter elements can remove contaminants in the 0.005-0.0005 micron range

2. A manifold type pneumatic network has the advantages that, being in the form of closed circuit, the velocity of the air the main will be reduced and the pressure drop will be less.
3. Two Quality levels of air required in pneumatic production and distribution systems.

4. In Non-Venting Type Pressure regulator case compressed air cannot escape to the atmosphere in the event of high backpressure acting on the diaphragm, as there is no exit path provided in the diaphragm for the trapped air.

5. A single, large air compressor is more efficient and less costly than the several smaller units if the demand is fairly constant

Review Questions

1. State the importance of fluid conditioning

2. Describe the function of air filter.

3. With help of neat sketch explain the working of air filter.

4. State clearly nine qualities of filtered air requirement and its application

5. Describe the function of an air pressure regulator

6. With the help of neat sketch explain the working of air regulator

7. Difference between venting and non-venting type of pressure regulator.

8. Describe the function of an air lubricator

9. With the help of neat sketch explain the working of air filter.

10. List commonly used oil in an air lubricator

11. State the advantage of main ring pneumatic net work

12. List five important considerations in pipe layout in pneumatic network

13. With the help of neat sketch, explain ring and manifold type of pneumatic network

14. List the guidelines for selection of pneumatic components and compressor

15. Discuss the three ways to remove the water from the air in the air distribution system

Answers

Fill in the Blanks
1. service
2. clogged
3. supply
4. pressure drop
5. diameter

State True or False

1. False
2. False
3. False
4. True
5. True

Lecture 37

PNEUMATICS ACTUATORS

Learning Objectives

Upon completion of this chapter, Student should be able to

- Explain the meaning of Pneumatic Actuator
- Classify the various types of Pneumatic actuators
- Explain the working of various pneumatic actuators
- Understand the importance of cushioning
- Study the seals used in the Pneumatic actuators
- Understand the working of rodless cylinders
- List advantages and disadvantages of various cylinder mounting
- Study the effect of buckling in cylinders
- Explain the various lever systems used with pneumatic cylinders
- List the various applications of different lever systems using actuators
- Explain the working of limited angle rotary actuators.
- Explain the working and application of air motors

1.1 PNEUMATICS ACTUATORS

Pneumatic actuators are the devices used for converting pressure energy of compressed air into the mechanical energy to perform useful work. In other words, Actuators are used to perform the task of exerting the required force at the end of the stroke or used to create displacement by the movement of the piston. The pressurised air from the compressor is supplied to reservoir. The pressurised air from storage is supplied to pneumatic actuator to do work.

The air cylinder is a simple and efficient device for providing linear thrust or straight line motions with a rapid speed of response. Friction losses are low, seldom exceeds 5 % with a cylinder in good condition, and cylinders are particularly suitable for single purpose applications and /or where rapid movement is required. They are also suitable for use under conditions which preclude the employment of hydraulic cylinders that is at high ambient temperature of up to 200 ℃ to 250 ℃

Their chief limitation is that the elastic nature of the compressed air makes them unsuitable for powering movement where absolutely steady forces or motions are required applied against a fluctuating load, or where extreme accuracy of feed is necessary. The air cylinder is also inherently limited in thrust output by the relatively low supply pressure so that production of high output forces can only be achieved by a large size of the cylinders.

1.2 TYPES OF PNEUMATICS ACTUATORS

Pneumatic cylinders can be used to get linear, rotary and oscillatory motion. There are three types of pneumatic actuator: they are

i) Linear Actuator or Pneumatic cylinders
ii) Rotary Actuator or Air motors
iii) Limited angle Actuators

1.2.1 Types of Pneumatic cylinders /Linear actuators

Pneumatic cylinders are devices for converting the air pressure into linear mechanical force and motion. The pneumatic cylinders are basically used for single purpose application such as clamping, stamping, transferring, branching, allocating, ejecting, metering, tilting, bending, turning and many other applications.

The different classification scheme of the pneumatic cylinders are given below

1. Based on application for which air cylinders are used

i) Light duty air cylinders

ii) Medium duty air cylinders

iii) Heavy duty air cylinders

2. Based on the cylinder action

i) Single acting cylinder

ii) Double acting cylinder

- Single rod type double acting cylinder
- Double rod type double acting cylinder

3. Based on cylinder's movement

i) Rotating type air cylinder

ii) Non rotating type air cylinder

4. Based on the cylinder's design

i) Telescopic cylinder

ii) Tandem cylinder

iii) Rod less cylinder

- Cable cylinder,
- Sealing band Cylinder with slotted cylinder barrel
- Cylinder with Magnetically Coupled Slide

iv) Impact cylinder

v) Duplex cylinders

vi) Cylinders with sensors

1.2.1.1 Based on application for which air cylinders are used

Air cylinders can be classified according to their intended use, as light duty, medium duty or heavy duty types. In the main this merely governs the strength of the cylinder, and thus typical choice of material of construction and the form of construction. Comparison is given in Table 1.1. It should be noted that classification by duty does not necessarily affect the output performance of the cylinder, as bore size for bore size; identical cylinder diameter will give the same thrust on the same line pressure, regardless of whether the cylinder is rated for light, medium or heavy duty. This form of rating , however, normally precludes the use of light classification for cylinders of large size (and thus high thrust) ; and medium classification for cylinders of even large size and very high thrust outputs.

All plastic construction has the advantage of being inherently free from corrosion and similar troubles but, in general is limited to smaller cylinder sizes and light duty applications. As originally introduced they were intended to provide low cost cylinders for light duty work , and where rigidity of the unit was not an important factor. The development of all-plastic cylinders for higher duties tends to nullify any cost advantage and the types has not, as yet, achieved any particular prominence, although the potentialities remain for corrosion- resistant duties.

Force limitation with air cylinders are purely matter of size and cost. Since line pressures available are usually very much lower than pressure common in hydraulic circuits, air cylinder must be very much larger in diameter than the hydraulic cylinders for the same thrust performance. Where a very high force is required the cost of the suitable size of air cylinder may work out at more than the cost of a complete hydraulic system to do the same job. In addition the cost of the compressed air feed such cylinders could also be prohibitive.

Table 1.1: Materials of construction for light, medium and heavy duty cylinders

Components	Type of cylinder		
	Light duty	**Medium duty**	**Heavy duty**
Cylinder tubes	Hard drawn seamless aluminium or brass tubes Plastics	Hard drawn seamless brass tubes Aluminium , brass, iron or steel castings	Hard drawn seamless tubing , brass , bronze, iron or steel casting
End covers	Aluminium alloy castings Fabricated aluminium , brass , bronze	Aluminium brass, bronze, iron or steel castings, fabricated brass, bronze,	High tensile castings
Pistons	Aluminium alloy castings	Aluminium alloy castings, Brass, cast iron	Aluminium alloy castings, Brass, cast iron
Piston rods	EN 8 or similar steel ground and polished or chrome plated	EN 8 steel, ground and polished or chrome plated. Ground and polished stainless steel	Ground and polished stainless steel
Mounting brackets	Aluminium alloy casting	Aluminium,brass,iron castings	High tensile castings or fabricated

1.2.1.2 Based on the cylinder action

Based on cylinder action we can classify the cylinders as single acting and double acting. Single acting cylinders have single air inlet line. Double acting cylinders have two air inlet lines. Advantages of double acting cylinders over single acting cylinders are

1. In single acting cylinder, compressed air is fed only on one side. Hence this cylinder can produce work only in one direction. But the compressed air moves the piston in two directions in double acting cylinder, so they work in both directions
2. In a single acting cylinder, the stroke length is limited by the compressed length of the spring. But in principle , the stroke length is unlimited in a double acting cylinder
3. While the piston moves forward in a single acting cylinder, air has to overcome the pressure of the spring and hence some power is lost before the actual stroke of the piston starts. But this problem is not present in a double acting cylinder.

A) Single acting cylinders.

Single acting cylinder has one working port. Forward motion of the piston is obtained by supplying compressed air to working port. Return motion of piston is obtained by spring placed on the rod side of the cylinder. Schematic diagram of single acting cylinder is shown in Figure 1.1

Single acting cylinders are used where force is required to be exerted only in one direction. Such as clamping, feeding, sorting, locking, ejecting, braking etc.,

Single acting cylinder is usually available in short stroke lengths [maximum length up to 80 mm] due to the natural length of the spring. Single Acting Cylinder exert force only in one direction. Single acting cylinders require only about half the air volume consumed by a double acting cylinder for one operating cycle.

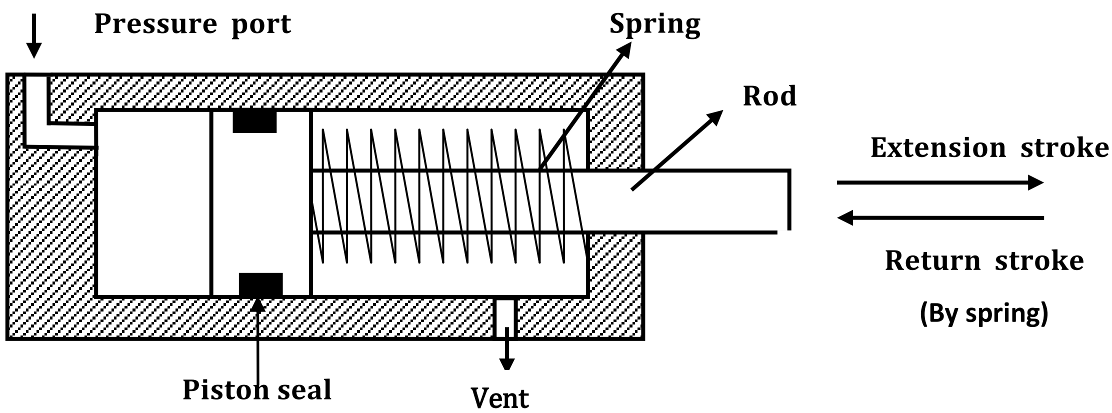


Figure 1.1 Construction features of single acting cylinder

There are varying designs of single acting cylinders including:

1. Diaphragm cylinder
2. Rolling diaphragm cylinder
3. Gravity return single acting cylinder
4. Spring return single acting cylinder

i) Diaphragm cylinder

This is the simplest form of single acting cylinder. In diaphragm cylinder , piston is replaced by a diaphragm is replaced by a diaphragm of hard rubber, plastic or metal clamped between the two halves of a metal casing expanded to form a wide, flat enclosure. Schematic diagram of diaphragm cylinder is shown in Figure 1.2.The operating stem which takes place of the piston rod in diaphragm cylinder can also be designed as a surface element so as to act directly as a clamping surface for example. Only short operating strokes can be executed by a diaphragm cylinder, up to a maximum of 50 mm. This makes the diaphragm type of cylinder particularly adaptable to clamping operations. Return stroke is accomplished by a spring built into the assembly or by the tension of diaphragm itself in the case of very short stroke. Diaphragm cylinders are used for short stoke application like clamping, riveting, lifting, embossing and riveting

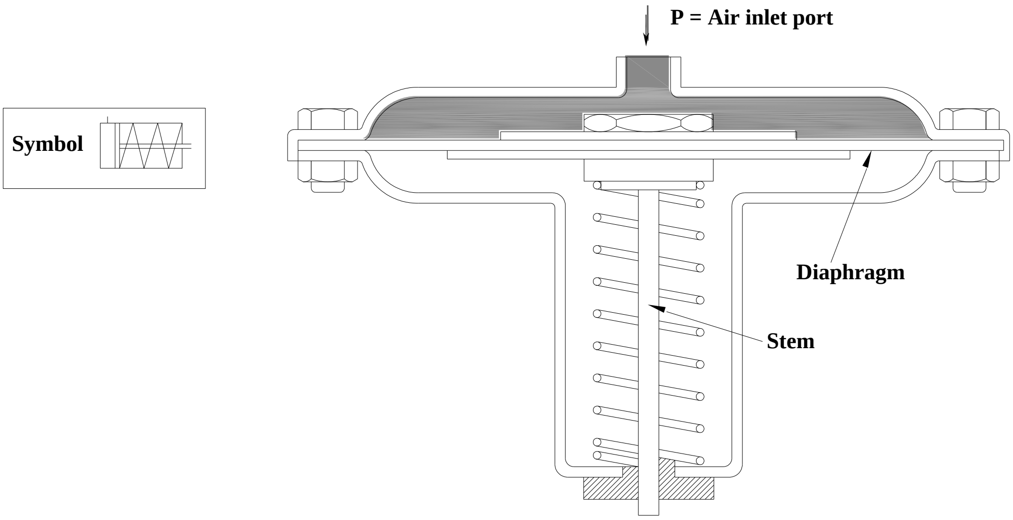


Figure 1.2 Construction features of diaphragm cylinder

ii) Rolling diaphragm cylinder

They are similar to diaphragm cylinders. Schematic diagram of Rolling diaphragm cylinder is shown in Figure 1.3.They too contain a diaphragm instead of piston, which is this instance rolls out along the inner walls of the cylinder when air pressure is applied to the device, thereby causing the operating stem to move outwards. Compared with the standard diaphragm type, a rolling diaphragm cylinder is capable of executing appreciably longer operating strokes (averaging from 50 mm to 800mm) . Separate guiding of stem is not normally provided in these designs, since the component being actuated by the cylinder usually cannot break out of set limits of motion. Any off-center displacement is compensated by the rolling diaphragm with no loss of power. Materials used for rolling diaphragms in present –day designs ensure good durability under normal operating conditions. On the other hand, even very small cracks or cuts in the diaphragm will generally lead to early failure because if high stresses are imposed on the flexible material as it unrolls at each stroke. If the actuator needs to be dismantled for any reason, it must accordingly be inspected carefully for any burrs or sharp edges inside. Metal cuttings also constitute a hazard if they are able to enter the cylinder housing.

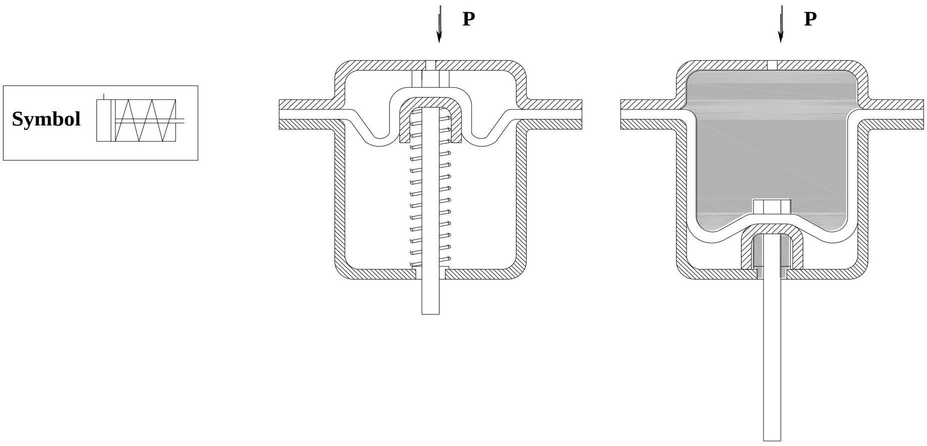


Figure 1.3 Construction features of rolling diaphragm cylinder

iii) Gravity Return Single Acting Cylinder

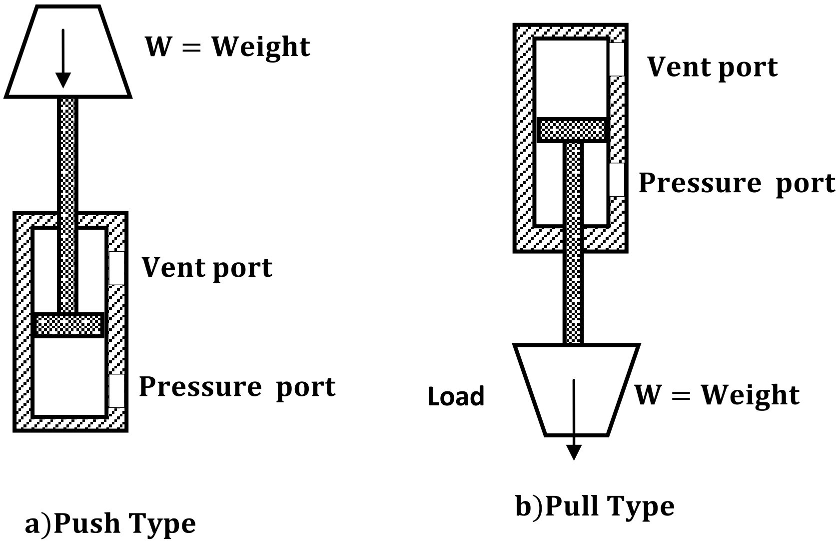


Figure 1.4 Gravity return Single Acting Cylinder

Figure 1.4 shows gravity return type single acting cylinders. In a push type (a), the cylinder extends to lift a weight against the force of gravity by applying oil pressure at the blank end. The oil is passed through blank end port or pressure port. The rod end port or vent port is open to atmosphere so that air can flow freely in and out of the rod end of the cylinder. To retract the cylinder, the pressure is simply removed from the piston by connecting the pressure port to the tank. This allows the weight of the load to push the fluid out of the cylinder back to tank.

In pull type gravity return type single acting cylinder the cylinder (b) lifts the weight by retracting. The blank end port is the pressure port and blind end port is now the vent port. This cylinder will automatically extend whenever the pressure port is connected to the tank.

iv) Spring Return Single Acting Cylinder

Spring return single acting cylinder is shown in Figure 1.5 in part (a) push type the pressure is sent through pressure port situated at blank end of the cylinder. When the pressure is released, the spring automatically returns the cylinder to the fully retracted position. The vent port is open to atmosphere so that air can flow freely in and out of the rod end of the cylinder.

Part (b) shows a spring return single acting cylinder. In this design cylinder retracts when the pressure port is connected to the pump flow and extend whenever the pressure port is connected to the tank. Here pressure port is situated at rod end of the cylinder.

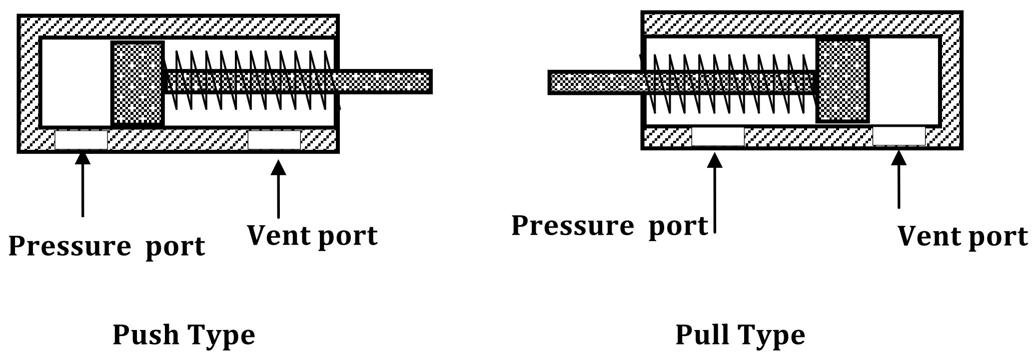


Figure 1.5 Push and Pull type Single Acting Cylinder

B) Double acting cylinders.

Schematic diagram of double acting cylinder is shown in Figure 1.6. Double Acting Cylinders are equipped with two working ports- one on the piston side and the other on the rod side. To achieve forward motion of the cylinder, compressed air is admitted on the piston side and the rod side is connected to exhaust. During return motion supply air admitted at the rod side while the piston side volume is connected to the exhaust. Force is exerted by the piston both during forward and return motion of cylinder. Double acting cylinders are available in diameters from few mm to around 300 mm and stroke lengths of few mm up to 2 meters

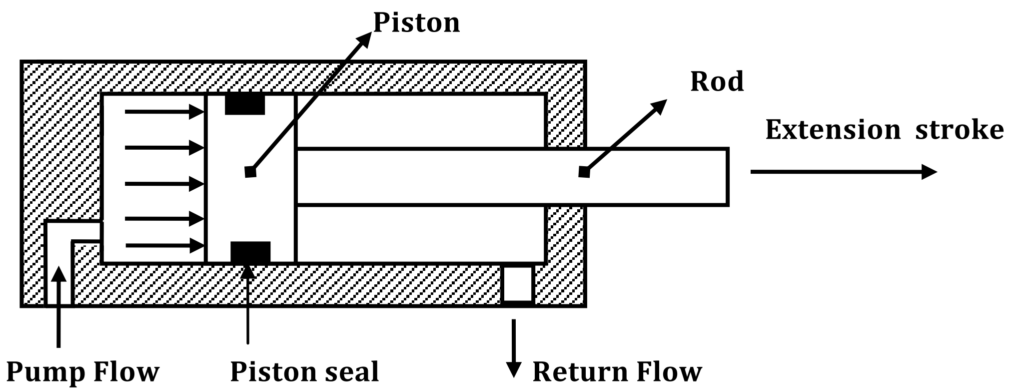


Figure 1.6 Double acting cylinder

Construction of Double acting cylinder

The construction features of double acting cylinder are shown in Figure 1.7. The construction of double acting cylinder is similar to that of a single cylinder. However, there is no return spring. In double acting cylinder, air pressure can be applied to either side (supply and exhaust) of the piston, thereby providing a pneumatic force in both directions. The double acting cylinders are mostly commonly used in the application where larger stroke length is required.

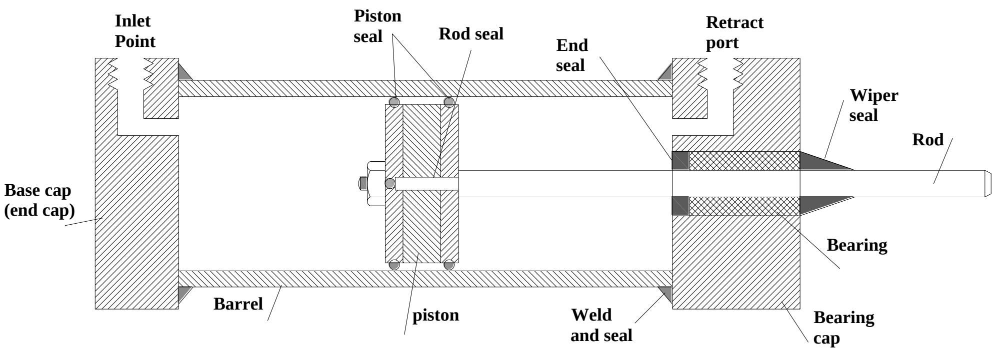


Figure 1.7 Construction features of double acting cylinder

The seven parts of the double acting cylinder are

1. Base cap with port connection
2. Bearing cap with port connection
3. Cylinder barrel
4. Piston
5. Piston rod
6. Scrapper rings
7. Seals

The **base cap** and **bearing cap** are made of cast material, aluminium or malleable cast iron. The two caps can be fastened to the cylinder barrel by tie rods, threads or flanges.

Cylinder barrel is usually made of seamless drawn steel tube to increase the life of the sealing components, the bearing surfaces of the cylinder are precision machined,. For special applications, the cylinder barrel can be made of aluminium, brass or steel tube with hard chromed bearing surface. These special designs are used where operation is infrequent or where there are corrosive influences.

The piston rod It is preferably made from heat treated steel. A certain percentage of chrome in the steel protects against rusting. Generally the threads are rolled to reduce the danger of fracture.

Piston seals are provided in between piston and barrel to avoid leakage. A **sealing ring** is fitted in the bearing cap to seal the piston rod. The bearing bush guides the piston rod and may be made of sintered bronze or plastic coated metal.

In front of this bearing bush is a scrapper ring.(wiper ring). It prevents dust and dirt particles from entering the cylinder space. Bellows are therefore not normally required.

The materials for the double cup packing sealing are

Perbunan, for -20 ℃ to + 80℃

Viton, for -20 ℃ to +190℃

Teflon for -80 ℃ to +200℃

O rings are normally used for static sealing.

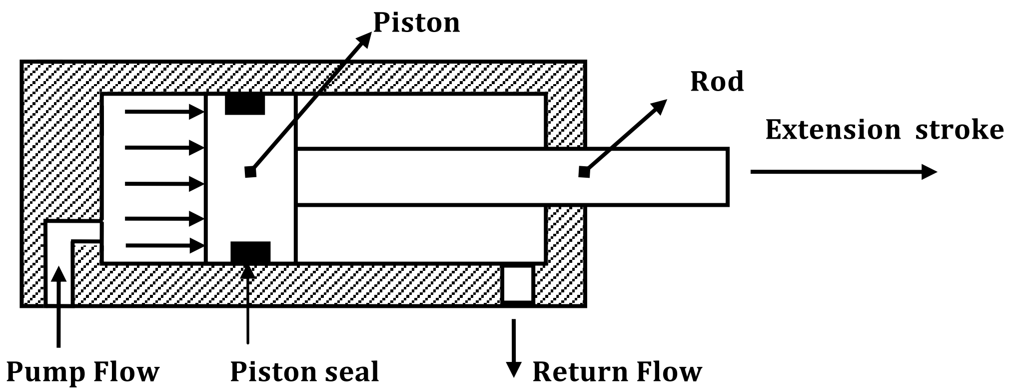
Construction of Double acting cylinder

There are two types of double acting cylinders.

i) Double acting cylinder with piston rod on one side.
ii) Double acting cylinder with piston rod on both sides

i) Double acting cylinder with piston rod on one side.

Figure 1.8 shows the operation of a double acting cylinder with piston rod on one side. To extend the cylinder, pump flow is sent to the blank end port as in Figure 1.8 (a). Fluid from the rod end port returns to the reservoir. To retract the cylinder, the pump flow is sent to the rod end port and fluid from the blank end port returns to the tank as in Figure 1.8 (b).



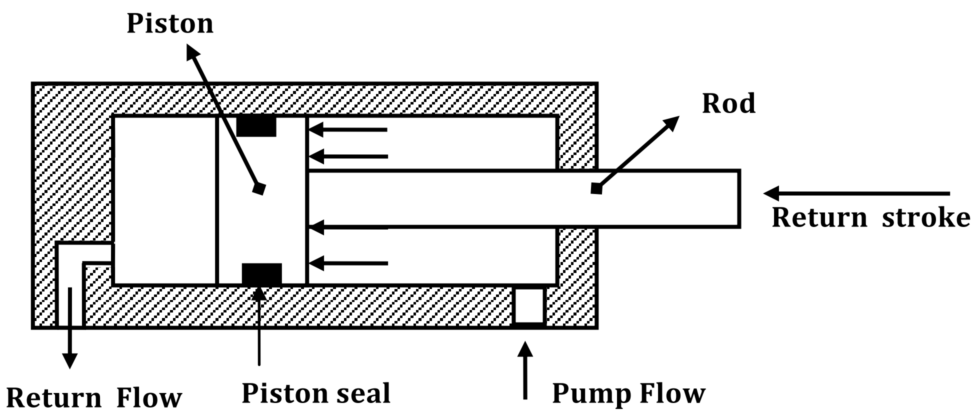


Figure 1.8 Double Acting Cylinder with piston rod on one side

ii) Double acting cylinder with piston rod on both sides

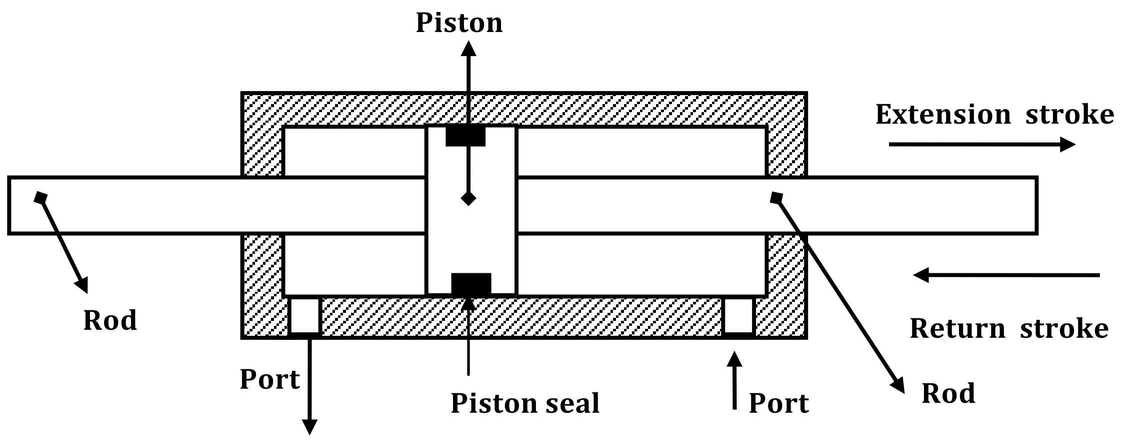


Figure 1.9 Double Acting Cylinder with piston rod on both side

A double acting cylinder with piston rod on both sides (Figure 1.9) is a cylinder with rod extending from both ends. This cylinder can be used in an application where work can be done by both ends of the cylinder, thereby making the cylinder more productive. Double rod cylinders can withstand higher side loads because they have an extra bearing one on each rod to withstand the loading. Double rod cylinders are used when there is bending load and accurate alignment and maximum strength is required. A further advantage is that rod is precisely located and may be used to guide the machine member coupled to it, dispensing with external guides or bearing in many cases, most standard production models are available either in single rod or double rod configuration A disadvantage of double rod configuration is that there is a reduction in maximum thrust due to the blanking effect of the rod cross section on the piston area and a slightly larger size of cylinder is required for a given duty. The thus will be the same on the ingoing stroke as that of a single rod double acting cylinder.

1.2.1.3 Based on the cylinder action

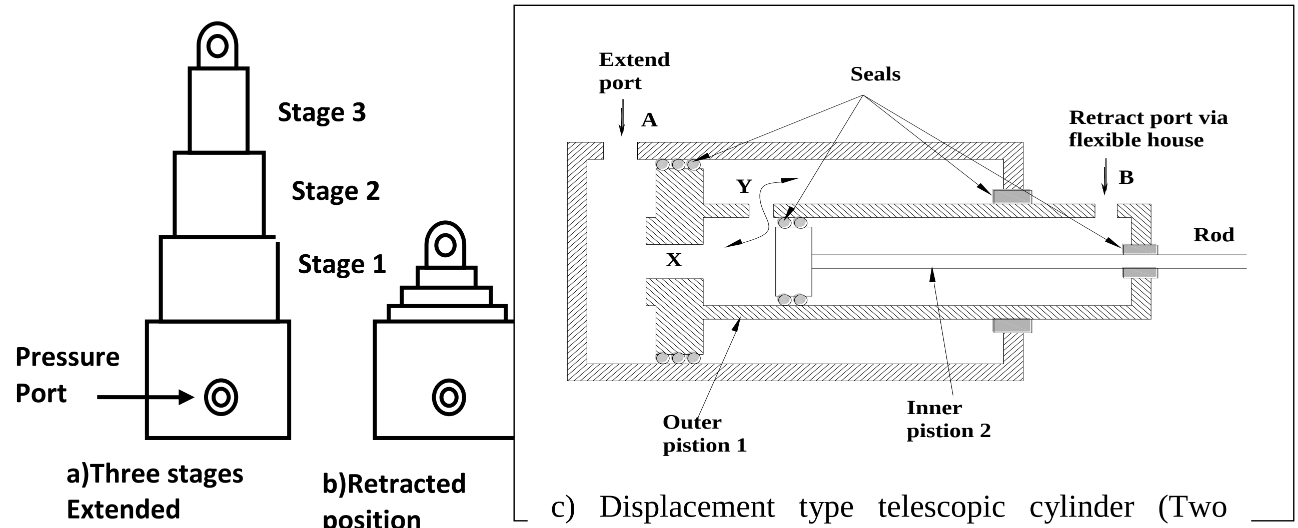
Rotating type of cylinders are used in applications where cylinder body is connected to a rotating member and air connection to the cylinder in a stationary housing. They are not widely used.

Non Rotating type of cylinders are widely used Industries. Cylinder body is connected air connection are mounted stationary housing and piston rod moves and exerts force.

1.2.1.4 Based on the cylinder’s design

In industry , differentiation is made between special design of regular cylinder and the special duty cylinders designed for a special purpose that are known by designation of their own. Special design cylinders are basically natural variations of single or double acting cylinders. Variations in special designs derived from standard production of cylinders and merely exchanging selected parts for others of different shapes or material. Special duty cylinders on the other hand are from the start designed to non-standard conditions of service or application. Following section deals with some of commonly used special design and special duty cylinders.

A) Telescopic Cylinder



a)Three stages Extended position

b)Retracted position

c) Displacement type telescopic cylinder (Two positions)

Figure 1.10 Double Acting Cylinder with piston rod on one side

Telescopic cylinder (shown in Figure 1.10 (a) and (b)) is used when long stroke length and short retracted length are required. They extend in stages, each stage consisting of a sleeve that fits inside of the previous stage. Figure 1.10 (c) shows the construction of a typical double acting telescopic cylinder with two pistons (Two stages).

Extension stroke: When the pressure is applied at port A, air flow through port X and Y and pressure is applied on both sides of Piston 1. But difference in areas causes the piston 1 to move to the right. Once the piston 1 fully extends, Inner Piston 2 will extend.

Retraction stroke: To retract, air is applied to port B. Air pressure will act on the annulus of the inner piston 2 and moves inner piston 2 to the left. When the inner piston moves to left and started to close port X, air from port B goes to annular side of the piston 1 via port Y and pushes the piston 1 to the left.

Figure shows the construction of a typical double acting telescopic cylinder with three pistons (three stages)

Forward stroke: when the pressurised air enters the port p, larger ram of diameter A moves first. Since the diameter of Ram A is relatively large, this ram produces large force at the beginning of the lift of the load. (usually in many application, initial inertial is high and larger force is required in the beginning, once initial inertia is overcome, smaller force is required to keep moving the weight). When ram A reaches the end of the stroke, ram B begins to move, providing smaller force. When ram B reaches its end position, Ram C will move outward to complete the stroke.

Retraction stroke: When the pressurised air enters the port T, then it acts on the annular area of Ram A and ram A is retrieved. Once the Ram A is retrieved, pressure continues to act on annular area of Ram B and retrieves Ram B. In similar way, the Ram C is also retrieved.

B) Tandem Cylinder

Schematic diagram of Tandem cylinder is shown in Figure 1.12. Tandem cylinders are two separate double acting air cylinders arranged in line to one cylinder body so that the power generated by the two is added together, thereby approximately doubling the piston output. A tandem cylinder is used in applications where a large amount of force is required from a small-diameter cylinder. Basically, a tandem cylinder is simply two or more separate cylinders stacked end to end in a unit and with all the pistons mounted on a common piton rod. Pressure is applied to both pistons, resulting in increased force because of the larger area. The drawback is that these cylinders must be longer than a standard cylinder of larger flow rate than a standard cylinder to achieve an equal speed because flow must go to both pistons

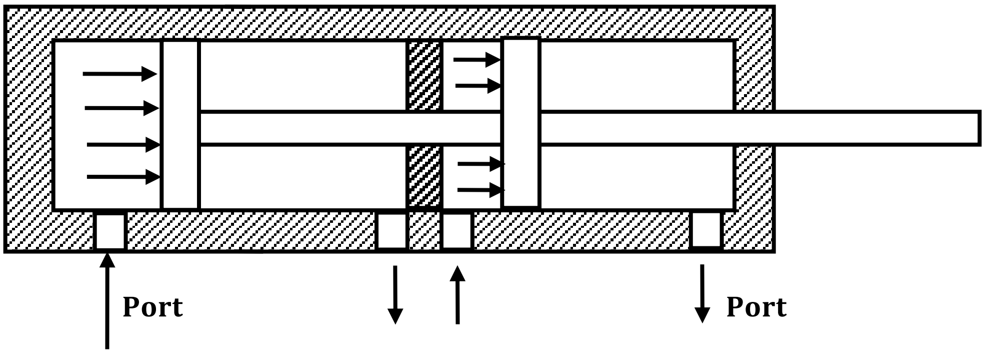


Figure 1.11 Double Acting Cylinder with piston rod on one side

Tandem cylinders are used where large output force is required with appreciable saving in bulk and weight. Tandem cylinders are employed where a small diameter of the assembly is required.

c) Rodless Cylinder

A rod less air cylinder differs from a basic air cylinder in that no piston rod extends outside the cylinder body. Instead, the internal piston is connected to an external carriage, by means of a magnetic or mechanical coupling system

There are three types of rod less cylinders, they are

i) Cable Cylinder
ii) Sealing band Cylinder with slotted cylinder barrel
iii) Cylinder with Magnetically Coupled Slide

i) Cable cylinder:

It is used for very long strokes, up to 2000mm. It consists of nylon jacketed cable which enters the cylinder barrel and is attached to one end of internal cylinder and exits through the gland seal and enters into the other end of the internal cylinder through the another gland seal as shown in the Figure1.12. When compressed air enters the cylinder the piston moves from end-to-end. The cables which are attached to either side of the piston and extend out the ends of the cylinder move as well. Depending on the direction of the piston both the carriage and any on-carriage tooling moves towards one end of the air cylinder or the other.

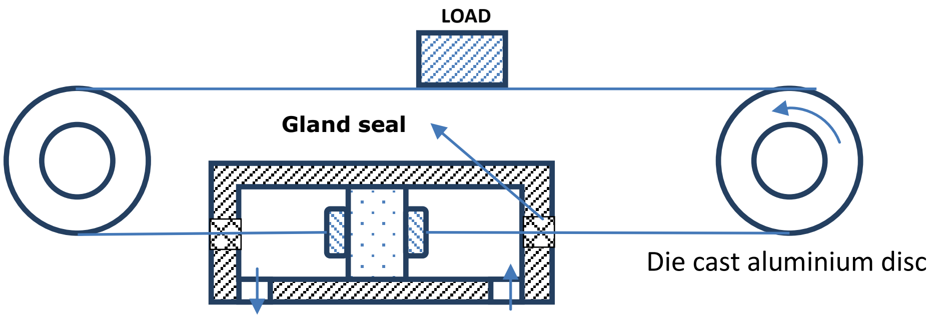


Figure 1.12 Cable cylinder

ii) Sealing band Cylinder with slotted cylinder barrel

Schematic diagram of sealing band cylinder with slotted cylinder barrel is shown in the Figure 1.13. Common components of a band cylinder are:

a. End caps
b. cylinder barrel
c. cylinder piston
d. carriage
e. mechanism for connecting carriage to piston
f. sealing bands / strips

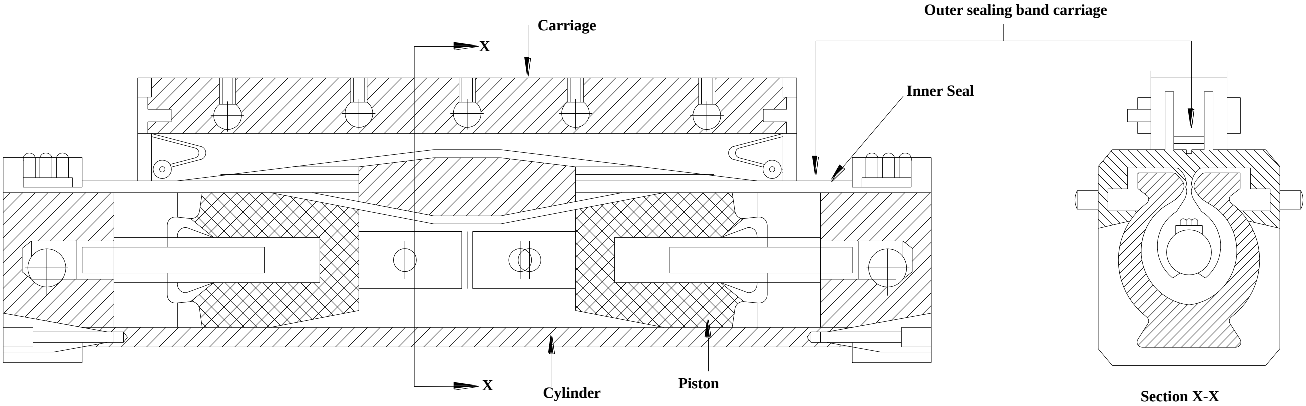


Figure 1.12 Sealing band cylinder with slotted cylinder barrel

As the carriage moves, the two band sealing strips are alternately opened in front and then closed behind of the moving carriage, regardless of the direction of travel. The seals on the pistons inside the cylinder barrel press the inner band seal tight against the barrel of cylinder, preventing air from leaking out.

The carriage on top of the band cylinder will have a wiper assembly at either end which will both remove any debris from the carriage path, and press the top seal tightly against the outside of the slot in the cylinder barrel, stopping any compressed air from escaping there.

iii) Cylinder with Magnetically Coupled Slide

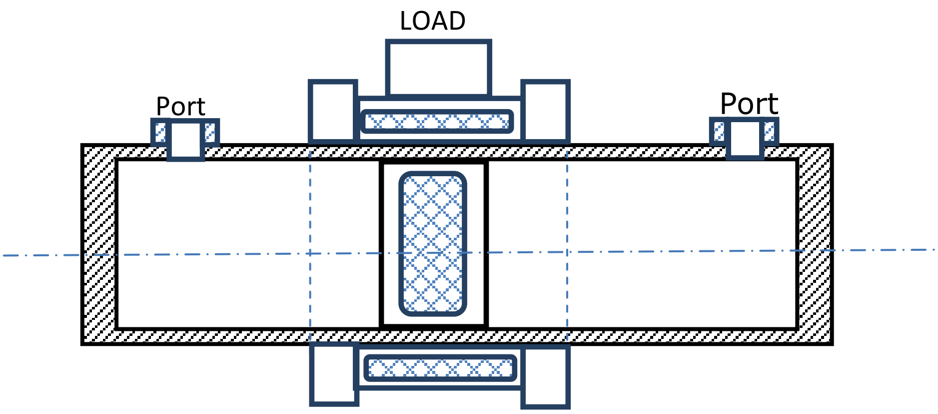


Figure 1.12 Schematic diagram of cylinder with Magnetically Coupled Slide

Figure 1.12 shows the schematic diagram of magnetic piston cylinder. Piston has powerful magnet which bonds the piston inside the cylinder with carriage outside which also contain powerful magnet. The magnetic cylinders are available upto the size of about 40 mm diameter and stroke lengths from 50mm to 4000 mm. They can operate at the speed of about 3000 mm/sec. The major advantages of this type of cylinders are

i) There is no leakage
ii) There is no direct contact of moving elements therefore the wear is less
iii) The orientation of the carriage can be changed easily,

D) Impact cylinders

Impact cylinders are used for high energy applications. Schematic diagram of impact cylinder is shown in Figure 1.13. These comprise, basically a normal pneumatic cylinder allied to a reservoir where by the speed of operation of the cylinder may be raised by some 15 times and the energy rating increases some 200 times without using larger valves or higher air supply pressures. Impact cylinders find particular application for high energy rate forming where, typically, an 200mm bore impact cylinder may accomplish a similar duty to that previously requiring a 50 Ton press.

If normal cylinders are used for forming operations, the thrust forces of the compressed air are limited. A cylinder producing high kinetic energy is the impact cylinder. In accordance with formula for kinetic energy (KE), the obvious means of obtaining high impact energy is by increasing the velocity.

$$\mathrm{KE} = \frac{1}{2} m v^2$$

$\mathrm{KE} = kinetic\ energy, Nm$

$m = mass\ in\ kg$

$v = speed\ in\ m/s$

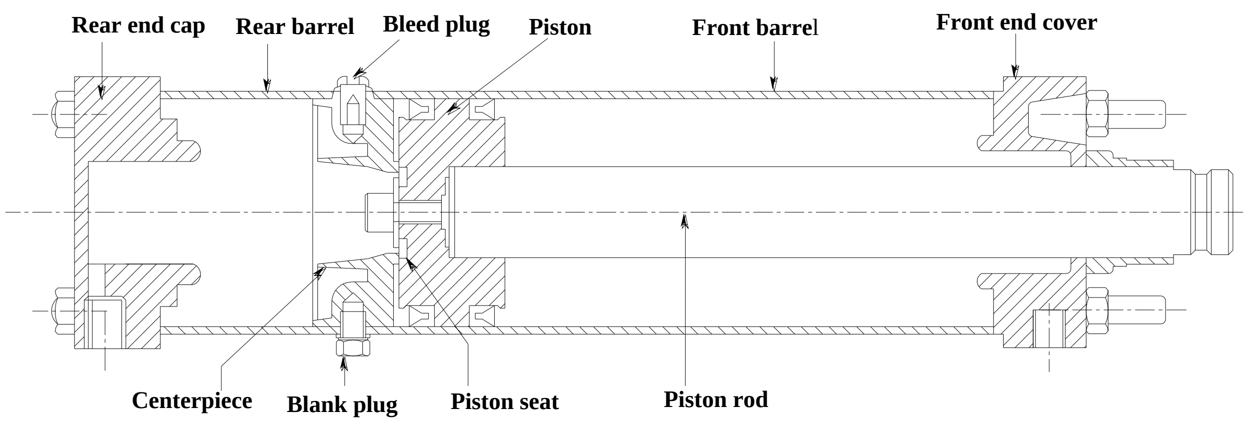


Figure 1.13 Schematic diagram of Impact cylinder

If the forming stroke is large, the velocity diminishes rapidly and hence also the impact energy, This type of cylinder is therefore not suitable for large forming strokes. They are used for stamping, punching, cutting, riveting, beading, embossing etc.

Sequence of operation is shown in the Figure 1.14. To the top end of conventional cylinder is added a reservoir opening into the cylinder proper via a poppet valve (centre piece) controlled by the piston movement. Ratio is the piston area to poppet valve (centre piece) area is of the order of 9:1 so that pressure in the reservoir is balanced by a pressure of approximately 1/9th in the exhaust end of the cylinder.

To charge the cylinder the reservoir is connected to the line with the pressure building up to the line pressure. Movement of the piston is opposed by exhaust pressure until this falls to 1/9th of the line pressure. Forces acting on the top and bottom of piston are then in balance and the piston starts to move away from the poppet seat (centre piece). Immediately the poppet valve is open and the reservoir pressure is applied over the entire whole of the piston area resulting in very rapid acceleration of the piston up to the maximum speed. The energy released by the air suddenly allowed to expand from the reservoir is thus converted into kinetic energy by the piston assembly which delivers an impact blow.

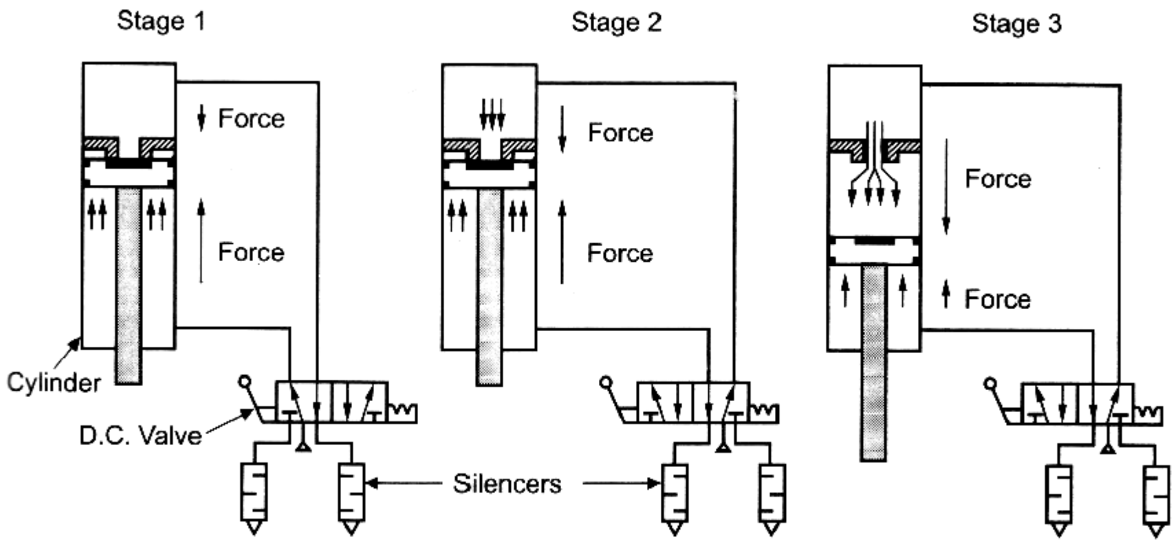


Figure 1.14 Sequence of operation of Impact cylinder

E) Duplex cylinder or opposite thrust or multi position cylinder

A special duty tandem arrangement is also used in duplex cylinder, although in this case the (two) pistons are mounted on separate connecting rods, one operating the other. Each piston is thus capable of independent motion via its own separate air supply control Thus a duplex cylinder may be used for tow motion duties- example clamping a work piece in position y motion of one piston, followed by a shearing, punching, pressing or similar operation powered by the other piston.

Opposed thrust or multi position cylinders are similarly a combination assembly of at least two double acting cylinders in one with the pistons and rods in opposed arrangement. This results in a four position cylinders as shown in Figure 1.15 (b) The characteristic of such opposed thrust cylinders is the availability of more than two definite fixed operating positions. Theoretically it is possible for a multiple number of pistons and rods to be combined into the one cylinder assembly so as to obtain an actuator with six or eight position. Three position cylinders as illustrated in Figure 1.15(a) can be obtained as standardized units. Designs with up to 12 operating positions are possible.

Application

- Filling shelves from a conveyor
- Lever actuation
- Sorting device (accept – reject- rework)

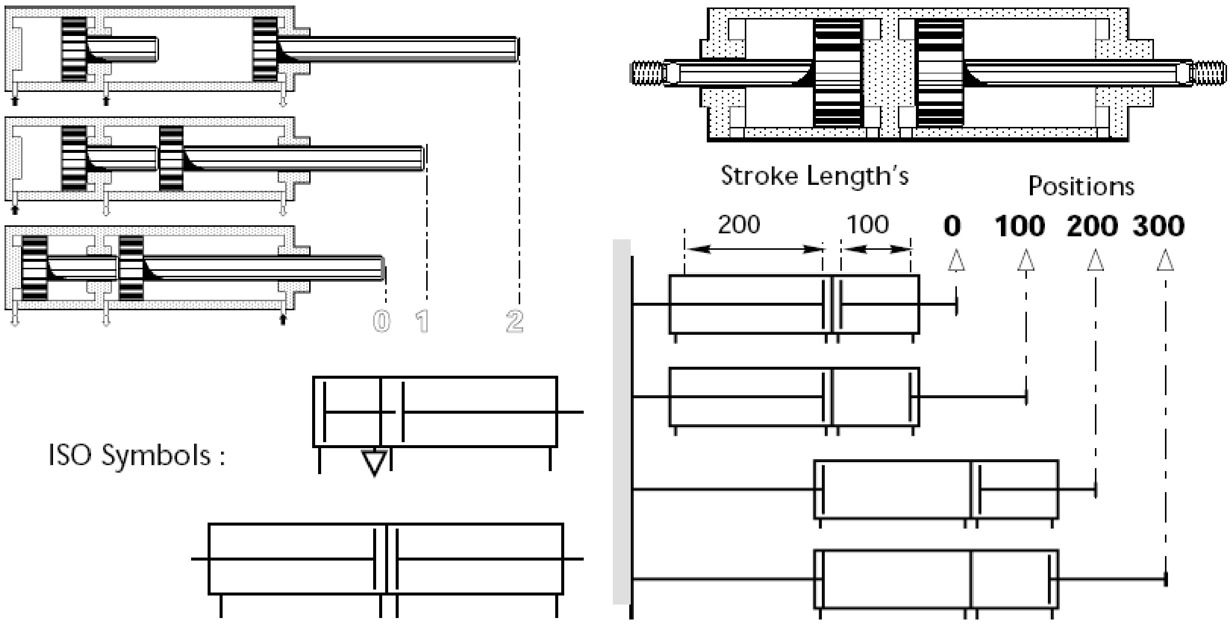


Figure 1.15 Duplex cylinder (a) three position b) four position

F) Position sensor for cylinders

In automation application, it is often required for a signal to be generated when the cylinder piston reaches a particular position along its stroke, so that the control system can initiate the next phase of the operation. This position may be at either end of the stroke or some point intermediate between the ends. There are several ways in which this can be done. Schematic diagram of cylinders with position sensor is shown in Figure 1.16

- The piston rod trips a micro-switch or pneumatic valve
- Pressure threshold sensors respond to a drop in exhaust pressure when the piston stops moving
- Magnetic sensors mounted directly on the cylinder barrel sense the magnetic field created by a permanent magnet incorporated in the piston and trigger reed switch.
- Hall effect sensors triggered by a magnetic piston
- Pneumatic reed valves. Triggered by a magnetic piston
- Miscellaneous such as photoelectric, inductive and capacitive detectors

Magnetic trip switches are the traditional method for position sensing and are still widely used and reliable. Pressure threshold sensor respond less to actual position than to piston velocity, so they are only suitable for detecting the end of the piston travel, they have work unreliably with slow moving pistons. Both of these are gradually being superseded by the two types of magnetic sensors.

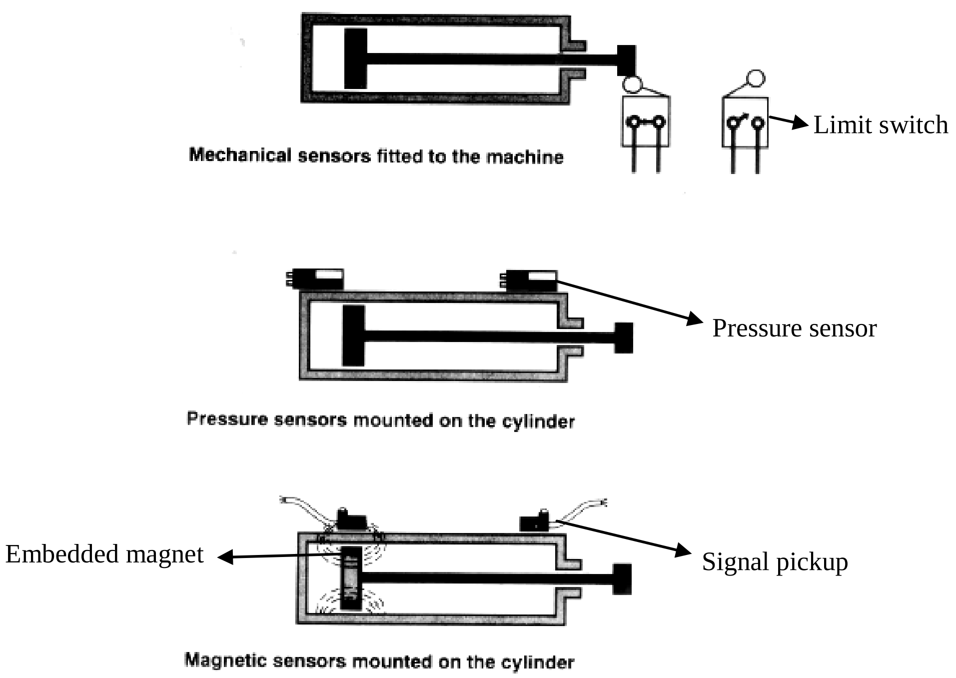


Figure 1.16 Cylinder with sensors (a) Mechanical b) pressure c) magnetic

The other type of sensors which are used are electronic sensor with magnetic detection. They are directly mounted on a magnetic cylinder tube. A permanent magnet is embedded in the piston. This creates a magnetic field. When the piston moves, the magnet actuates the electronic system of the sensor and provides the desired signal. These electronic position sensors work at 10-24 volts. The maximum current is around 150mA. The leakage current is around 10 mA at 24 Volts and internal voltage drop is less than 0.5 Volts for 100 mA. They are capable of working between -10℃ to +60 ℃

1.3 Standard metric cylinders

BS: 5785 1980 gives tables (Table 1.2) of preferred sizes for the cylinder bore and rod diameter of metric cylinders. Most cylinder manufacturers have based their standard range of metric cylinders on these recommendations, offering two rod sizes for each cylinder bore.

A number of combinations have a piston rod to piston diameter ratio in the region of 0.7, which gives an annulus area of approximately one-half the full bore area. This area ratio is of use in regenerative circuits to give similar values of speed and thrust on both the extension and retraction strokes.

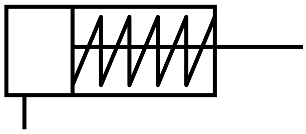
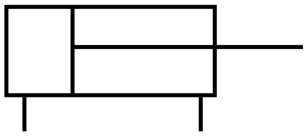
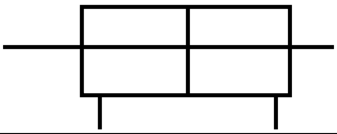
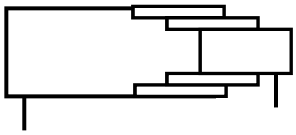
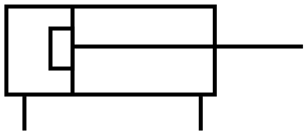
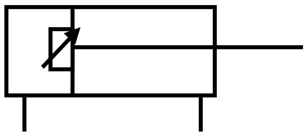
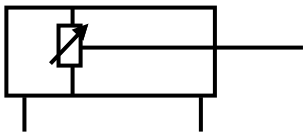
1.3.1 Graphical Symbols for cylinders.

There are several symbol systems and conventions in use around the world for pneumatic cylinders. Most official recognized by standard bodies, commonly used is ISO 1219-1. Table 1.3 gives the symbols for most commonly cylinders in industry.

Table: 1.2 Standard cylinder sizes

Piston Diameter(mm)		40	50	63	80	100	125	140	160	180	200	220	250	280	320
Piston Rod	small	20	28	36	45	56	70	90	100	110	125	140	160	180	200
Diameter(mm)	large	28	36	45	56	70	90	100	110	125	140	160	180	200	220

Table 1.3 Graphical symbols of cylinders

Sl No	Graphical Symbols	Explanation
1		**Single acting cylinder with unspecified return:** Air pushes the piston in one direction and the piston is return is unspecified. External dock or lever may push
2		**Single acting cylinder with spring return.** Air pushes the piston in one direction and piston returns by spring on rod side
3		**Double acting cylinder –single piston rod:** the force exerted by compressed air moves the piston in both direction.
4.		**Double acting cylinder –double piston rod**It has piston rods extending from both ends of the cylinder. It produces equal force and speed on both sides of the cylinder
1.		**Telescopic cylinder –double acting** is used where space is constraint. It is used for long stroke application like in pneumatic cranes, dump trucks, lift fork trucks, dipper wagon
7.		**Double acting cylinder –** fixed cushion on one side, Cushioning is used in the end position to prevent sudden impact which otherwise may damage parts**.**
8		**Double acting cylinder – variable cushion on one side–** fixed cushion on one side, cushioning is variable in one direction by adjusting the orifice opening.
9		**Double acting cylinder – variable cushion on both sides–** fixed cushion on one side, cushioning is variablein both direction.

5.4 Cylinder mountings.

The way in which the cylinder is mounted influences service life, maintenance frequency and success of the entire installation. Poor mounting design can cause excessive side loads and stresses which will bring about early failure of some vital component. There are three main categories of cylinder mounting. The selection of these mountings depends on the application and machine configuration.

1. Fixed Centreline mountings
2. Pivoted centreline mountings
3. Fixed non centreline mountings

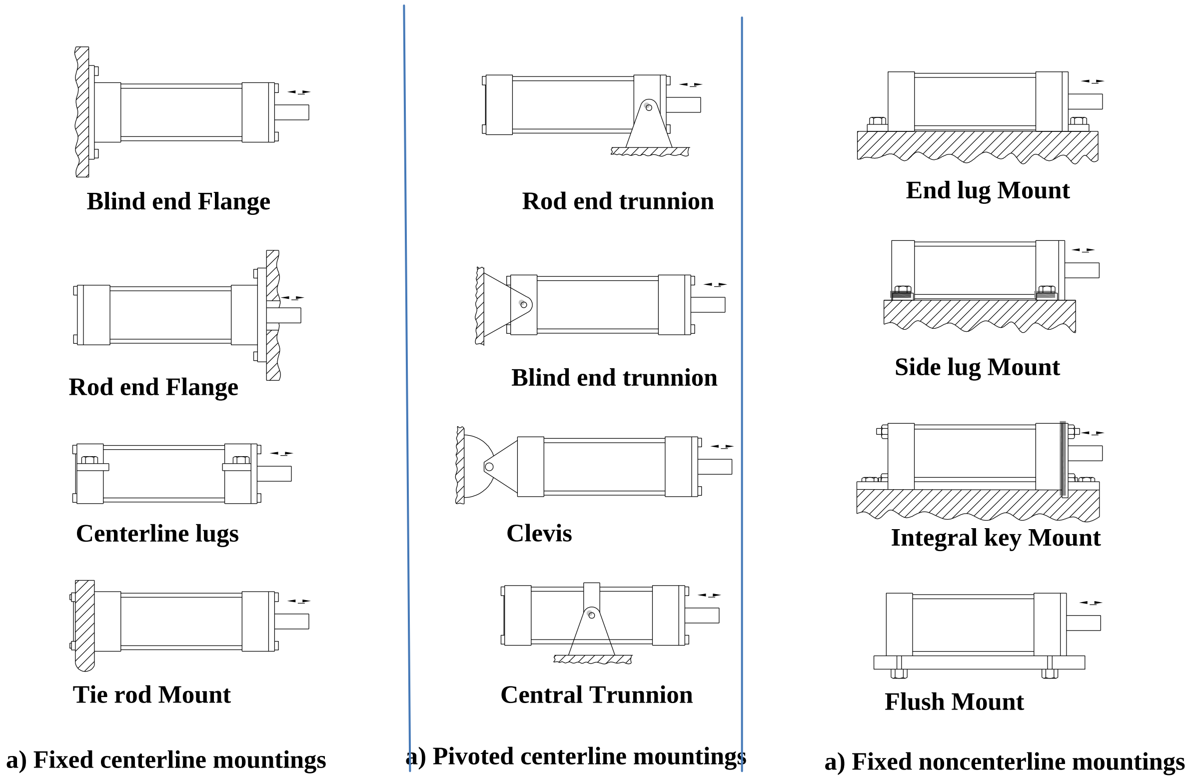


Figure 1.17 Types of mounting a) Fixed centreline b) Pivoted centreline c) Fixed noncenterline

Figure 1.17 shows all three types of mountings.

Fixed Centreline mountings: In this mounting, the cylinder is supported along its centre line. The mounting bolts are thus subjected to shear or simple stress. This mounting needs accurate alignment. Misalignment is not tolerable.

Pivot centreline mounting: Many applications need rotational degree of freedom for a cylinder as it reciprocates. The pivot mounting can be clevis type or trunnion type. This mounting permits rotational freedom in one plane. If universal joint is used, greater degrees of freedom are possible.

Fixed non centreline mounting: This mounting of cylinder introduces torque under loaded condition. The cylinder may rotate or bend about its mounting bolt when loaded. The stress level on the cylinder is higher as compared to the centre line mounting.

Following points should be considered while mounting the cylinder

1. Cylinders with centreline mountings tend to lean under load.
2. Cylinder with non-centreline mounting generally require strong machine frames to resist bending moments

3. If the motion of the machine part acted on by the cylinder rod movement is essentially linear, a fixed mounted cylinder should be used.
4. If the cylinder has a long stroke, a pivoted mounted cylinder may be required to prevent the piston rod buckling. Where long stroke and fixed mounting are necessary , support is needed to prevent vibration and excessive sag
5. The mounting selection depends on the resulting force (compression or tension) in the cylinder rod. The blind end or cap flange mounting is best for compressive loads. The rod end or head flange mounting is best where the rod is in tension
6. Alignment problem are always critical. If misalignment can occur between the cylinder and machine part it moves, it is necessary to compensate for this in the selection of cylinder mounting. For example, a simple centreline pivot mounting will compensate for misalignment if it occurs in only one plane. Where misalignment can occur in more than one plane, the cylinder must be fitted with a universal (ball and socket) pivot joint. It is important that both ends of the pivot mounted cylinder should be supplied with flexible connections.

1.4.1 Piston rod buckling

The piston rod in a pneumatic cylinder will act as a strut when it is subjected to a compressive load or it exerts a thrust. Therefore the rod must be of sufficient diameter to prevent buckling. Euler's strut theory is used to calculate a suitable piston rod diameter to withstand buckling. Euler's formula states that

$$k = \frac{\pi^2 EI}{L^2}$$

Where K = buckling load (kg),

E = modulus of elasticity (kg/cm^2) (2.1 ×10^6 kg/$cm^2$for steel),

I = second moment of area of the piston rod (cm^4) ($\pi d^2/64$ for a solid rod of diameter d cm), an L = free (equivalent) buckling length (cm) depending on the method of fixing the cylinder and piston rod. The values of L for various configuration is shown in Figure 1.19

The maximum safe working thrust or load F on the piston rod is given by

F = k /S,

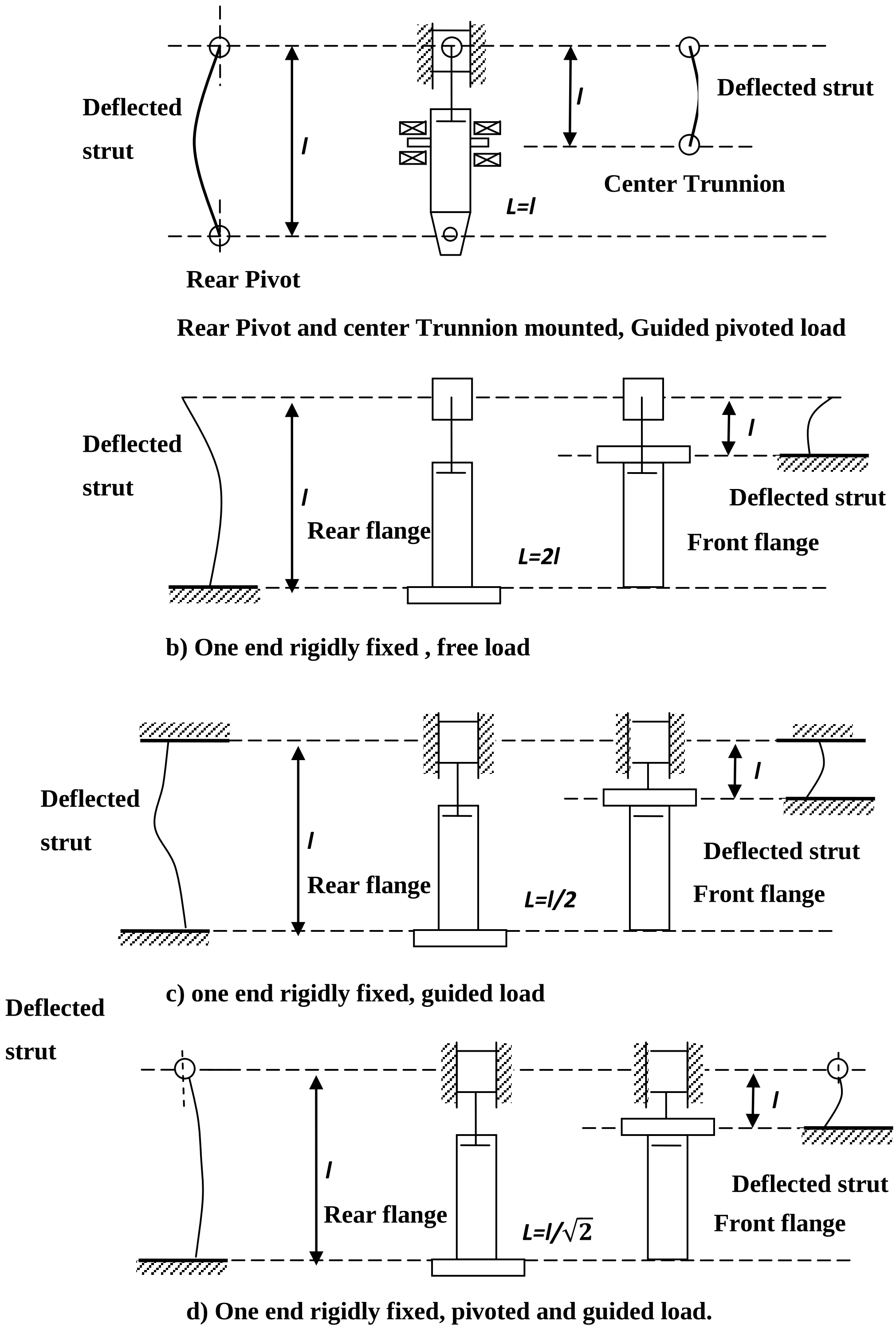


Figure 1.19 Relationship between piston rod, free buckling length and method of fixing

Where S is a factor of safety which is usually taken as 3.1. The free or equivalent buckling length L depends on the method of fixing the piston rod end and the cylinder, and on the maximum distance between the fixing points, i.e. the cylinder fully extended. In cases where the cylinder is rigidly fixed or pivoted at both ends there is a possibility of excessive side loading occurring. The effect of side loading can be reduced by using a stop tube inside the cylinder body to increase the minimum distance between the nose and the piston bearings. Schematic diagram is shown in Figure 1.20. The longer the stop tube, the lower will be the reaction force on the piston owing to the given value of side load. Obviously the stop tube reduces the effective cylinder stroke

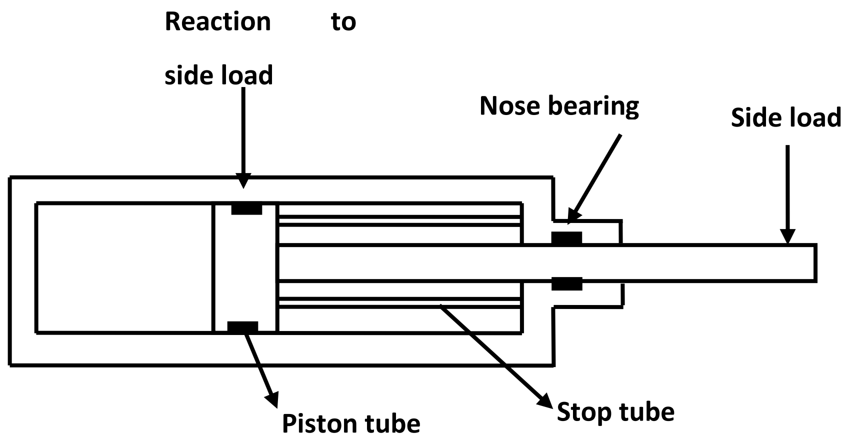


Figure 1.20 Use of stop tube to minimize side loading.

1.5 Cylinder seals

Seals are used in cylinders to prevent the losses caused by leakage and to make the effective use of the compressed air energy medium. Important characteristics needed for seals are

1. Long life
2. Low friction
3. Resistance to heat
4. Stability of form
5. Higher range of working pressure
6. Higher Range of temperature
7. Mechanical strength

1.1.1 Classification

1. Static seals

2. Dynamic seals

Static seals are used to provide a sealing between the stationary parts of a cylinder. For example, end cap and barrel. O ring is most commonly used static seal. O rings are used for small cylinders.

Dynamic seals are used provided for surfaces which are moving. Dynamic seals are used provided for surfaces which are moving. Cup seals and Z seals are commonly used dynamics seals.

There are a variety of seals required within a pneumatic cylinder. Single acting cylinders use less number of seals. Double acting seals use at least five different types of seals as shown in Figure 1.21

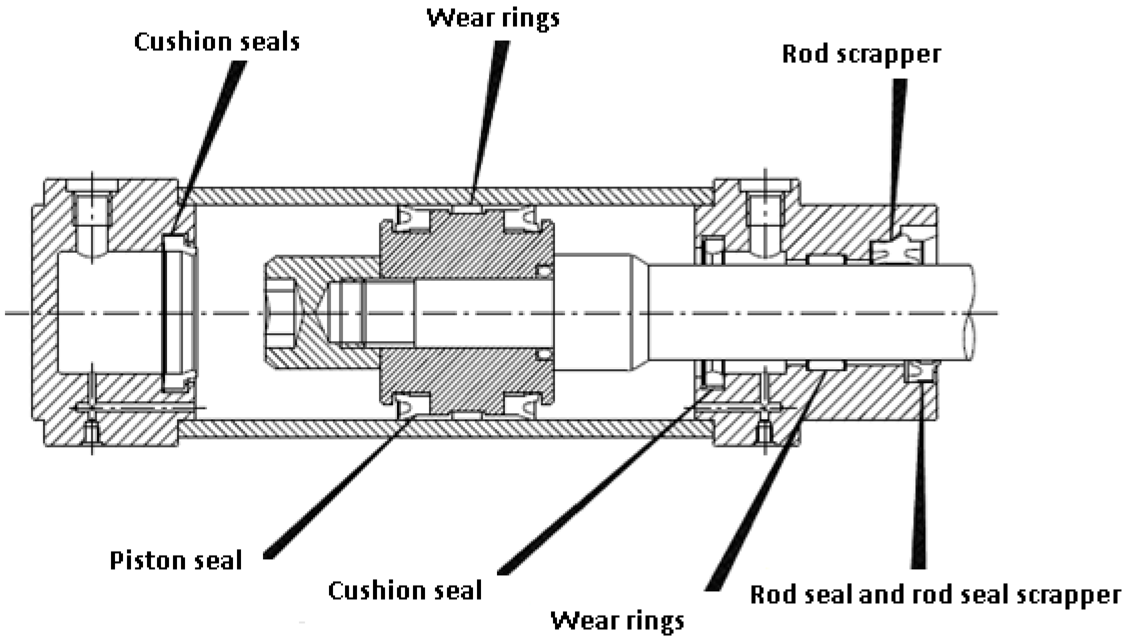


Figure 1.21 Different types of seals used in pneumatic cylinders

1. Cushion seals
2. Wear ring
3. Piston seals
4. Barrel seals
5. Piston rod/wiper seal

Cushion seals: Cushioning protects a cylinder and its load by absorbing energy at the end of the stroke. When the piston moves to the left of pneumatic cylinder, the air on the left side of the piston in the left chamber must be exhausted to allow full travel of the piston and rod. In a cushioned cylinder, this air cannot escape from the port by virtue of the cushion seals which seals against the piston rod. The only escape path is through the cushion orifice, which is normally a small hole. When the piston reaches the cushion seal, the piston travel is slow down due to cushioning of the air. Thus the cushion seals perform a dual role of a seal and non-return valve.

Wear ring is a open band fixed around the piston. It is made from a hard plastic material normally Teflon compounded with polyphenylene sulphide or good quality bearing bronze to provide best wear resistance and excellent bearing support. Wear ring also guards the rod against scoring by piston.

Piston seal: An O-ring piston seal is used for small bore sizes. For medium and large bore double acting cylinders cup seals are used. They are cheap and easy to fit but may be easily damaged by dirt.

Barrel seal: O-ring piston seal is used as barrel seal. It is used in the near the end caps and it is tight fit in the grooves location

Piston rod seal/ Wiper seal.

A piston rod seals, also known as wiper seal, is used for harsh environment. This seals acts as pressure seal and wiper seal. The external body of this seal is a pressure-tight fit within the bearing housing. Cleaning action of the seal removes abrasive particles that can settle on the rod during the outward stroke of the cylinder or due to some other operations

5.6 End cushioning in pneumatic cylinders

The inherent capability of all air cylinders to perform rapid reciprocating motion also supplies a disadvantage in that considerable kinetic energy is dissipated in the form of shock at the completion of the stoke. This can set up severe stresses in cylinders itself or end covers, or on frame work of the machines to which it is attached. This can be overcome by providing cushioning to produce gradual deceleration of the piston as it approaches the end of its stroke. Schematic diagram is shown in **Figure 1.22**

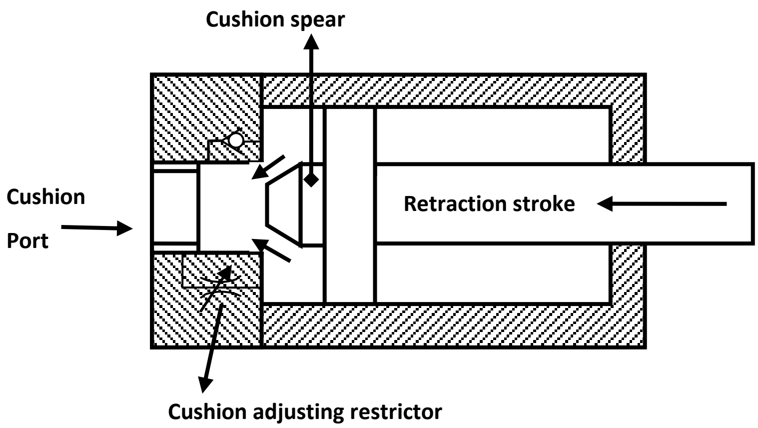


Figure 1.22 Operation of cylinder cushions.

Need for cushioning when the pressurised air is supplied at the inlet port of the cylinder, piston accelerates and travels inside the cylinder. Piston has to be slowed down at the end of the stroke to prevent excessive impact on the end caps. These shock loads arise not only from the fluid pressure, but also from the kinetic energy of the moving parts of the cylinder and load. These end travel shock loads can be reduced by decelerating the piston at the end of stroke by cushioning.

For the prevention of shock due to stopping loads at the end of the piston stroke, cushion devices are used. Cushions may be applied at either end or both ends. They operate on the principle that as the cylinder piston approaches at the end of stroke exhaust fluid is forced to go through an adjustable needle valve which is set to control the escaping fluid at the given rate. This allows the deceleration characteristics to be adjusted for different loads. When the cylinder piston is actuated, fluid enters the cylinder port and flows through the little check valve so that entire piston area can be utilized to produce force and motion.

There are two types of cushioning is possible in pneumatic cylinder

1. Fixed cushioning
2. Adjustable cushioning

Fixed cushioning: This type of cushioning is used for small bore cylinder. These cylinders make use of synthetic rubber buffers to give a simple fixed cushion effect. These shock absorbent disks placed into the end-covers suction the impact of the piston. Schematic diagram is shown in Figure 1.23

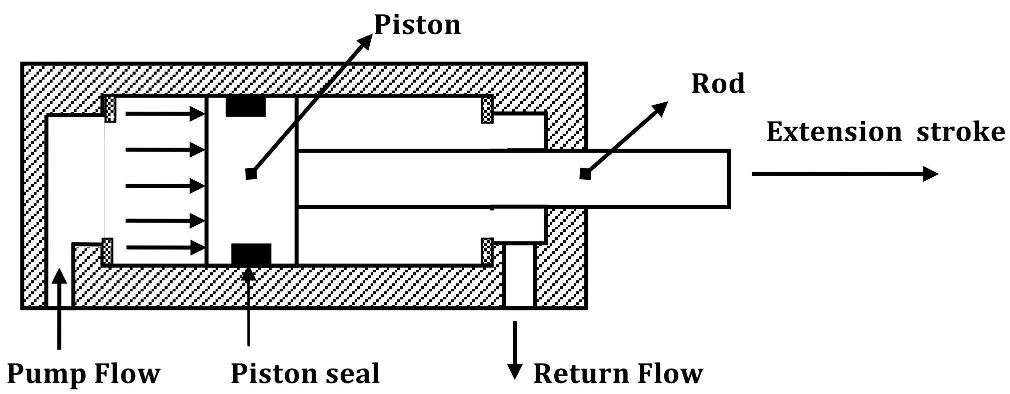


Figure 1.23 Operation of cylinder cushions

Adjustable cushioning: As a rule, cushions are applied to cylinders whose piston speed exceeds 0.1 m/s. Cushion can be applied at one end or both ends of the cylinder. Adjustable cushioning is possible with two methods

1. Using the cushion seal
2. Using the piston with the plunger

Using the cushion seal: Cushioning using cushion seal was explained earlier in cylinder seal section 1.1.1.In a cushioned cylinder, this air cannot escape from the port by virtue of the cushion seals which seals against the piston rod. The only escape path is through the cushion orifice, which is normally a small hole. When the piston reaches the cushion seal, the piston travel is slow down due to cushioning of the air

Using the piston with plunger and cap: In this type exhaust air is unrestricted until the plunger enters the cap. The exhaust flow route is through deceleration valve which reduces the end of travel speed. The needle valve is adjustable to allow the deceleration rate. Schematic diagram of the construction of cylinder cushion is shown in Figure 1.24

Construction and working:

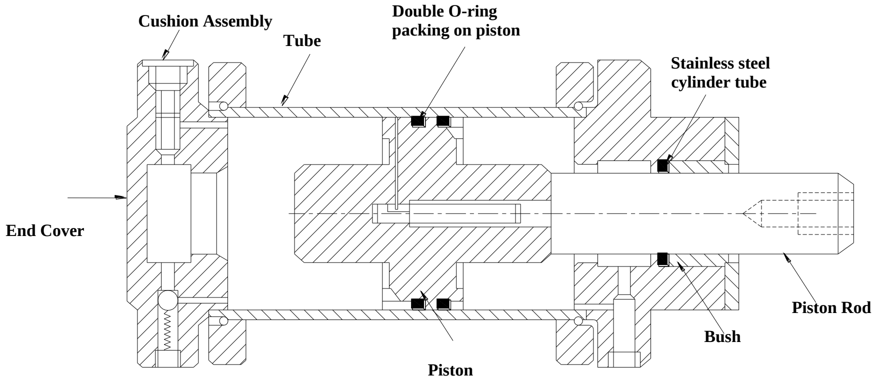


Figure 1.24 construction of cylinder cushions

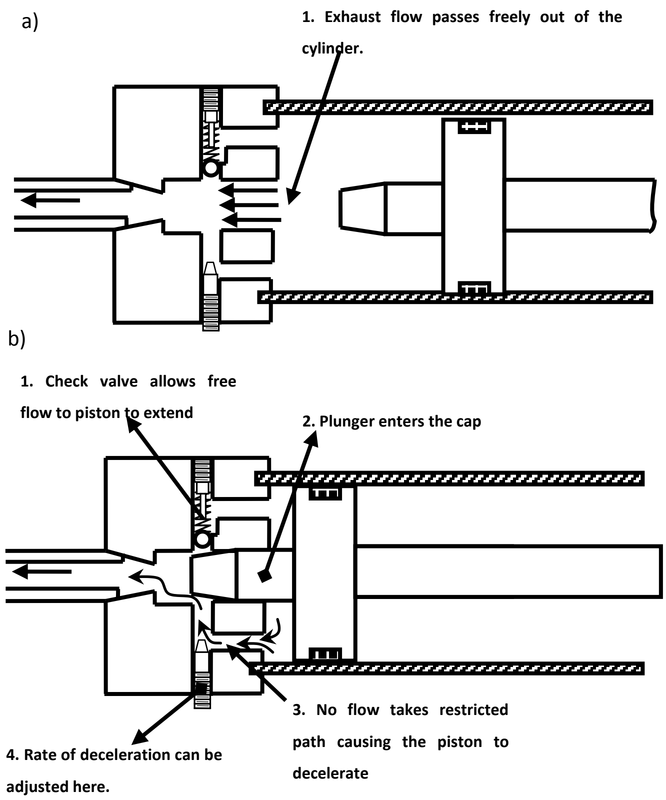


Figure 1.25 Operation of cylinder cushions a) normal position b) end position

A typical cylinder cushioning operation arrangement is shown in the figure 1.25(a) and 1.25(b)

As the piston approaches the end of its stroke, the plunger enters the end cap port and thus blocks the free flow. Now the fluid is trapped between the piston and end cap. This fluid can escape only by passing through the adjustable restrictor, as shown in the Figure 1.25(b). This fluid flow through the restricted flow path causes the piston to decelerate. The rate of deceleration of the piston can be controlled by adjustable needle valve. A non-return valve or check valve is provided to allow free flow of fluid to the cylinder quickly during the return stroke.

1.7 CYLINDER FORCE, VELOCITY AND POWER

The output force (F) and piston velocity (V) of double acting cylinders are not the same for extension and retraction strokes.

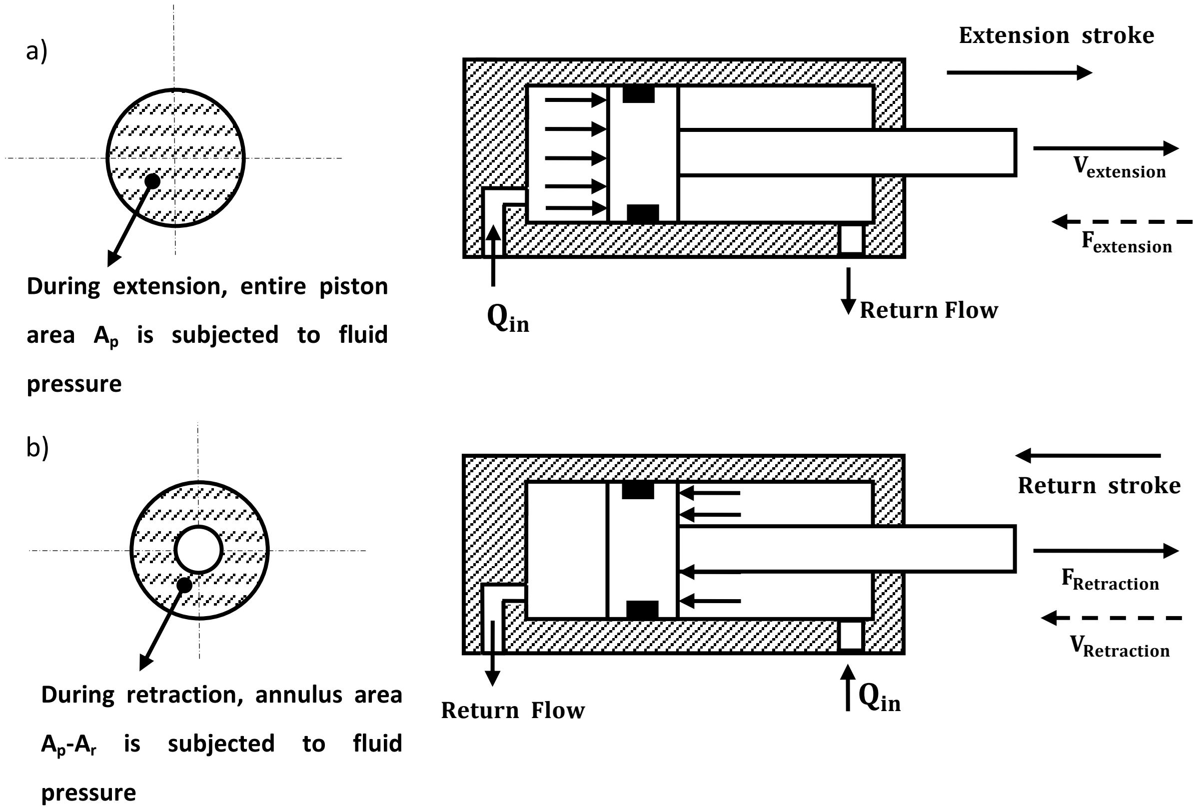


Figure 1.26 Effective areas during extension and retraction strokes

During the extension stroke shown in Figure 1.26(a), the fluid pressure acts on the entire circular piston area Ap. During the retraction stroke, the fluid enters the rod end side and the fluid pressure acts on the smaller annular area between rod and cylinder bore (A_p- A_r) as shown by the shaded area in Figure 1.26 (b). A_r is the area of the piston rod. Due to the difference in the cross sectional area, the velocity of piston changes. Since A_p is greater than (A_p-A_r), the retraction velocity (V_{ret}) is greater than the extension velocity (V_e) for the same flow rate.

During the extension stroke, fluid pressure acts on the entire piston area (A_r), while during retraction stroke, the fluid pressure acts on the annular area (A_p-A_r). This difference in area accounts for difference in output forces during extension and retraction strokes. Since (A_r) is greater than (A_p-A_r), the extension force is greater than the retraction force for the same operating pressure.

Force and Velocity during Extension stroke

$v_{ext} = \frac{Q_{in}}{A_p}$ ----------------------(1)

$F_{ext\,=}\, p \times A_p$ -------------------------(2)

Force and Velocity during Retraction stroke

$v_{ext} = \frac{Q_{in}}{A_p - A_r}$---------------------(3)

$F_{ext\,=}\, p \times (A_p - A_r)$ -----------------------------(4)

Power developed by a pneumatic cylinder (both in extension and retraction)

$Power = Force\ \times velocity = F \times V$ -----------(5)

In metric units, the kW power developed for either extension or retraction stroke is

$Power\ (kW) = v_p \left(\frac{m}{s}\right) \times F(kN) = Q_{in}(\frac{m^3}{s}) \times p(kPa)$ --------(6)

Power during extension

$P_{ext\,=}\, F_{ext} \times V_{ext} = p \times A_p x \frac{Q_{in}}{A_p} = p \times Q_{in} ------(7)$

Power during retraction

$P_{ret\,=}\, F_{ret} \times V_{ret} = p \times (A_p - A_r) \times \frac{Q_{in}}{A_p - A_r} = p \times Q_{in} --(8)$

Comparing equations (A) and (B), we can conclude that the powers during extension and retraction strokes are same.

1.8 ACCELERATION AND DECELERATION OF CYLINDER LOADS

Acceleration

To calculate the acceleration of cylinder loads, the equations of motion must be understood.

Let

u = initial velocity

v = velocity after a time *t*

s = distance moved during time *t*

a = acceleration during time *t*.

The standard equations of motion are:

$v = u + at$ --------------------(9)

$v^2 = u^2 + 2as$ -----------------(10)

$s = ut + 1/2\, at^2$-----------------(11)

and

$s = ½\,(u + v)\,t$

The force F to accelerate a weight W horizontally with an acceleration a is given by

Force = mass × acceleration

F = (W/g) a

Where g is the acceleration due to gravity and is $9.81\ {}^{m}/_{s^2}$. The force P required to overcome friction is given by P = μW, where μ is the coefficient of friction.

Note: Dynamic cylinder thrust

In dynamic applications the load inertia, seal friction, load friction, etc. must be allowed for in calculating the dynamic thrust.

As a first approximation, the dynamic thrust can be taken as 0.9 times the static thrust. (It must be realized that this is only an approximation and can be considerably in error, dependent on load conditions and associated circuitry.)

Cylinder seal friction varies with seal and cylinder design. The pressure required to overcome seal friction is not readily available from the majority of cylinder manufacturers. The seal friction breakout pressure can be taken as 5 bar for calculation purposes. It will reduce when the piston starts to move. The pressure required to overcome seal friction will reduce as the cylinder bore size increases and will vary according to seal design.

Example 1: A pneumatic cylinder is required to move a 1000 N load 150mm in 0.5s. What is the output power?

Solution

Velocity,

$V = {}^{d}/_{t} = {}^{0.15}/_{0.5}$=0.3m/s

Power,

$P = F \times V = 10^3 \times 0.3$=300W = 0.3 kW

Example 2: A pneumatic cylinder is required to extend at a minimum speed of 0.75 m/s in a system with a flow rate of 50 LPM. What cylinder size is required?

Solution

$$Q = 50lpm = \frac{50}{1000} {}^{m^3}/_{min} = \frac{50}{1000\times60} = 8.33 \times 10^{-4}\, {}^{m^3}/_{s}$$

$$Q = A_p \times v$$

$$8.33 \times 10^{-4} = \frac{\pi}{4} d_p^2 \times 0.75$$

Solving we get $d_p = 37.6\, mm$

Example 3: An 8 cm diameter pneumatic cylinder has a 4 cm diameter rod. If the cylinder receives flow at 100 LPM and 6 bar, find the (a) Extension and retraction speeds (b) Extension and retraction load carrying capacities.

Solution

Qin=100 LPM=100/(1000×60)=1/600 (m^3/s)

d_p= diameter of cylinder= 8 cm = 8 x 10^{-2} m

d_r = diameter of piston rod= 4cm = 4x10^{-2} m

p = 6 bar =6x10^5 N/rn^2 or Pa

(a) $V_{ext}=\frac{Q_{in}}{A_p}=\frac{1/600}{\pi d_p^2/4}$=0.3315m/s

(b) $V_{ret}=\frac{Q_{in}}{(A_p-A_r)}=\frac{1/600}{\pi(d_p^2-d_r^2)/4}=0.442m/s$

(c) $F_{ext}=p\times a_p=6\times10^5\frac{\pi(8\times10^{-2})^2}{4}=2513.3$ N

(d) $F_{ret}=p\times(A_p-A_r)=\frac{5\times10^5\times\pi\left[(8\times10^{-2})^2-(4\times10^{-2})^2\right]}{4}=1885\text{N}=1.89\text{kN}$

1.11 Rotary Actuators

Rotary Actuators are used to achieve angular motion. Rotary actuators are devices which produce high torque output and have a limited rotary movement. Standard rotations are 90°, 180°, and 270°. Rotary actuators are mainly available in three designs.

i) Vane type limited rotation motors

- Single vane rotation motor
- Double vane rotation motor

ii) Rotary Actuator of Rack and Pinion Type

iii) Helix spine rotary actuator

1.11.1 Vane type actuators

Where the torque and motion is all produced in a rotary sense, the construction limits the rotation to less than one rotation.

Piston type actuators are essentially linear actuators mechanically connected to translate the linear force to produce an output torque and rotational movement. These devices are capable of providing an output motion of one revolution or more but not continuous rotation.

Both type give bi-directional output motion, and most produce the same torque in both senses. Also output torque is generally constant throughout the stroke. There is no linkages and lost motion associated with cylinder- crank rod arrangement.

While the most often used actuators for pneumatic drives are cylinders for translational movements, there are many applications that require a turning or twisting movement of up to 360 degrees. Examples are turning components over in a drilling jig, providing a wrist action on a pick-and-place device or operating process valves. They are used in bench grinders, agitators, mixers, feeders, hoists, vibrators, pipe threaders etc.

i) Single Vane limited rotation actuators

The single vane actuator consists of a cylindrical housing, through which passes a central shaft to which the vane is rigidly attached. The housing has shoe or a stopper fixed to internal diameter of housing as shown in the Figure 1.43(a), thus dividing the interior space into two chambers. Pressurised air enters through port A and rotates the vane in the clockwise direction and air in the other chamber moves out of the port B. Similarly, when the air pressure is applied to the port B, the vane rotates in anti-clockwise direction and air in the other chamber moves out of the port A. Design geometry normally limits the rotary movement of a single vane actuator to about 280 maximum

ii) Double vane limited rotation actuators

It is possible to modify the design to have two vanes fixed to the output shaft 180 ° apart and two fixed stoppers in the casing providing two separate operating halves each with two chambers as shown in Figure 1.43(b). This gives twice the maximum torque output of a single vane device for the same supply pressure. Obviously the maximum angle of rotation is reduced and because of second stopper only 100° is usually possible.

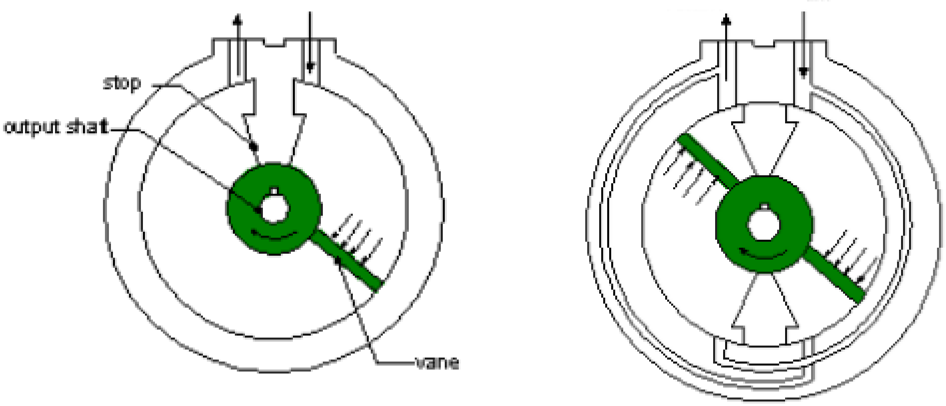
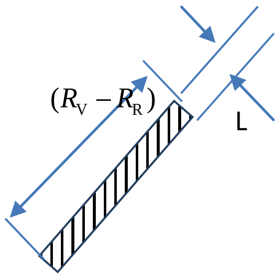


Figure 1.43 Vane actuators a) single vane b) double vane

iii) ANALYSIS OF TORQUE CAPACIT OF SINGLE VANE LIMITED ROTATION ACTUATOR

R_R = Outer radius of the output shaft (m)
R_V = Outer radius of the vane (m)
L = Width of the vane (m)
p = Hydraulic pressure (Pa)
F = Hydraulic force acting on the vane (N)
A = Surface area of vane in contact with oil (m^2)
T = Torque capacity (N m)



The force on the vane equals the pressure times the vane surface area:

$$F = pA = p(R_V - R_R)L$$

The torque equals the vane force times the mean radius of the vane:

$$T = p(R_V - R_R)L\frac{R_V + R_R}{2}$$

On rearranging, we have

$$T = p(R^2_V - R^2_R)L \text{ ----- (1)}$$

A second equation for torque can be developed by noting the following relationship for volumetric displacement V_D :

$$V_D = \pi(R^2_V - R^2_R)L \quad \text{----- (2)}$$

Combining equation 1 and 2 yields

$$T = \frac{p \times V_D}{2\pi} \quad \text{------- (3)}$$

For a two vane actuator the theoretical torque is twice that for a single vane. For the given design and style of vane actuator the torque rating is directly proportional to the maximum supply pressure. The volume displacement of a single vane actuator is given by

$$Q = \emptyset\ (R_V^2 - R_R^2) \times \frac{L}{8}$$

Where $\emptyset$ is rotary movement in radians

From this speed of rotation ω (rad/s) can be calculated for a supplied flow rate Q(mL/s) with the width and both diameters expressed in cm as

$$\omega = \frac{8Q}{L(R_V^2 - R_R^2)}$$

Although the total volume in a two vane actuator will be the same for the same body size, its speed will be halved for the same flow rate because two chambers must be filled. If necessary, speed can be controlled by throttling or by fitting orifice plugs controlling inflow or outflow. The most critical feature of the vane actuator is the length requiring sealing around the vane between the end faces and the internal bore of the casing. Since this a moving seal there is usual compromise between good sealing with low leakage and resulting higher friction. The sealing surface with the bore is also at the largest radius and operating at the higher sliding speed. Seals are usually pressure activated and may give a nonlinear torque relation with the pressure. In addition to the vane seal, further seals are required on the shaft where it emerges from the body of the actuator and between the shaft and the fixed stopper. The problem of leakage may also be evident where the actuator has to be locked with the load held by a pressurized fluid column as in a linear cylinder. The lock will not be positive due to internal leakage unless provision is made to supply extra fluid to compensate for the leakage. The shaft support may be plain bearing, roller bearing or needle roller type. Friction is responsible for the difference between the starting torque and dynamic torque. Starting torque can be estimated as 80% of the dynamic torque. Overall efficiency of vane actuator is between 70 % to 90%.

1.11.2 Rotary Actuator of Rack and Pinion Type

Schematic diagram of rotary actuator of rack and pinion type is shown in Figure 1.44. They are special duty rotary actuators. It has a high torque and small installation dimensions. The actuator has double pistons, which transmit the turning moment to the output shaft. The toothed piston rods act on the output shaft in a rack-and-pinion type arrangement. Each piston and toothed rod is of integral construction. The rack-and pinion type arrangement gives an even turning moment throughout the rotation movement. The drive shaft is robustly supported in bushings of self-lubricating type. There are key-ways on the output end of the shaft, while the opposite end of the shaft has a stub that can be used to accommodate end-position indication, or to facilitate hand operation of the actuator. The turning limits of the rotary actuator should be determined by external stop lugs, in order to protect the unit from the effects of excessive load inertia. Compressed air is fed into the piston chambers via a connection plate and drilled galleries in the central part and end covers. This rotary actuator has a cylinder block of natural anodized aluminum, with end covers of black anodized aluminum. The unit is available in 5 different sizes, covering a turning-moment range of 20 to 200 Nm. As standard, all sizes are available with a turning angle of either 90° or 180°. Three dimensional view of Rack and pinion type is shown in Figure 1.45

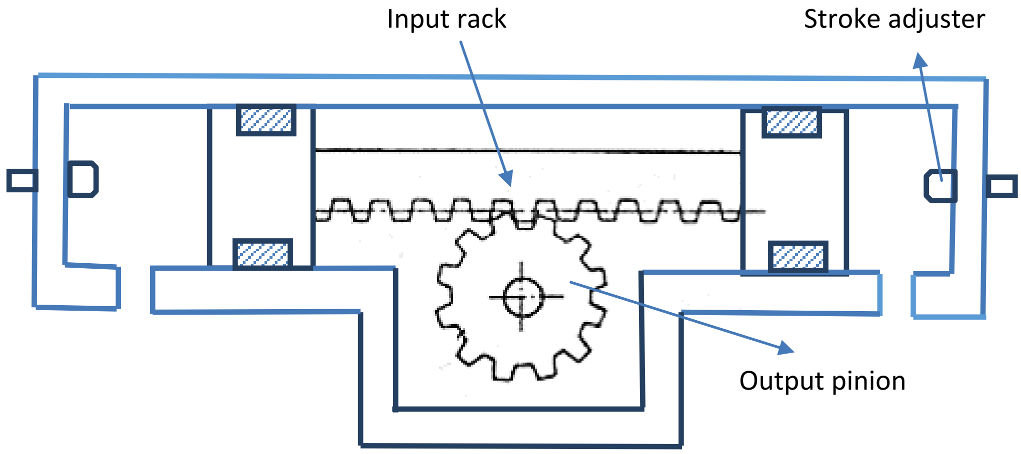


Figure 1.44 Rotary Actuator of Rack and Pinion Type (single rack)

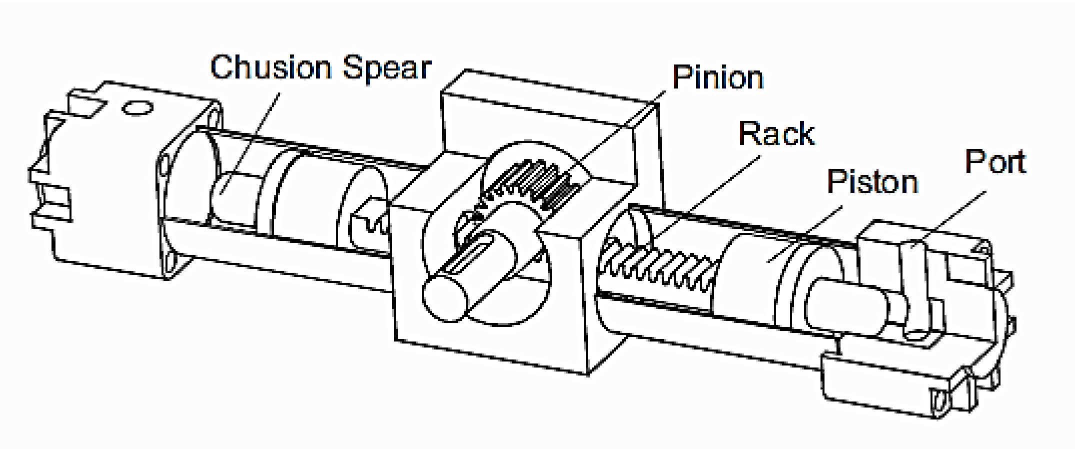


Figure 1.45 Three dimensional view of Rotary Actuator of Rack and Pinion Type

Torque can be doubled by adding another actuator as shown in the Figure 1.46

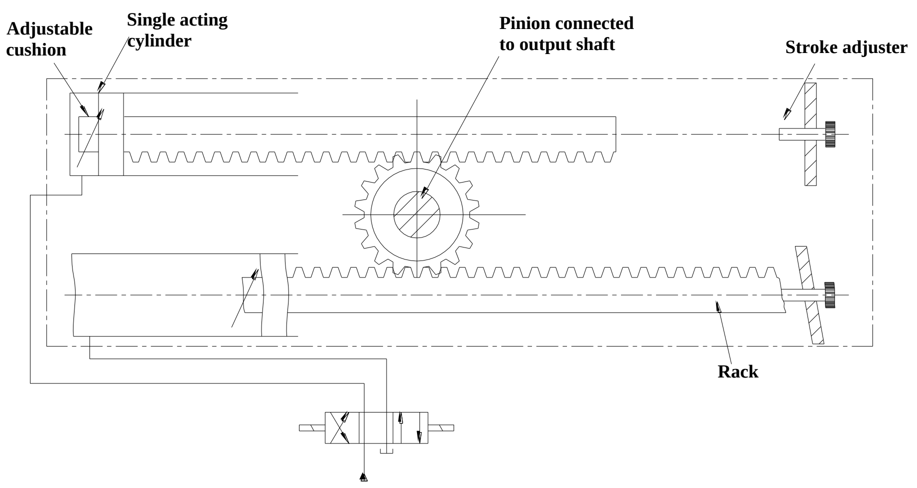


Figure 1.46 Rotary Actuator of Rack and Pinion Type (two rack)

5.11.3 Helix spine rotary actuator

Figure 1.47 shows a simplified cutaway view of a spiral-shaft rotary actuator. Thespiral-shaft rotary actuator has a keyed, non-rotating piston with a hollow rod. The hollow rod has a set of internal spiral grooves that mesh with the spiral shaft. The spiral-grooved shaft only has rotational movement and extends through the housing as an output shaft. With fluid piped to the CW port, the output shaft turns clockwise. With fluid piped to the CCW port, the output shaft turns counter clockwise

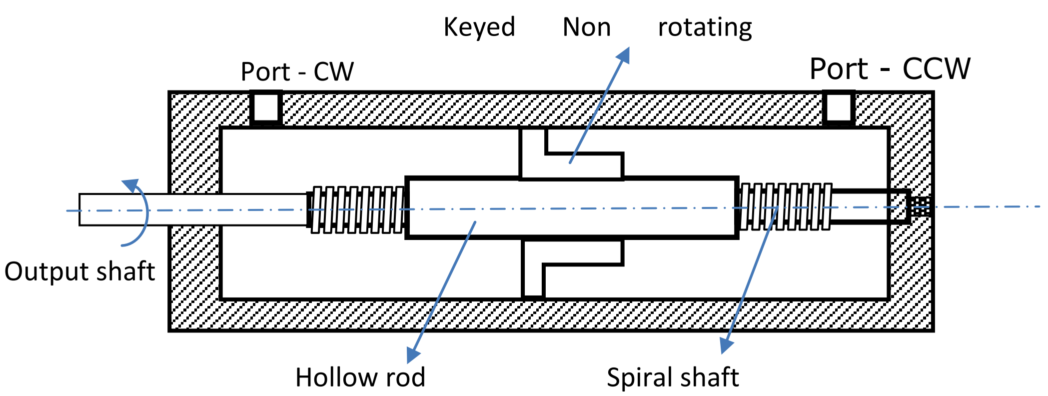


Figure 1.47 Helix spine rotary actuator

Objective Type Questions

1. Single acting cylinders can produce work in -------------- direction
2. In---------------------- type of cylinder, the force on the piston rod is almost doubled.
3. Telescopic cylinders are normally used for ---------------- stroke lengths
4. In first order lever system, fixed hinge point is located ----------- the cylinder and loading point
1. Cylinder with -------mounting generally require strong machine frames to resist bending moments

6. The single acting cylinder converts the compressed air energy into ------in the form of force and linear movement in one direction only.

7. Pneumatic ------ element convert the energy in the compressed air into force and motion. The pneumatic drive elements can move in a linear, reciprocating or rotating motion.

8. Typically, the piston can have diameters of as much as ---- mm

9.------- acting cylinders are used in the assembling and packing automated lines to move, lift, feed, eject, press or push objects or to clamp parts. Practically, they are suitable for oil- free operation.

10. The compressed air is applied only on the bottom side of the piston that is why the cylinder can move loads or perform mechanical work in a forward motion only and that the effective force is reduced by the ---------

11.The working speed of piston is in the range of 50 to ----- mm/s

12 In -----------type of cylinder , it is equipped with a compressible chamber with flexible sides that can be expanded to draw air in and compressed to force the air out

13.An alternative construction is a single acting cylinder, spring ------- with a spring in the piston area causing the piston to extend. Such pistons are used in the automotive industry and these are mounted in the air brakes for vehicles and trains

14.Another special form of single-acting cylinders is -------return spring in which the piston's return stroke is caused by external forces or by its own weight.

11.Short stroke cylinders are used when short strokes are required together with high ------

State True or False

1. Telescopic cylinder is used when long stroke length and short retracted length are required.

2.. Compared to first class lever, Second-class lever also results in a smaller load stroke for a given cylinder stroke

3 For a second-class lever system the cylinder rod pin lies between the load road pin and fixed-hinge pin of the lever.

4. In first order lever system, the loading point is in between the cylinder and the hinge point.

1. In pneumatic cylinders dynamic thrust can be taken as 0.1 times the static thrust

Review Questions

1. What is the function of a pneumatic actuator?

2. How can you classify the pneumatic actuators?

3. How do hydraulic actuators differ from pneumatic actuators?

4. What is the function of a pneumatic cylinder?

1. Explain the working of double acting double rod cylinder with a neat sketch

6. Explain the working of tandem and telescopic cylinders with a neat sketch

7. Mention few applications of pneumatic cylinders

8. Differentiate between single acting and double acting cylinder

9. What are the main advantages of magnetic cylinder?

Answers

Fill in the Blanks

1. only one
2. Tandem
3. long
4. in-between
1. non-centreline

6. mechanical energy
7. drive
8. 100mm
9. single
10. return spring
11. 500
12. diaphragm
13.extend
14. without
11. forces

State True or False

1. True
2. True
3. False
4. False
1. False

Lecture 38

PNEUMATIC CONTROL VLAVES

Learning Objectives

Upon completion of this chapter, Student should be able to

- Define the function of a valve
- Classify the valves
- Identify the DCVs as per ISO designation
- Explain the various types of Directional control valves
- Explain the various method of valve actuation
- Describe the function of various Non return valves
- Understand the working of quick exhaust valves
- Differentiate pressure control valve and sequence valve

1.1 VALVES

Valve are defined as devices to control or regulate the commencement, termination and direction and also the pressure or rate of flow of a fluid under pressure which is delivered by a compressor or vacuum pump or is stored in a vessel.

Values of one sort or another, perform three main function in pneumatic installation

- They control the supply of air to power units, example cylinders
- They provide signal which govern the sequence of operation
- They act as interlock and safety devices

The type of valve used is of little importance in a pneumatic control for most part. What is important is the function that can be initiated with the valves, its mode of actuation and line connection size, the last named characteristics also determining the flow size of the valve. Valves used in pneumatics mainly have a control function that is when they act on some process, operation or quantity to be stopped. A control function requires control energy, it being desirable to achieve the greatest possible effect with the least effort. The form of control energy will be dictated by the valve's mode of actuation and may be manual, mechanical, electrical hydraulic or pneumatic.

Valve available for pneumatic control can be classified into four principal groups according to their function:

1. Direction control valve
2. Non return valves
3. Flow control valves
4. Pressure control valves

1.2 DIRECTION CONTROL VALVES

Pneumatic systems like hydraulic system also require control valves to direct and regulate the flow of fluid from the compressor to the various devices like air actuators and air motors. In order to control the movement of air actuators, compressed air has to be regulated, controlled and reversed with a predetermined sequence. Pressure and flow rates of the compressed air to be controlled to obtain the desired level of force and speed of air actuators.

The function of directional control valve is to control the direction of flow in the pneumatic circuit. DCVs are used to start, stop and regulate the direction of air flow and to help in the distribution of air in the required line.

6.2.1 TYPES OF DIRECTION CONTROL VALVES

Directional valves control the way the air passes and are used principally for controlling commencement, termination and direction of air flow. The different classification scheme of the pneumatic cylinders are given below

1. Based on construction

    i) Poppet or seat valves

    - Ball seat valve
    - Disc seat valve
    - Diaphragm Valves

    ii) Sliding spool valves

    - Longitudinal slide valve
    - Suspended spool valves
    - Rotary spool valves

2. Based on the Number of ports

i) Two way valves

ii) Three way valves

iii) Four way valves

3. Based on methods of actuation

i) Mechanical

ii) Electrical

iii) Pneumatic

4. Based on Size of the port

Size refers to a valve's port size. The port sizes are designated as M5, G1/8, and G1/4 etc. M refer to Metric thread, G refer to British standard pipe (BSP) thread.

5. Based on mounting styles

i) Sub base

ii) Manifold

iii) In-line

iv) Valve island

6.2.1.1 ISO DESIGNATION OF DIRECTION CONTROL VALVES

Valves are represented by symbols because actual construction is quite complex. A symbol specifies function of the valve, method of actuation, no of ports and ways. Pneumatic symbols have been standardised in ISO 1219-1:2006. (Fluid power systems and components – Graphic symbols and circuit diagram). Another standard ISO 1219-2:1995 establishes the rules for drawing diagrams of fluid power systems using symbols from ISO 1219-1. Port designations are described in ISO 5599.

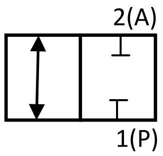
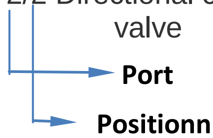
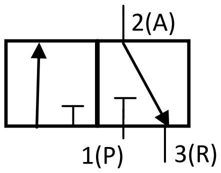
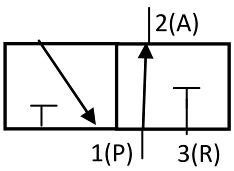
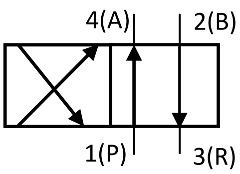
Port markings: As per the ISO 5599, ports are designated using a number system. Earlier, a letter system was used to designate a port. Table 1.1 gives port markings.

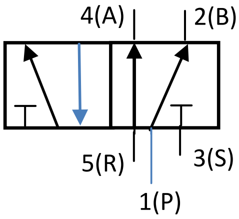
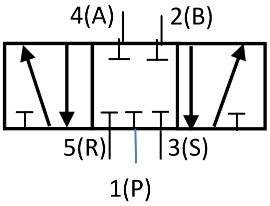
Port	Old (Letter) system	ISO (Number) System	Remarks
Pressure port	P	1	Supply port
Working port	A	2	3/2 DCV
Working ports	A, B	4, 2	4/2 or 5/2 DCV
Exhaust port	R	3	3/2 DCV
Exhaust ports	R, S	5,3	%/2 DCV
Pilot ports	Z or Y	12	Pilot line (flow 1-2)
Pilot ports	Z	14	Pilot line (flow 1-4)
Pilot ports	Z or Y	10	Pilot line (no flow)
Internal pilot ports	Pz, Py	81, 91	Auxiliary pilot line

Table 1.1: Port Markings of Direction Control Valve

Ports and position: DCVs are described by the number of port connections or ways they control. For example: Two way, three – way, four way valves. Table 1 shows the Port markings of DCVs and Table 1.2 shows commonly used DCVs with old and new designations.

Table 1.2: Port designation of DCV

Port and position	
	2/2 Directional control valve 
	3/2 Directional control valve (normally closed)
	3/2 Directional control valve (normally open)
	4/2 Directional control valve

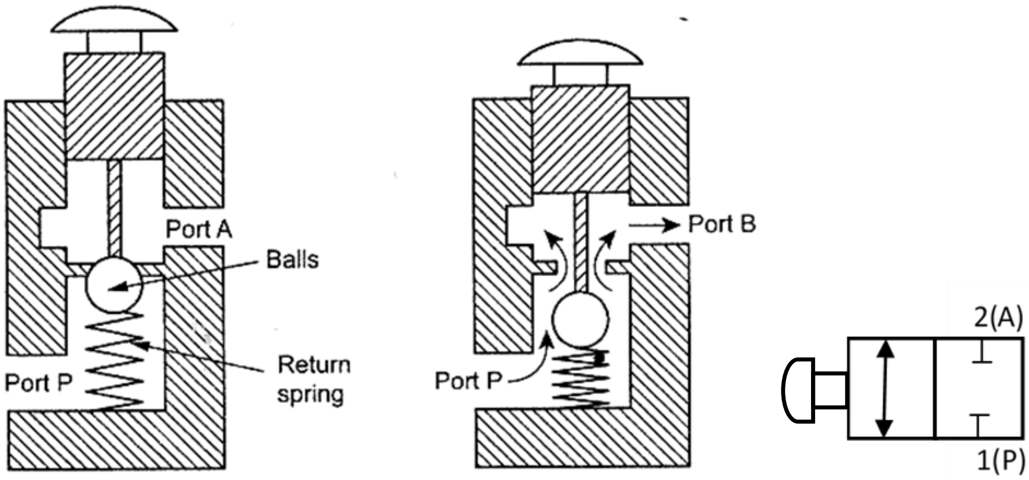
	5/2 Directional control valve
	5/3 Directional control valve

1.2.1.2 POPPET DIRECTION CONTROL VALVES

There are two different types of poppet valves, namely ball seat valve and disc seat valve.

A.Ball seat valve.

In a poppet valve, discs, cones or balls are used to control flow. Figure 1.1 shows the construction of a simple 2/2 normally closed valve. If the push button is pressed, ball will lift off from its seat and allows the air to flow from port P to port B. When the push button is released, spring force and air pressure keeps the ball back and closes air flow from port P to port B. Valve position are shown in Figure 1.1(a) 1.1 (b)   1.1(C)



(a) Poppet position 1 *(b) Poppet position 2* **(c) Symbol**

Figure 1.1 Two/Two Ball seat Poppet valve

B. Disc seat poppet valve

Figure 1.2 shows the construction of a disc type 3/2 way DCV. When push button is released, ports 1 and 3 are connected via hollow pushbutton stem. If the push button is pressed, port 3 is first blocked by the moving valve stem and then valve disc is pushed down so as to open the valve thus connecting port 1 and 3. When the push button is released, spring and air pressure from port 1 closes the valve.. Comparison between Ball seat and disc seat valve is given in Table 1.3

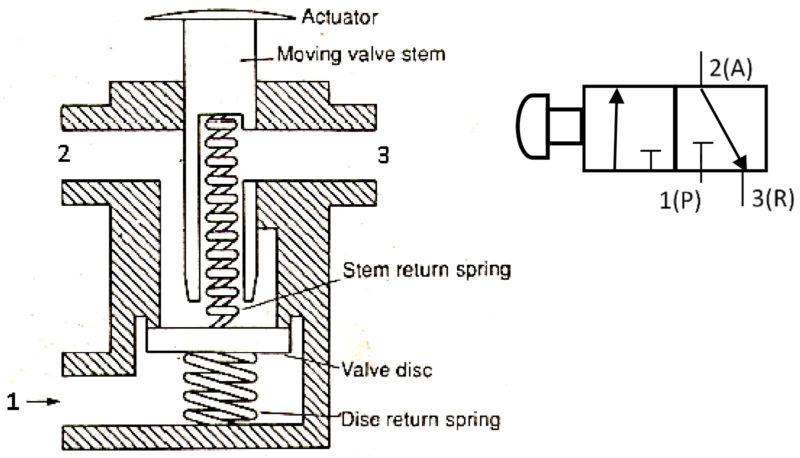


Figure 1.2 Disc seat poppet valve

Advantages of poppet valves are as follows

i) Response of poppet valve is very fast- short stroke to provide maximum flow opening
ii) They give larger opening (larger flow) of valves for a small stroke
iii) The valve seats are usually simple elastic seals so wear is minimum
iv) They are insensitive to dust and dirt and they are robust, seats are self cleaning
v) Maintenance is easy and economical.
vi) They are inexpensive
vii) They give longer service life: short stroke and few wearing parts give minimum wear and maximum life capabilities

Disadvantages of poppet valves are as follows

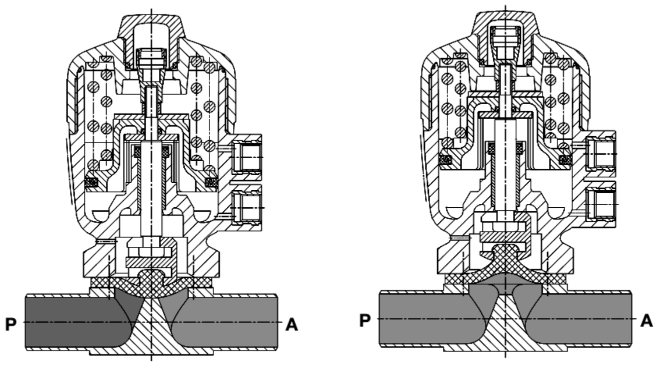
i) The actuating force is relatively high, as it is necessary to overcome the force of the built in reset spring and the air pressure.
ii) They are noisy if flow fluctuation is large.

Table 1.3 Comparison of Ball seat and Disc seat valves

Ball seat valves	*Disc seat valve*
They are inexpensive	Offer large area and lift required is very small
They are relatively small in size	Time response is good
Insensitive to dirt and dust	Insensitive to dirt and dust
Can be operated manually or mechanically	Can be actuated manually , mechanically, electrically or pneumatically

C.Diaphragm valves

The diaphragm between the actuator and valve body hermetically isolates the fluid from the actuator. The valves are maintenance-free and extremely robust and can be retrofitted with a comprehensive range of accessories, e.g. electrical position feedback, stroke limitation or manual override. **Figure 1.4** shows unactuated and actuated position of diaphragm valves.



a) Unactuated b) actuated

Figure 1.4 Diaphragm valve: unactuated position, actuated position

Closed position: When de-energized, the valve is closed by spring action

Open position: If the actuator is pressurized by the control pressure, it simultaneously lifts the control piston and the valve spindle to open the valve.

6.2.1.3 SPOOL DIRECTION CONTROL VALVES

A. Hand operated 3/2 DCV

The cross sectional views of 3/2 DCV (normally closed) based on spool design is shown below. When the valve is not actuated, port 2 and 3 are connected and port 1 is blocked. When the valve is actuated then port 2 and 1 are connected and port 3 is blocked.

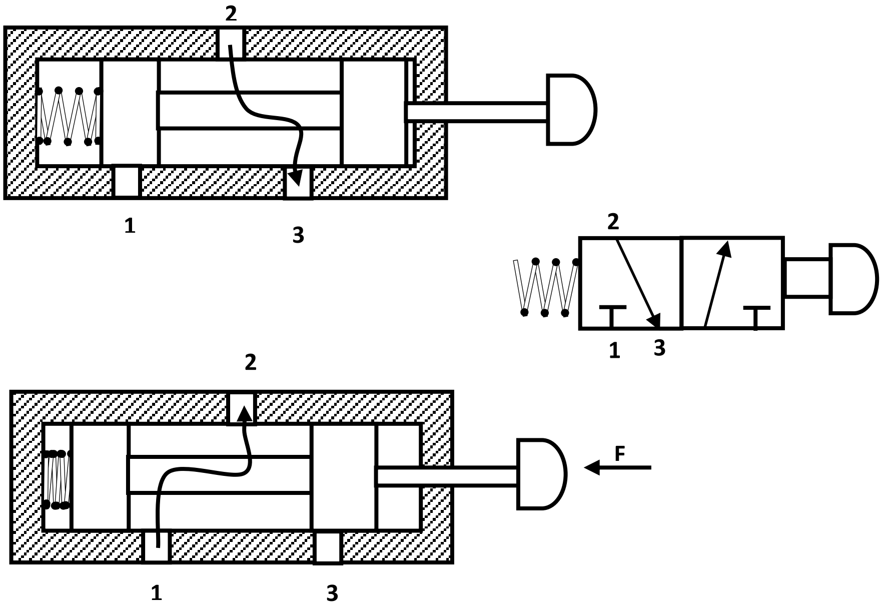


Figure 1.5 3/2 Directional control valve (Normally closed)

Figure 1.5 shows schematic diagram of 3/2 spring operated valve. There are three ports common port, normally open port and normally closed. When the valve is not actuated, there is flow from NO port to common port. When the valve is actuated there is flow from NC to common port.

B.Pneumatically actuated 3/2 DCV

The cross – sectional views of pneumatically actuated NC type 3/2 DCV in normal position and actuated positions are shown in the Figure 1.7

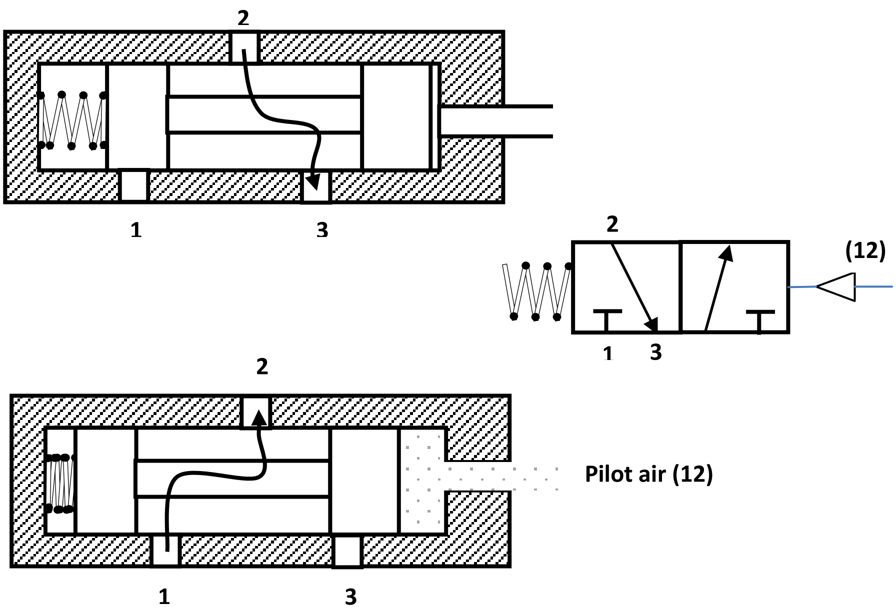


Figure 1.7 3/2 Directional control valve (pneumatically operated)

In normal position, the working port (2) is closed to the pressure port (1) and open to the exhaust port (3). When the compressed air is applied through the pilot port (12), the spool is moved against the spring. In the actuated position, the working port (2) is open to the pressure port(1) and closed to the exhaust port(3). Thus, the application of the compressed air to the port 12 causes the pressure port (1) to be connected to the working port (2).

Pneumatically actuated valves have following advantages

i. Great flexibility for use in simple as well as complex control system
ii. Adaptability for use in safety circuits.
iii. Various control functions can be easily incorporated as and when required
iv. Feedback signals from sensors can be applied conveniently for the purpose of controlling the pilot ports of these main valves. This means existing pneumatically actuated control circuits can be modified easily to incorporate any additional control requirement.

C.Pneumatically actuated 4/2 DCV

The valve shown in Figure 1.9 is a 4/2 way valve pneumatically operated DCV. Switch over is effected by direct application of pressure. If compressed air is applied to pilot spool through control port 12, it connects port 1 with 2 and 4 is exhausted through port 3. If the pilot pressure is applied to port 14, then 1 is connected with 4 and line 2 exhausted through port 3. On disconnecting the compressed air from the control line, the pilot spool remains in its current position until spool receives a signal from the other control side.

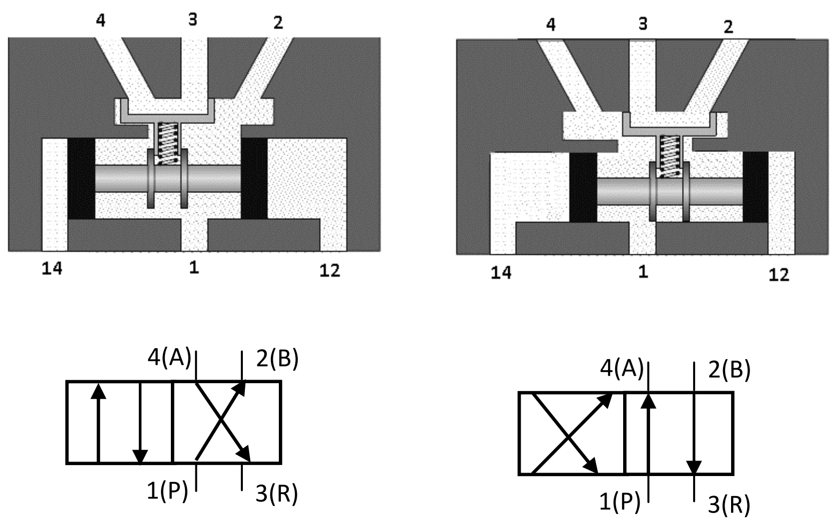


Figure 1.9 Schematic diagram of 4/2 way valve

D. Suspended Disc Direction Control Valves

This valve is quite similar to 4/2 way spool valve. Schematic diagram is shown in Figure 1.11. In this design disc is used instead of a spool. This suspended disc can be moved by pilot pressure or by solenoid or by mechanical means. In this design, main disc connects port 1 to either port 4 or 2. The secondary seat discs seal the exhaust port 3 whichever is not functional. These values are generally provided with manual override to manually move the cylinder.

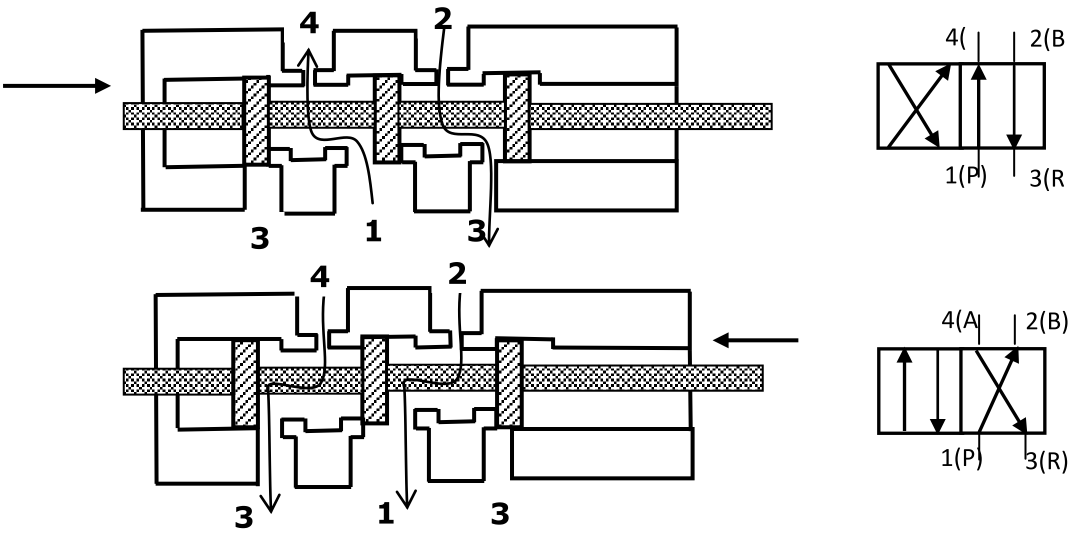


Figure 1.11 4/2 Directional control valve (suspended disc type)

Figure 1.12 below shows 5/2 way valve which uses suspended disk instead of spool. In spool type valve, spool controls the opening and closing of ports. In this type, suspended disc controls the opening and closing of ports. This suspended disc can be moved by pilot pressure at port 14 or port 12. When the pilot pressure acts through port 14. The ports 1 - 2 and 4 - 5 are connected and 3 is blocked. When the air is given to pilot line 12, then 2 - 3 and 4 -1 are connected and 5 is blocked

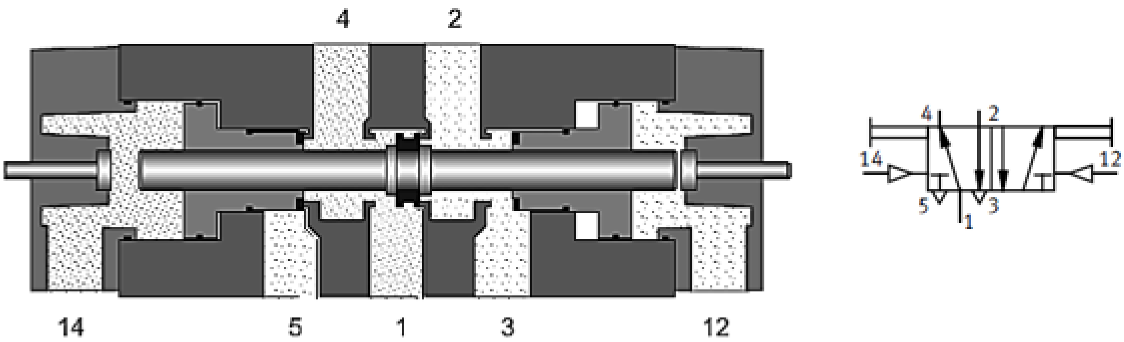


Figure 1.12 5/2 Directional control valve (suspended disc type)

Advantages

i) They have short actuation movement

ii) They are quick to operate because of small switching movement

iii) If signals are applied at both ports, first signal will be dominant

Disadvantages

i) Construction of the valve is complex

ii) Expensive

E.Rotary valves

The rotary spool directional control valve (Figure 1.13) has a round core with one or more passages or recesses in it. The core is mounted within a stationary sleeve. As the core is rotated within the stationary sleeve, the passages or recesses connect or block the ports in the sleeve. The ports in the sleeve are connected to the appropriate lines of the fluid system.

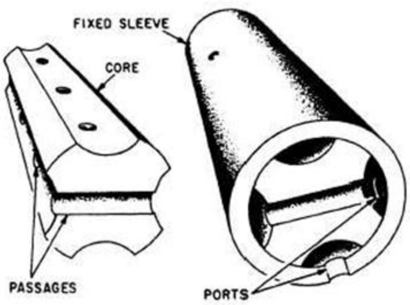


Figure 1.13 Parts of a rotary spool directional control valve.

Figure 1.14 shows the construction of a rotary spool directional control valve. We connect different ports by rotating the handle. By rotating the handle, core gets connected to different holes to give the required configuration of the valve. This type of the valve can be directly mounted on panel using bolt.

Figure 1.15 shows three different position of the core when the handle is rotated. Left most envelope of DCV connects P to B and A to T. Middle envelope of DCV blocks all ports. Right most envelope of DCV connects P to A and T to B.

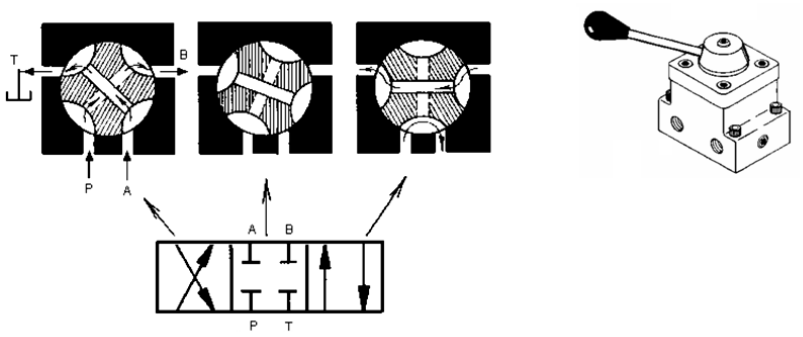
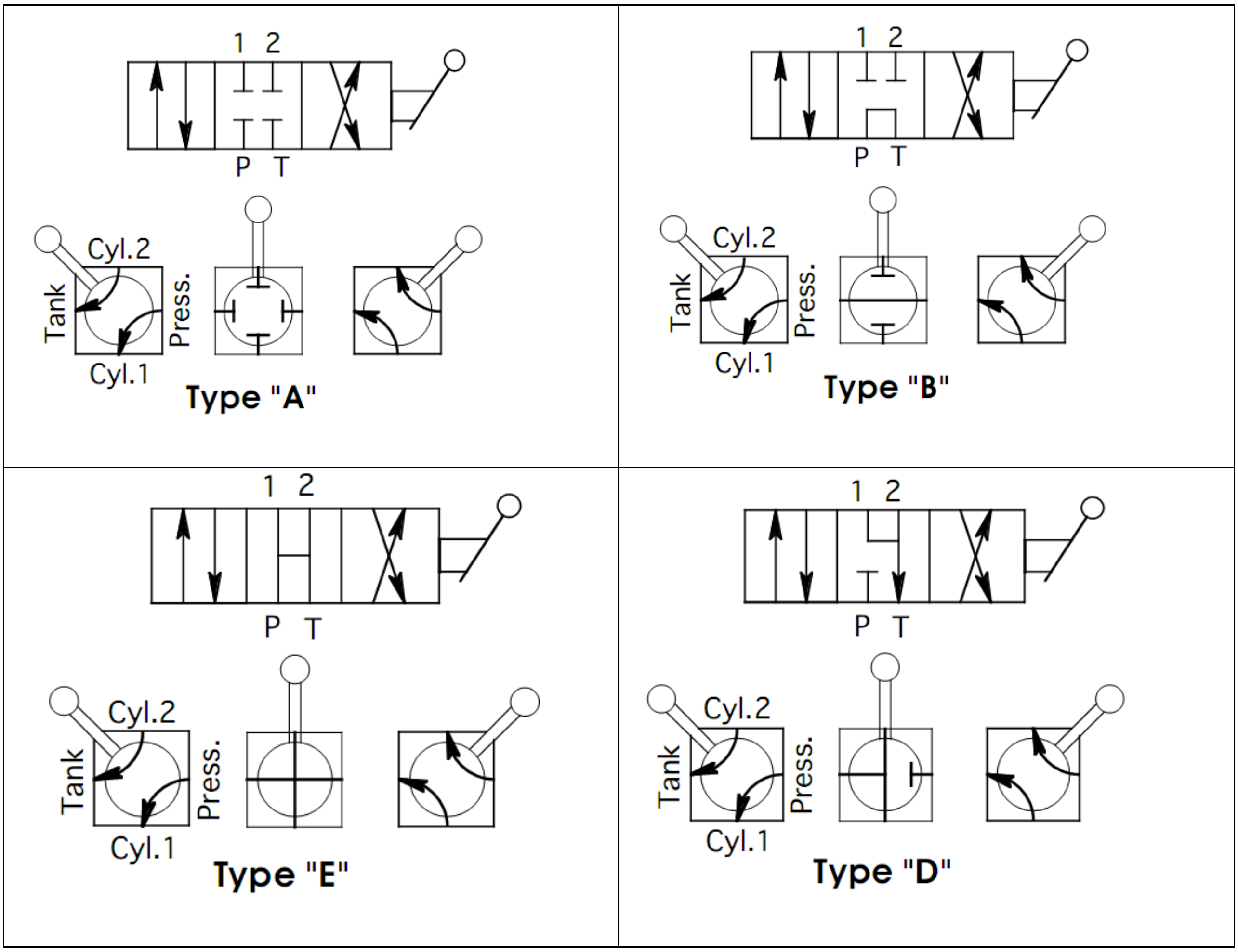


Figure 1.15 Three different positions of 4/3 way rotary spool directional control valve.

Table 1.4 shows schematically the different position of core and sleeve for various middle position of 4/3 way Direction control valve.

Table 1.4 Different position of core and sleeve for various mid position of 4/3 DCV



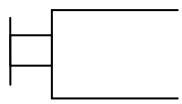
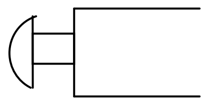
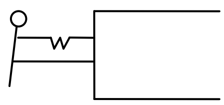
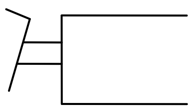
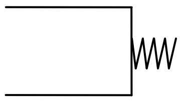
6.2.1.5 METHODS OF ACTUATION.

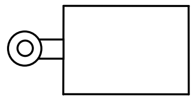
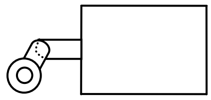
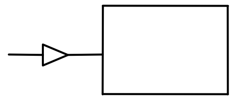
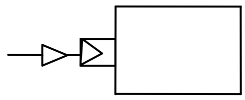
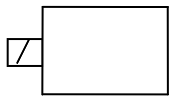
The methods of actuation of pneumatic directional control valves depend upon the requirements of the task. (Table 1.5) The types of actuation vary;

- manually actuated
- mechanically actuated
- pneumatically actuated
- electrical
- combined actuation

The symbols of the methods of actuation are detailed in DIN ISO 1219. When applied to a directional control valve, consideration must be given to the method of initial actuation of the valve and also the method of return actuation. Normally these are two separate methods. They are both shown on the symbol on either side of the position boxes. There may also be additional methods of actuation such as manual overrides, which are separately indicated.

Table 1.5 Methods of actuation

Type of actuation	Type of control	Symbol
Manual	General	
	Pushbutton	
	Detent lever operated	
	Foot pedal	
Mechanical	Spring return	
	Spring centered	

	Roller operated	
	Idle roller	
Pneumatic	Direct	
	Indirect, pilot operated	
Electrical	Single solenoid	
	Double solenoid	
Combined	Double solenoid with pilot operated	

1.2.1.7 BASED ON MOUNTING STYLES

Directional control valves can be mounted in two ways; inline and subplate.

Inline means that there are threaded connections in the valve itself. Fittings are screwed directly into the valve. This method has several disadvantages. Each time the valve is disconnected there is the possibility of damaging the valve by stripping the threads. The threads will also wear each time the unit is disconnected, causing contamination and an increased possibility of leakage. In the subplate method, the bottoms of the valves have unthreaded connections. The valve is then attached to a subplate that has matching connections.

The subplate has the threaded connection to which the fittings are attached. Sealing at the valve /subplate interface is accompanied through the use of o-rings, which fit into small recesses around the DCV ports. The subplate methods results in less leakage, less contamination, and a smaller probability of doing damage to the valve during assembly and disassembly. Valve replacement is simpler and less time – consuming task.

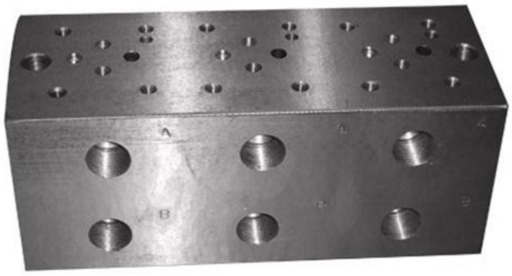


Figure 1.34 Manifold for three valves

1.2.2 NON RETURN VALVES

Non return valves permit flow of air in one direction only, the other direction through the valve being at all times blocked to the air flow. Mostly the valves are designed so that the check is additionally loaded by the downstream air pressure, thus supporting the non-return action.

Among the various types of non-return valves available, those preferentially employed in pneumatic controls are as follows

i) Check valve
ii) Shuttle valve
iii) Restrictor check valve
iv) Quick exhaust valve
v) Two pressure valve

A. Check valve

The simplest type of non-return valve is the check valve (Figure 1.35 (a)) , which completely blocks air flow in one direction while permitting flow in the opposite direction with minimum pressure loss across the valve. As soon as the inlet pressure in the direction of free flow develops a force greater than that of the internal spring, the check is lifted clear of the valve seat. The check in such valve may be plug, ball, plate or diaphragm.

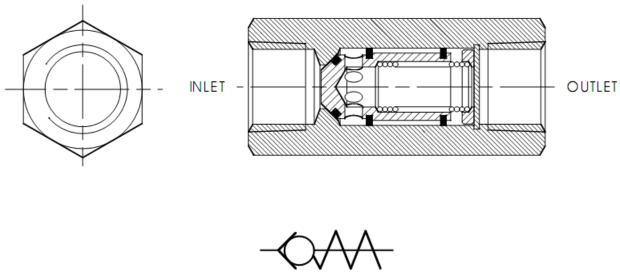


Figure 1.35 Check valve

B. Shuttle valve

It is also known as a double control valve or double check valve. A shuttle valve has two inlets and one outlet. At any one time, flow is shut off in the direction of whichever inlet is unloaded and is open from the loaded inlet to the outlet (Figure 1.36). A shuttle valve may be installed, for example, when a power unit (cylinder) or control unit (valve) is to be actuated from two points, which may be remote from one other.

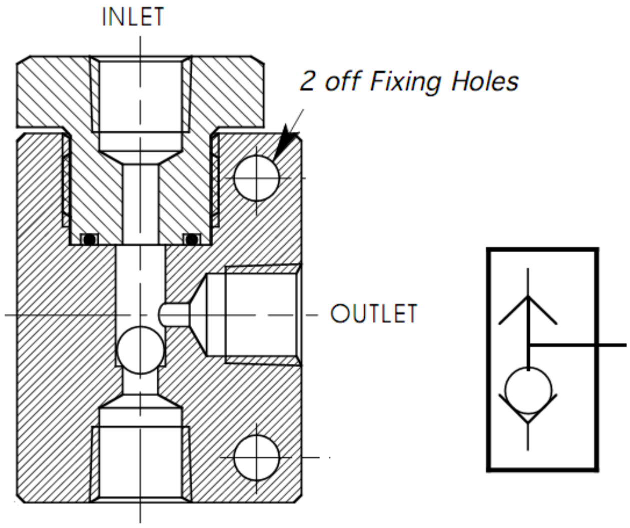


Figure 1.36 Shuttle valve

C. **Restrictor check valve**

It also termed speed control valve for pneumatic applications are actually hybrid type of unit. By reason of their throttling function they are flow control valves and they are indeed used as flow control valves in pneumatics. Incorporation of check function also makes them non –return valves and it is as such that they are generally classified.

Usually the throttle of a restrictor check valve is adjustable so as to permit regulations of air flow through the valve. Throttling function is effective only in one direction of flow, while in the other direction free flow is provided through the check.(Figure 1.37). When restrictor check valves are used to control the speed of pneumatic cylinders, differentiation is made between supply-air and exhaust air-throttling.

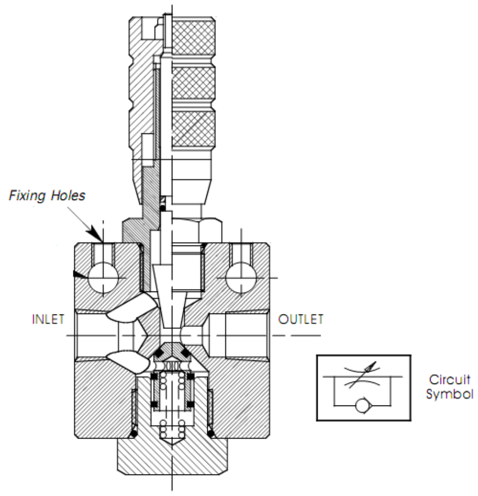


Figure 1.37 Functional diagram of restrictor check valve.

D. Quick Exhaust Valves

A quick exhaust valve is a typical shuttle valve. The quick exhaust valve is used to exhaust the cylinder air quickly to atmosphere. Schematic diagram of quick exhaust valve is shown in Figure 1.38. In many applications especially with single acting cylinders, it is a common practice to increase the piston speed during retraction of the cylinder to save the cycle time. The higher speed of the piston is possible by reducing the resistance to flow of the exhausting air during the motion of cylinder. The resistance can be reduced by expelling the exhausting air to the atmosphere quickly by using Quick exhaust valve.

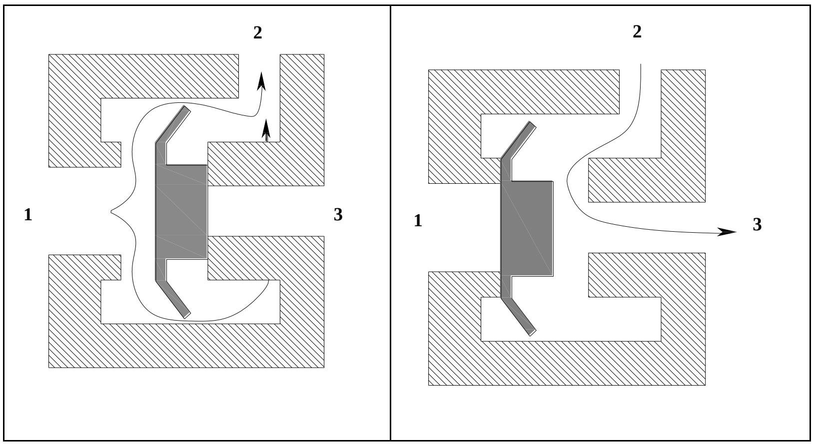


Figure 1.38 Functional diagram of quick exhaust valve.

The construction and operation of a quick exhaust valve is shown in Figure 1.38. It consist of a movable disc (also called flexible ring) and three ports namely, Supply port 1, which is connected to the output of the final control element (Directional control valve). The Output port, 2 of this valve is directly fitted on to the working port of cylinder. The exhaust port, 3 is left open to the atmosphere

Forward Motion: During forward movement of piston, compressed air is directly admitted behind the piston through ports 1 and 2 Port 3 is closed due to the supply pressure acting on the diaphragm. Port 3 is usually provided with a silencer to minimise the noise due to exhaust.

Return Motion: During return movement of piston, exhaust air from cylinder is directly exhausted to atmosphere through opening 3 (usually larger and fitted with silencer) .Port 2 is sealed by the diaphragm. Thus exhaust air is not required to pass through long and narrow passages in the working line and final control valve

Typical applications of quick exhaust valves for single acting and double acting cylinders are shown in Figure 1.39

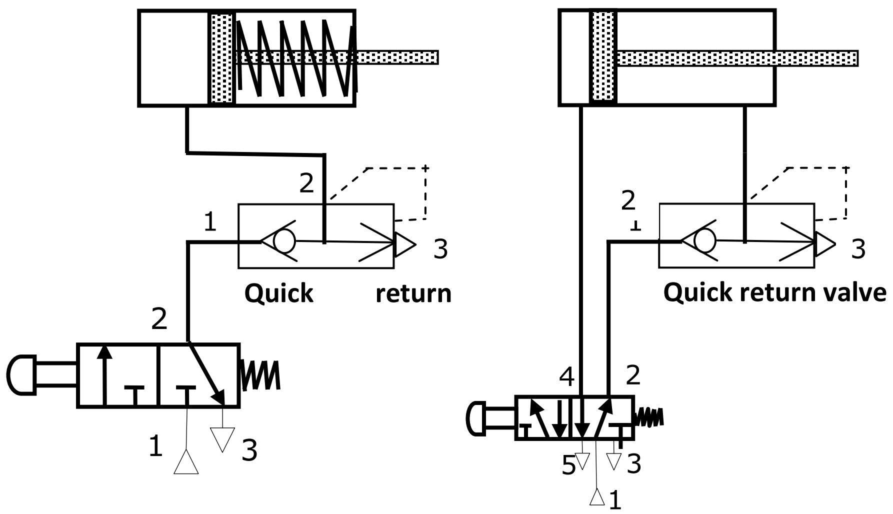


Figure 1.39 Application of quick exhaust valve.

E. Two Pressure Valve

This valve is the pneumatic AND valve. It is also derivate of Non Return Valve. A two pressure valve requires two pressurised inputs to allow an output from itself. The cross sectional views of two pressure valve in two positions are given in Figure 1.40 As shown in the figure, this valve has two inputs 12 and 14 and one output 2. If the compressed air is applied to either 12 or input 14, the spool moves to block the flow, and no signal appears at output 2. If signals are applied to both the inputs 12 and 14, the compressed air flows through the valve, and the signal appears at output 2.

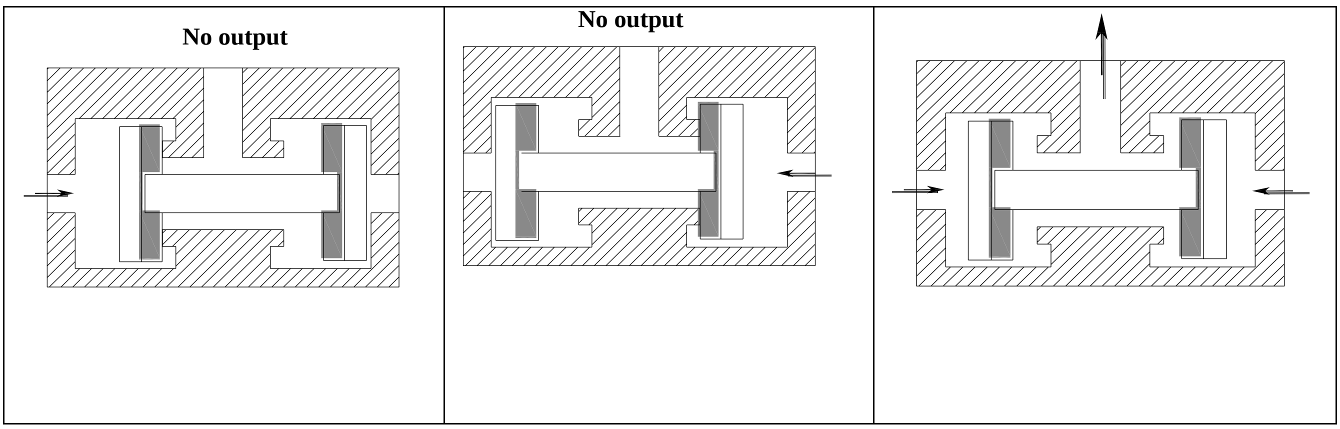


Figure 1.40 Two pressure valve.

1.2.3 FLOW CONTROL VALVES

Function of a flow control valve is self –evident from its name. A flow control valve regulates the rate of air flow. The control action is limited to the air flow passing through the valve when it is open, maintaining a set volume per unit of time. Figure 1.41(a) shows a variable restrictor type flow control valve (manifold type). Figure 1.41(b) shows a variable restriction type flow control valve (inline type). Figure 1.42 shows another design of Flow control valve, in which flow can be set by turning the knob.

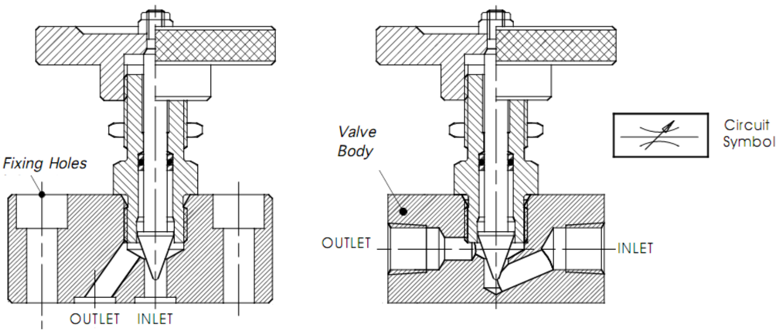


Figure 1.41 Flow control valve a) manifold b) inline

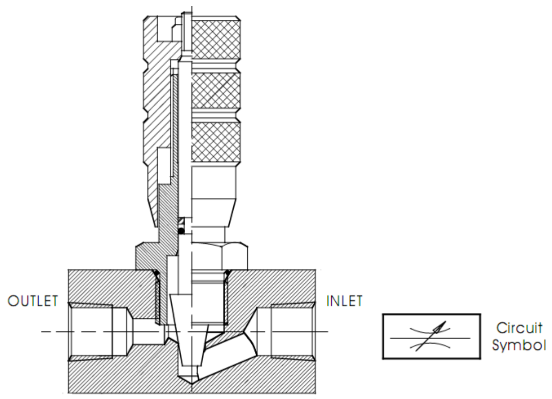


Figure 1.42 Flow control valve (adjustable)

1.2.4 PRESSURE CONTROL VALVE.

Compared with hydraulic systems, few pressure control valves are brought into use in pneumatics. Pressure control valves control the pressure of the air flowing through the valve or confined in the system controlled by the valve.

There are three types of pressure control valves

1. Pressure limiting valve

2. Pressure sequence valve
3. Pressure regulator or pressure reducing valve

A.Pressure limiting valve.

Prevents the pressure in a system from rising above a permissible maximum. Construction feature of pressure limiting valve is shown in Figure 1.43. It is a standard feature of compressed air production plant but is hardly ever used in pneumatic controls. These valves perform a safety relief function by opening to the atmosphere if a predetermined pressure is exceeded in the system, thus releasing the excess pressure. As soon as the pressure is thus relieved to the desired figure, the valve closed again by spring force.

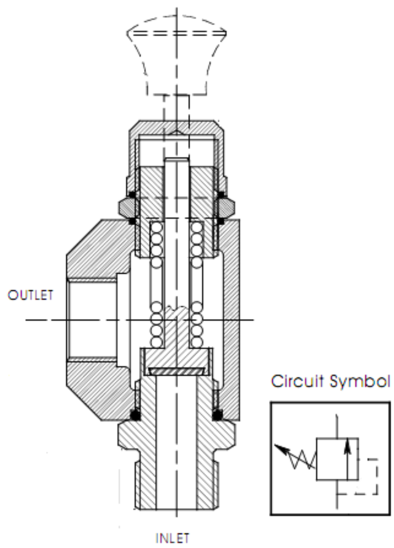


Figure 1.43 Pressure limiting valve

B.Pressure sequence valve

Function of the sequence valve is very similar to that of a pressure limiting valve. It is however used for a different purpose. Outlet of the pressure sequence valve remains closed until pressure upstream of it builds up to a predetermined value. Only then the valve opens to permit the air from inlet to outlet. Sequence valve must be incorporated into a pneumatic control where a certain minimum pressure must be available for a given function and operation is not be initiated at any pressure lower than that. There are also used in systems containing priority air consumers, when other consumers are not to be supplied with air until ample pressure is assured.

C.Pressure reducing valve or regulator

Pressure regulators, commonly called pressure-reducing valves, maintain constant output pressure in compressed-air systems regardless of variations in input pressure or output flow. Regulators are a special class of valve containing integral loading, sensing, actuating, and control components. Available in many configurations, they can be broadly classified as general purpose, special purpose, or precision. Three dimensional view of pressure reducing valve is shown in Figure 1.44

General-purpose or utility regulators have flow and regulation characteristics that meet the requirements of most industrial compressed-air applications. Such regulators provide long service life and relative ease of maintenance at competitive prices. Precision regulators are for applications where regulated pressure must be controlled with close tolerances. Such regulators are used when the outcome of a process or the results of a test depend on accurate pressure control.

Special-purpose regulators often have a unique configuration or special materials for use with fluids other than compressed air. Regulator construction can range from simple to complex, depending on the intended application and the performance requirements.

However, the principle of operation and the loading, actuating, and control components are basic to all designs. Most regulators use simple wire coil springs to control the downstream pressure. Various size springs are used to permit regulation of the secondary pressure within specific ranges. Ideally, the required pressure should be in the center one-third of the rated outlet pressure range. At the lower end of the pressure range, the spring loses some sensitivity; at the high end, the spring nears its maximum capacity.

Regulators can use either a piston or diaphragm to sense downstream pressure. Diaphragms are generally more sensitive to pressure changes and react more quickly. They should be used where sensitive pressure settings are required (less than 0.0025 bar). Pistons, on the other hand, are generally more rugged and provide a larger effective sensing area in a given size regulator. The functional difference between precision and general-purpose regulators is the degree of control accuracy of the output pressure. Output pressure accuracy is determined by the droop due to flow changes (regulator characteristics).

Pressure droop is most pronounced when the valve first opens. Factors contributing to droop are: load change with spring extension, effective area change with diaphragm displacement, and unbalance of area forces on the valve. The amount that output pressure changes with variations in supply pressure is called the regulation characteristic and is influenced by the ratio of diaphragm area to valve area and the degree of valve unbalance.

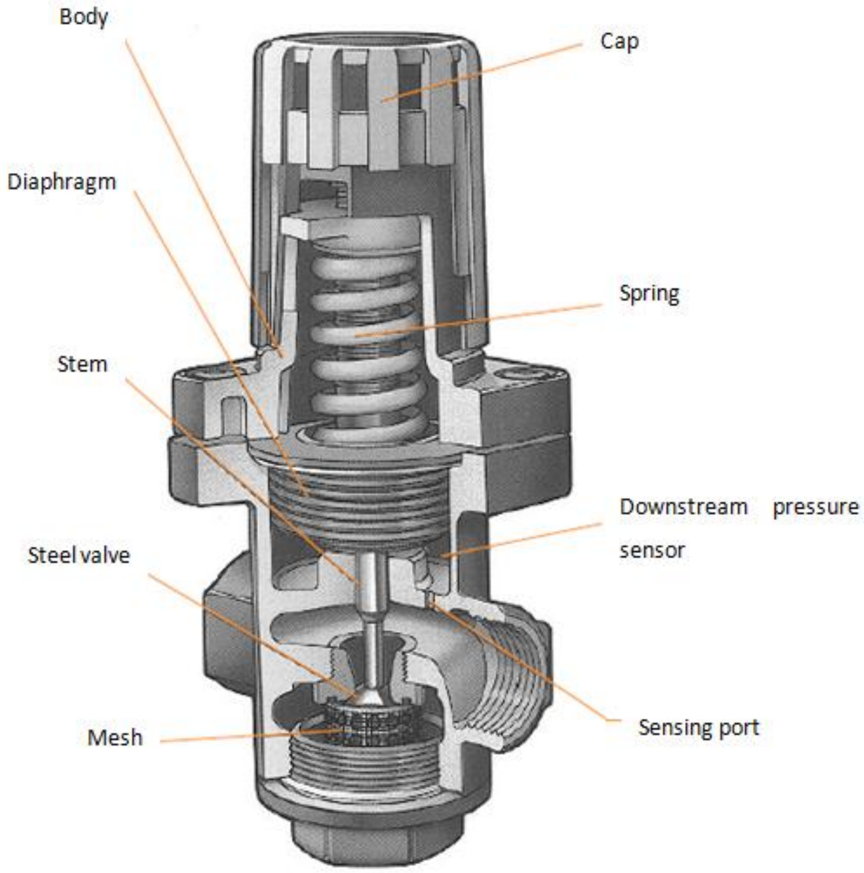


Figure 1.44 Three dimensional figure of pressure regulating valve.

When selecting a pressure regulator, the important factors to consider are:

1. Normal line pressure.
2. Minimum and maximum regulated pressure required: Regulators can have a broad adjustment range and may require a specific spring or accessory to match the requirements. Also, minimum and maximum pressure should be within the middle third of the regulator range.
3. Maximum flow required at regulated pressure.
4. Pipe size: Not all regulators are available in all pipe sizes; note where adapters are required. Also, pipe size should be consistent with flow requirements.
5. Regulator adjustment frequency: A number of different adjusting methods are possible. When selecting a regulator, consider the location, application, adjusting method, and user.
6. Degree of pressure precision required.
7. Accessories or options include gages and panel mounting.

8. Environmental or fluid conditions that could be incompatible with materials used in the regulator.
9. Special features such as high relief or remote control.
10. The consequences of a regulator malfunction or failure: A damper or relief valve might be needed to protect personnel or equipment. Also, dead-end service or intermittent actuation may require positive valve shutoff, bleed units, or close control of pressure-relief points. Filters, lubricators, relief devices, and other system options should be considered in the selection process.

Objective Type Questions

1. Valves are defined as devices to control, or regulate the commencement and -------of air
2. On resetting valves, for example those equipped with a return spring, the -----position is the position assumed by the moving parts of the valve when it is connected but not actuated.
3. Direct control of a valve means that valve is caused to operate directly by actuating element without any -----------elements being operated.
4. ----------------automatically limit flow to a single direction at the point where they are installed in an air line.
5. In pressure regulating valve inlet pressure is ----------than the outlet pressure.
6. Poppet valves give --------- stroke and few wearing parts give minimum wear
7. The quick exhaust valve is used to exhaust the cylinder air quickly to -----------------
8. compared with hydraulic systems, -----varieties of pressure control valves are used in pneumatics
9. As per ISO 1219-1:2006 designation, number 12 or 14 indicate ------------- ports
10. In rotary spool directional control valve , rotating part is called ----- and stationary part is called-----
11. The two most common basic flow control devices used in a pneumatic system are fixed-sized orifices and _____ valves.
12. The meter ----flow-control circuit is the preferred method to use for controlling the operating speed of cylinders in pneumatic circuits.

State True or False

1. Restrictor check valves are non return valves which are also employed as flow control valves.
2. Quick exhaust valves are designed to decrease the position speed in the cylinder
3 spring force set on a pressure limiting valve or sequence valve corresponding to minimum permissible or minimum desired pressure of the controlled fluid.
4. Ball seat valves ensure perfect sealing at all times in pneumatic circuit
5. Two way valves are used where a pure straightway function is required, that is when downstream equipment does not need exhausting to the atmosphere via this valve.
6. The pressure sequence valve holds the working pressure largely constant.
7. Two pressure valve is the pneumatic OR valve

8. Shuttle valve is the pneumatic OR valve.

9. Pressure loss in manifold type valve is more than inline type valves

10. Suspended disc type valve has very long actuation movement

Review Questions

1. List the function of a pneumatic Valve?

2. How can we classify the pneumatic valves?

3. How can we classify Direction control valve(DCV)

3. How do 2/2 way differ from 4/2 way pneumatic Direction control valve

4. What are the advantages of poppet valve over ball valve

5. Explain the working of 3/2 Direction control valve with a neat sketch

6. Explain the working of 5/3 Direction control valve with a neat sketch

7. Mention few applications of 4/3 Direction control valve

8. Differentiate between Rotary valve and spool type valve

9. How do you classify Non return valves?

10. Mention few applications of Non return valves

11. Explain with the help of neat sketch the construction and working of quick exhaust valves

12. How do you classify Pressure control valves?

13. Explain the difference between pressure limiting valve and sequence valve

14. Explain the working of pressure limiting valve

15. Compare and contrast two way valve and shuttle valve. Mention its application.

Answers

Fill in the Blanks

1. direction
2. neutral
3. intermediate
4. Non return valves
5. higher
6. Shorter
7. atmosphere.

8.few
9. pilot
10. core/sleeve
11. Needle
12 Out

State True or False

1. True
2. False
3. False
4. False
5. True
6. True
7. False
8. True
9. False
10. False

Lecture 39

SINGLE ACTUATOR CIRCUITS

Learning Objectives

Upon completion of this chapter, Student should be able to

- Differentiate between pneumatic circuit and pneumatic circuit diagram
- State basic rules used in design of pneumatic circuits
- Explain the memory, delay, OR , AND and NOT functions
- Explain the direct and indirect control of single acting cylinder
- Explain the direct and indirect control of double acting cylinder
- Differentiate supply and exhaust air throttling
- Study various methods of checking end positions of a cylinder
- Design pressure and time dependant circuits

1.1 Pneumatic circuit and pneumatic circuit diagram.

Pneumatic control systems can be designed in the form of pneumatic circuits. A pneumatic circuit is formed by various pneumatic components, such as cylinders, directional control valves, flow control valves, pressure regulator, signal processing elements such as shuttle valve, two pressure valve etc. Pneumatic circuits have the following functions

- To control the entry and exit of compressed air in the cylinders.
- To use one valve to control another valve
- To control actuators or any other pneumatic devices

A pneumatic circuit diagram uses pneumatic symbols to describe its design. Some basic rules must be followed when drawing pneumatic diagrams.

To be able to design pneumatic circuits, it is better for one to have basic knowledge on the designing simple pneumatic circuits. With this foundation, one would be able to move on to the designing more complicated circuits involving many more cylinders.

1.2 SINGLE ACTING CYLINDER CONTROL

1.2.1 DIRECT CONTROL OF SINGLE ACTING CYLINDER.

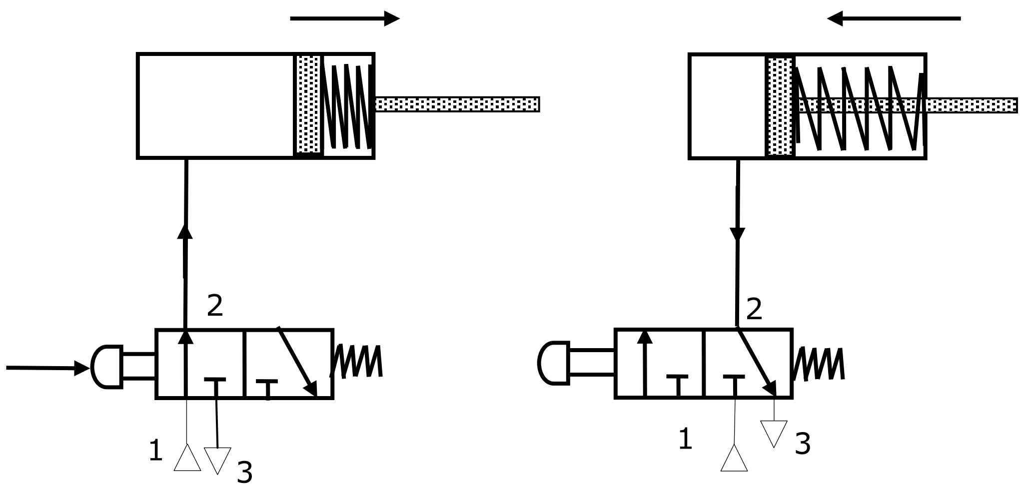


Figure 1.1 Direct control of a single acting cylinder

Pneumatic cylinders can be directly controlled by actuation of final directional control valve (Figure 1.1). These valves can be controlled manually or electrically. This circuit can be used for small cylinders as well as cylinders which operates at low speeds where the flow rate requirements are less. When the directional control valve is actuated by push button, the valve switches over to the open position, communicating working source to the cylinder volume. This results in the forward motion of the piston. When the push button is released, the reset spring of the valve restores the valve to the initial position [closed]. The cylinder space is connected to exhaust port there by piston retracts either due to spring or supply pressure applied from the other port.

Example 1: A small single acting cylinder is to extend and clamp a work piece when a push button is pressed. As long as the push button is activated, the cylinder should remain in the clamped position. If the push button is released, the clamp is to retract. Use additional start button. Schematic diagram of the setup is shown in Figure 1.2

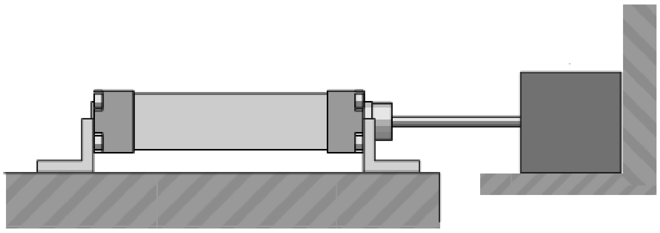


Figure 1.2

Solution

The control valve used for the single acting cylinder is the 3/2 way valve. In this case, since the cylinder is of small capacity, the operation can be directly controlled by a push button 3/2 way directional control valve with spring return.

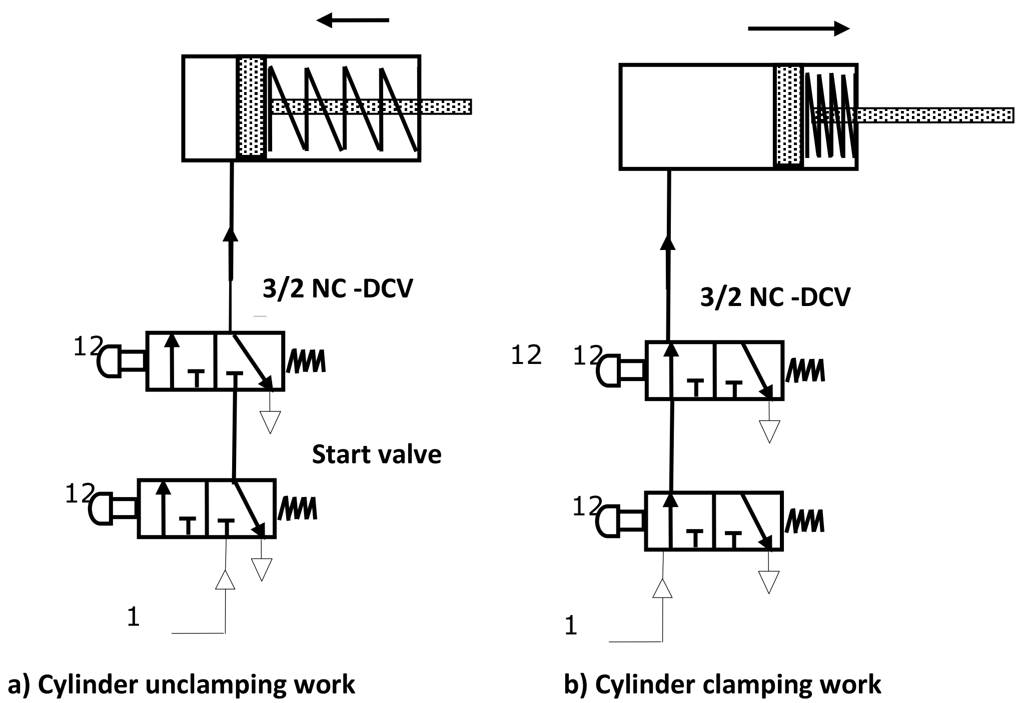


Figure 1.3

When start button and 3/2 NC valve is operated, cylinder moves forward to clamp the work piece. When start button and 3/2 way valve is released cylinder comes back to the retracted position as shown in **Figure 1.3**

1.2.2 INDIRECT CONTROL OF SINGLE ACTING CYLINDER

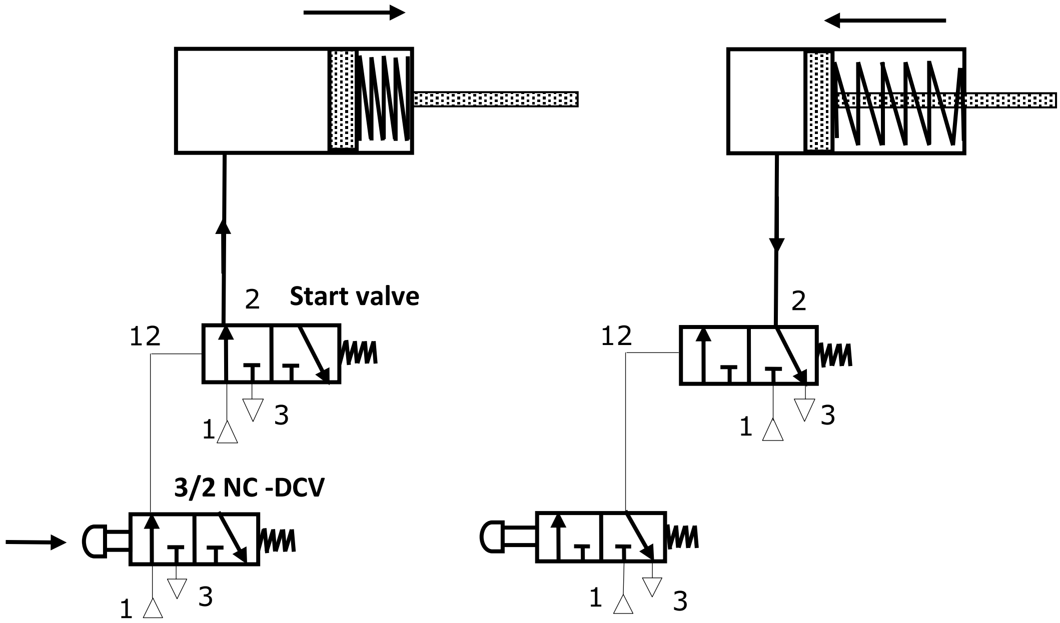


Figure 1.4 Indirect control of a single acting cylinder

This type of circuit (Figure 1.4) is suitable for large single cylinders as well as cylinders operating at high speeds. The final pilot control valve is actuated by normally closed 3/2 push button operated valve. The final control valves handle large quantity of air. When the push button is pressed, 3/2 normally closed valve generate a pilot signal 12 which controls the final valve thereby connecting the working medium to piston side of the cylinder so as to advance the cylinder. When the push button is released, pilot air from final valve is vented to atmosphere through 3/2 NC – DCV.

The signal pressure required can be around 1-1.5 bar. The working pressure passing through the final control valve depends on the force requirement which will be around 4-6 bar. Indirect control as permits processing of input signals. Single piloted valves are rarely used in applications where the piston has to retract immediately on taking out the set pilot signal.

Example 2: A large single acting cylinder is to extend and clamp a work piece when a push button is pressed. As long as the push button is activated, the cylinder should remain in the clamped position. If the push button is released, the clamp is to retract. Use additional start button.

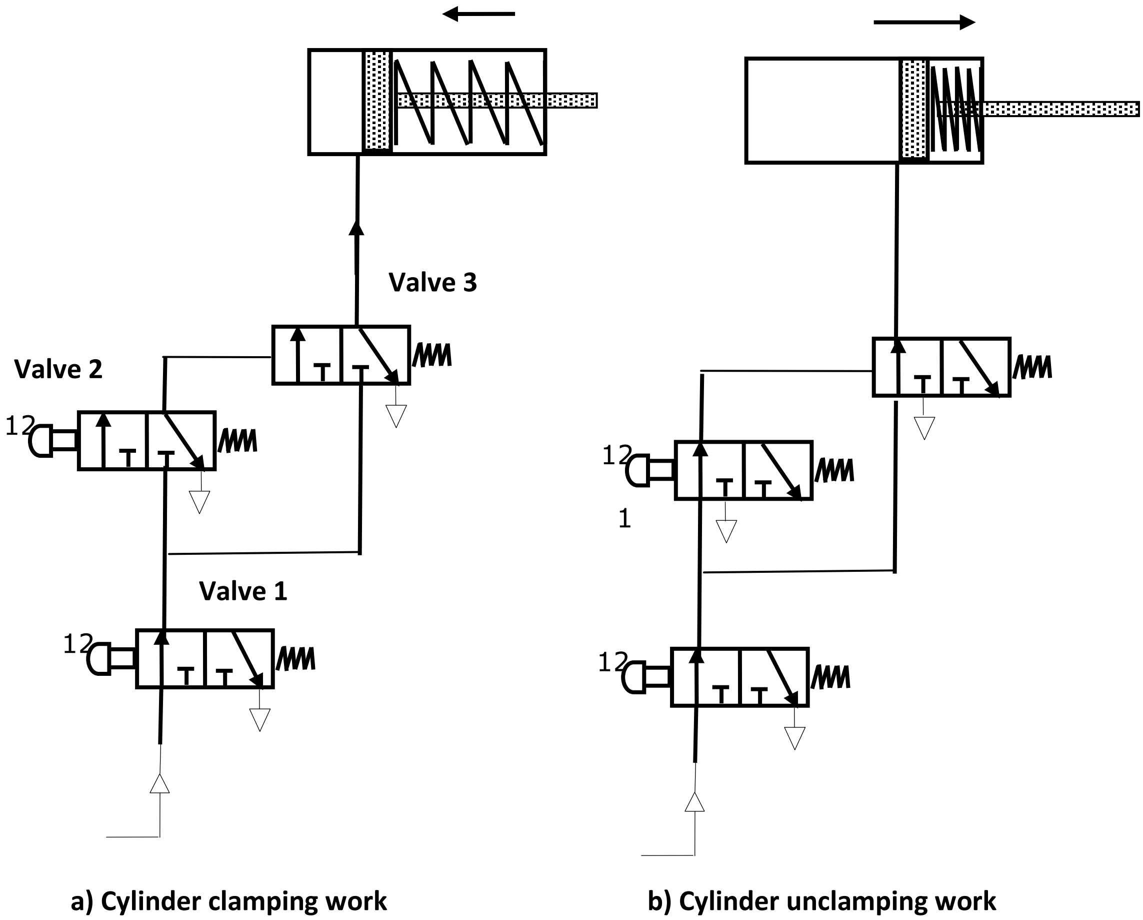


Figure 1.5

The control valve used for the single acting cylinder is the 3/2 way valve. In this case, since the cylinder is of large capacity, the operation cannot be directly controlled by a push button 3/2 way directional control valve with spring return. Indirect control is to be used as shown in the Figure 1.5

Valve 2 is a small capacity valve which controls the large capacity valve 3. When the valve 2 is unactuated the cylinder is in the retracted condition. When the valve 2 is actuated the cylinder is in the extended position to clamp the work piece.

1.2.3 CONTROL OF SINGLE ACTING CYLINDER USING "OR" VALVE

Shuttle valve is also known as double control valve or double check valve. A shuttle valve has two inlets and one outlet (Figure 1.6). At any one time, flow is shut off in the direction of whichever inlet is unloaded and is open from the loaded inlet to the outlet. This valve is also called an OR valve. A shuttle valve may be installed for example, when the cylinder or valve is to be actuated from two points, which may be remote from one another.

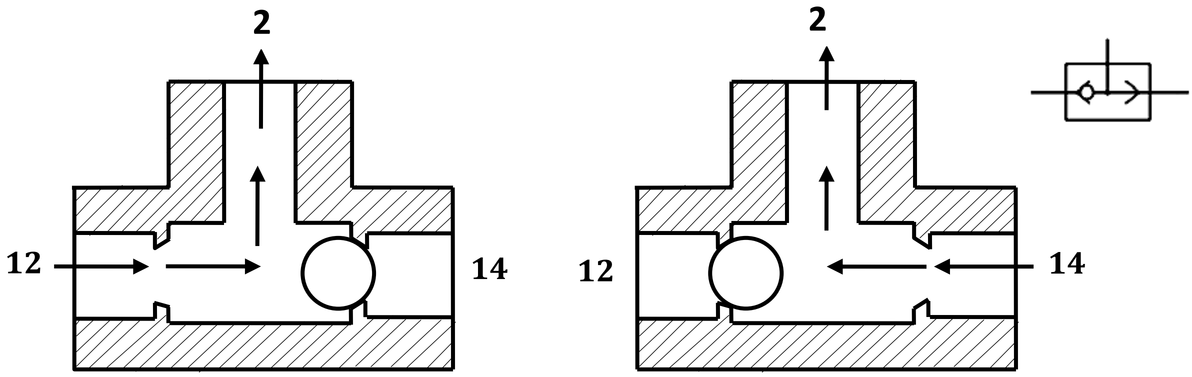


Figure 1.6 Shuttle valve (OR valve)

The single acting cylinder in Figure 1.7 can be operated by two different circuits. Examples include manual operation and relying on automatic circuit signals, that is, when either control valve ① or control valve ② is operated, the cylinder will work. Therefore, the circuit in Figure 1.7 possesses the OR function.

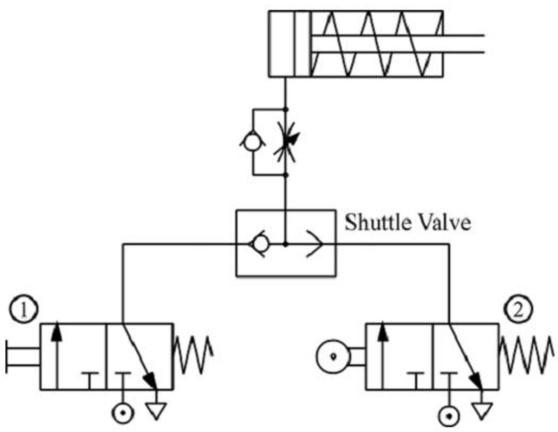


Figure 1.7 Control of a single acting cylinder using OR valve

1.2.4 CONTROL OF SINGLE ACTING CYLINDER USING "AND" VALVE

This valve is the pneumatic AND valve. It is also derivate of Non Return Valve. A two pressure valve requires two pressurised inputs to allow an output from itself. The cross sectional views of two pressure valve in two positions are given in **Figure 1.8** As shown in the **Figure 1.8**, this valve has two inputs 12 and 14 and one output 2. If the compressed air is applied to either 12 or input 14, the spool moves to block the flow, and no signal appears at output 2. If signals are applied to both the inputs 12 and 14, the compressed air flows through the valve, and the signal appears at output 2.

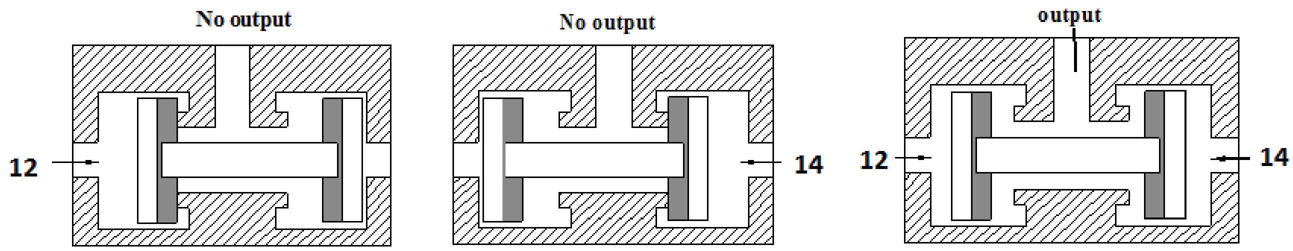


Figure 1.8 control of a single acting cylinder using OR valve

Another name for an AND function is interlock control. This means control is possible only when two conditions are satisfied. A classic example is a pneumatic system that works only when its safety door is closed and its manual control valve is operated. The flow passage will open only when both control valves are operated. **Figure 1.9** shows the circuit diagram of an AND function circuit. The cylinder will work only when both valve ① and ② are operated.

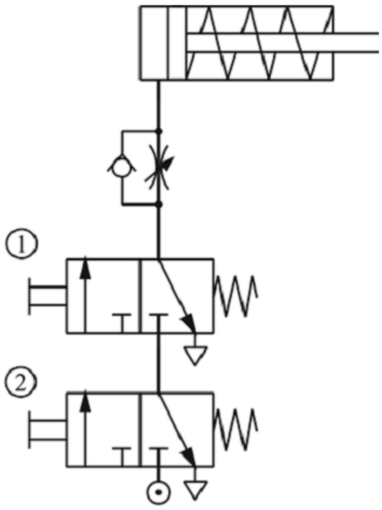


Figure 1.9 Control of a single acting cylinder using AND valve

1.2.5 CONTROL OF SINGLE ACTING CYLINDER USING "NOT" VALVE

Another name for a NOT function is inverse control. In order to hold or lock an operating conveyor or a similar machine, the cylinder must be locked until a signal for cancelling the lock is received. Therefore, the signal for cancelling the lock should be operated by a normally open type control valve. However, to cancel the lock, the same signal must also cancel the locks on other devices, like the indication signal ③. **Figure 1.10** shows how the normally closed type control valve ① can be used to cut off the normally open type control valve ② and achieve the goal of changing the signal.

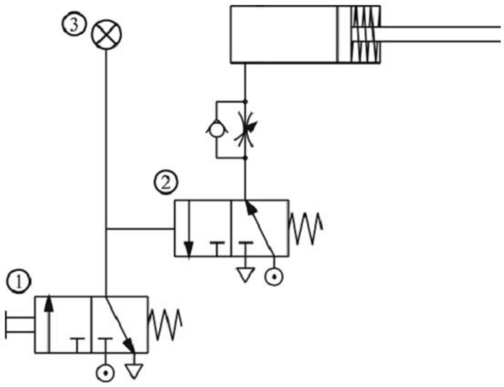


Figure 1.10 Control of a single acting cylinder using NOT valve

1.3 DIRECT CONTROL OF DOUBLE ACTING CYLINDER

The only difference between a single acting cylinder and a double acting cylinder is that a double acting cylinder uses a 5/2 directional control valve instead of a 3/2 directional control valve (**Figure 1.11**). Usually, when a double acting cylinder is not operated, outlet 'B' and inlet 'P' will be connected. In this circuit, whenever the operation button is pushed manually, the double acting cylinder will move back and forth once

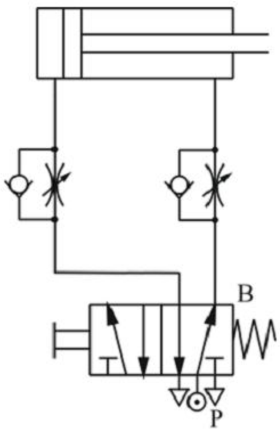


Figure 1.11 Direct control of a double acting cylinder

In order to control the speed in both directions, flow control valves are connected to the inlets on both sides of the cylinder. The direction of the flow control valve is opposite to that of the release of air by the flow control valve of the single acting cylinder. Compared to the throttle inlet, the flow control valve is tougher and more stable. Connecting the circuit in this way allows the input of sufficient air pressure and energy to drive the piston.

Example 3: Pneumatic system is to be designed to operate a door of public transport vehicles. (**Figure 1.12**). Assuming that the opening and closing of the doors are controlled by two button switches ON and OFF. When the button switch ON is pressed, the door will open. When the button switch OFF is pushed, the doors will close.

Solution.

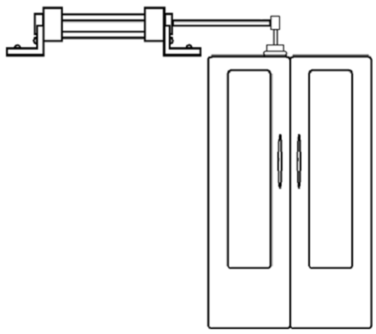


Figure 1.12 Operation of pneumatic system that controls the door of vehicle

Solution

Solution is given in Figure 1.13, which is self explanatory

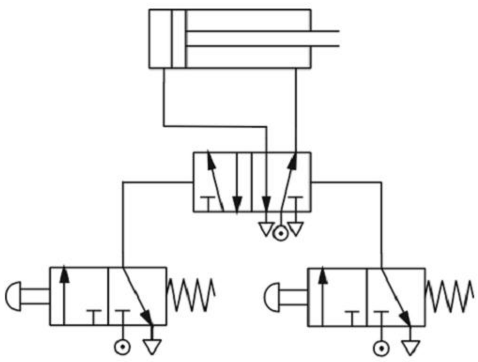
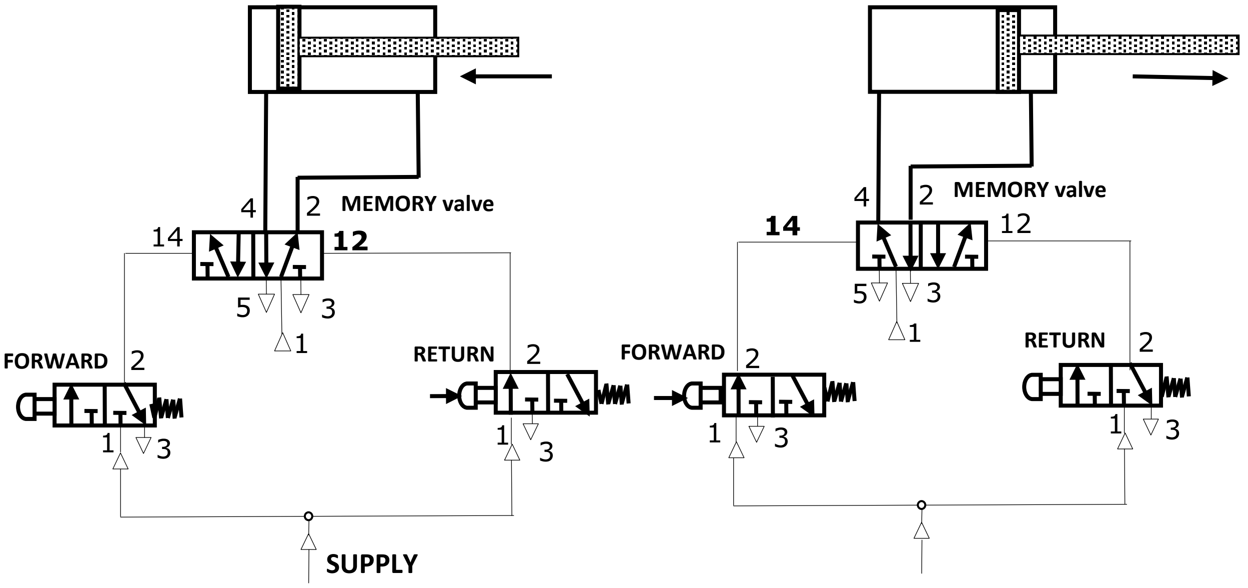


Figure 1.13 Pneumatic circuit to control the door of vehicle

1.3.1 IN DIRECT CONTROL OF DOUBLE ACTING CYLINDER USING MEMORY VALVE



a) Signal from the valve “Return” b)Signal from the valve “Forward”

Figure 1.14 Indirect control of Double acting cylinder using memory valve

When the 3/2 way valve meant for Forward motion (Figure 1.14b) is pressed, the 5/2 memory valve switches over through the signal applied to its pilot port 14. The piston travels out and remains in the forward end position. Double piloted valve is also called as the Memory valve because now even if this push button meant Forward is released the final 5/2 control valve remains in the actuated status as the both the pilot ports of 5/2 valves are exposed to the atmosphere pressure and the piston remains in the forward end position.

When the 3/2 way valve meant for return motion (Figure 1.14a) is pressed, the 5/2 way valve switches back to initial position through the signal applied to its pilot port 12. The piston then returns to its initial position and remains in the rear end position. Now even if the Return push button is released the status of the cylinder will not change.

The circuit is called a memory circuit because it uses a 5/2 way double pilot memory valve. 5/2 way valve can remember the last signal applied in terms of the position of the spool in the absence of reset springs, thus memorising or storing the pneumatic signal. Double piloted 4/2 way valve also can be used as memory valve

1.4 SUPPLY AIR THROTTLING AND EXHAUST AIR THROTTLING

It is always necessary to reduce the speed of cylinder from maximum speed based on selected size of final control valve to the nominal speed depending on the application. Speed control of Pneumatic Cylinders can be conveniently achieved by regulating the flow rate supply or exhaust air. The volume flow rate of air can be controlled by using flow control valves which can be either two way flow control valve or one way flow control valve

There are two types of throttling circuits for double acting cylinders:

i) Supply air throttling

ii) Exhaust air throttling

1.4.1 Supply air throttling.

This method of speed control of double acting cylinders is also called meter –in circuit (Figure 1.15a). For supply air throttling, one way flow control valves are installed so that air entering the cylinder is throttled. The exhaust air can escape freely through the check valve of the throttle valve on the outlet side of the cylinder. There is no air cushion on the exhaust side of the cylinder piston with this throttling arrangement. As a result, considerable differences in stroking velocity may be obtained even with very small variations of load on the piston rod. Any load in the direction of operating motion will accelerate the piston above the set velocity. Therefore supply air throttling can be used for single acting and small volume cylinders.

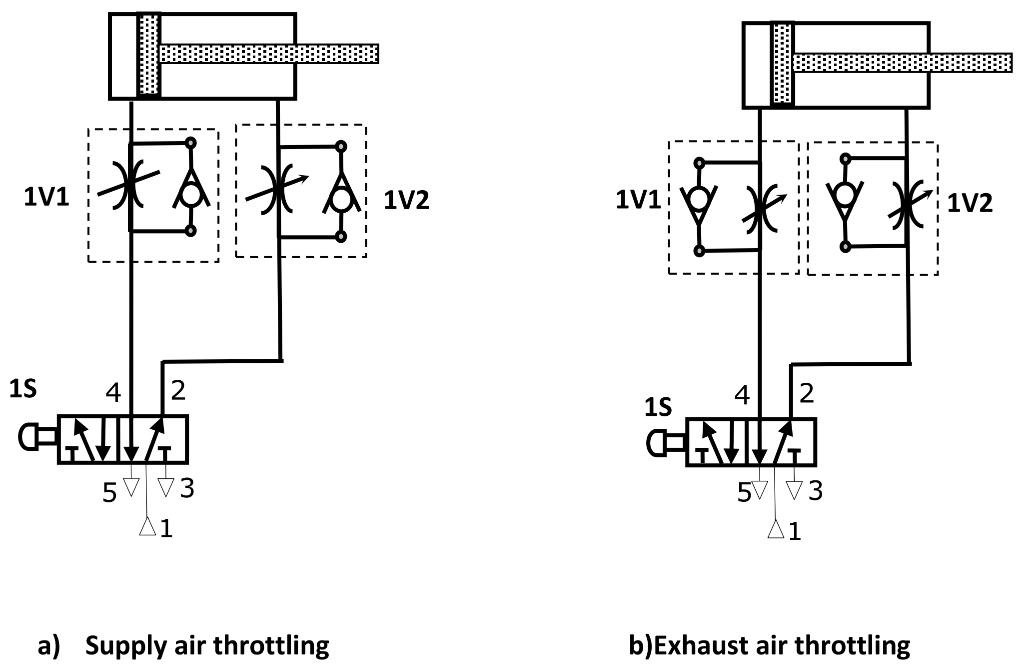


Figure 1.15 Throttling Circuits

1.4.2 Exhaust air throttling.

This method of speed control of double acting cylinders is also called meter-out (Figure 1.15b). In exhaust air throttling throttle relief valves are installed between the cylinder and the main valve in such a way that the exhaust air leaving the cylinder is throttled in both directions of the motion of the cylinder. The supply air can pass freely through the corresponding check valves in each case. In this case, the piston is loaded between two cushions of air while the cylinder is in motion and hence a smooth motion of the cylinder can be obtained. The first cushion effect is due to supply air entering the cylinder through check valve, and second cushion effect is due to the exhaust air leaving the cylinder through the throttle valve at a slower rate. Therefore, exhaust air throttling is practically used for the speed control of double acting cylinders. Arranging throttle valves in this way contributes substantially to the improvement of feed behaviour.

1.5 VARIOUS METHODS OF CHECKING END POSITION OF A CYLINDER

The following methods are commonly used to know the end positions of piston in the cylinder:

a) Mechanically operated limit switches

   i) Roller lever type ii) Idle return roller type

b) Reed sensors – Normally used in cylinder with magnetically coupled slide

   i) With Electrical output ii) With pneumatic output

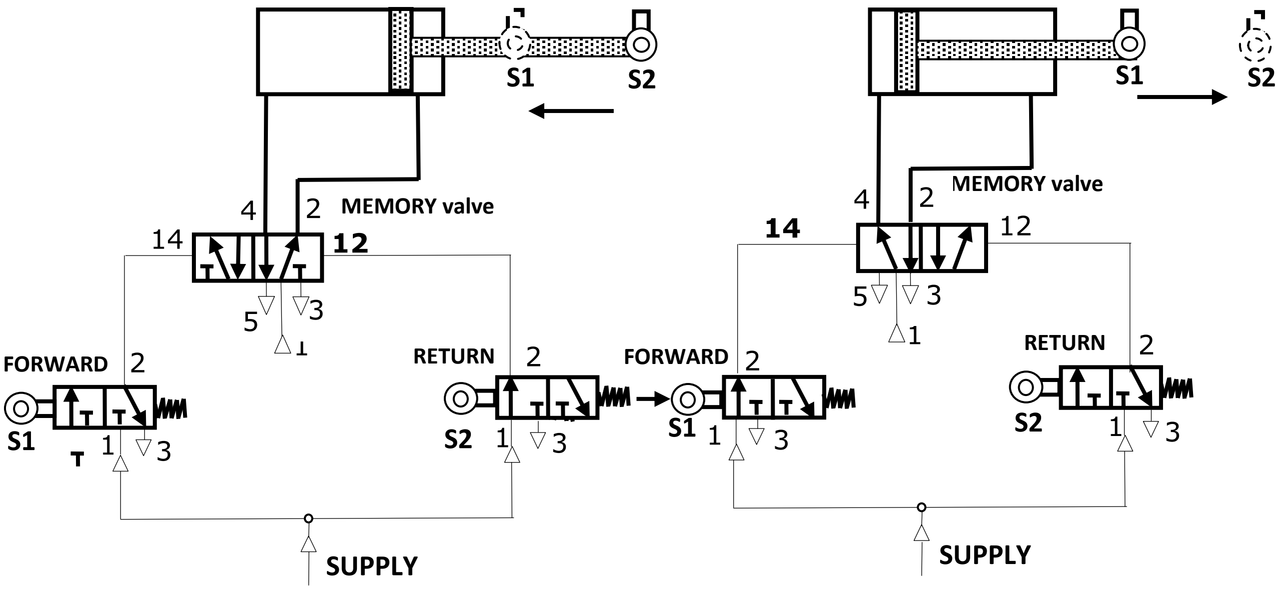
c) Electrical proximity switches

d) Pneumatic Signal generators

In this chapter we shall only discuss use of limit switches to get the end position of piston in the cylinder.

1.5.1 Use of Limit Switches

S1 and S2 are the limit switches corresponding to home position and extended position. Although they are located in the path of the movement of piston rod, normal practice is to represent the symbol of the limit switches on either side of the 3/2 way control valve without put signals connected to the pilot ports of the valve.(in this Figure 1.16 pilot signals actuations are shown for clarity) The limit switches of Roller lever type are essentially 3/2 way ball seat or disc seat type of valves handling pneumatic signals. These are available with direct actuation type and internally pilot actuation type versions. Limit switches of idle return roller type are used for actuation only in one direction are used as signal elimination device in case of signal overlap.



a)Signal from the valve "Return" b) Signal from the valve "Forward"

Figure 1.16 : Use of limit switches in pneumatic circuits

1.6 PRESSURE DEPENDENT CONTROLS

Pressure sequence valve is essentially a switch on or off valve. Sequence Valve generates a pneumatic signal if the sensing pressure [signal input] is more than the desired set pressure. This generated output signal is used to control the movement of cylinder by using it as a set signal or reset signal to the final control valve to obtain forward or return motion respectively. Used for applications such as bonding cylinders, clamping cylinder etc. to ensure desired minimum pressure in the cylinder. This is a combination valve, having two sections. One of the section is a 3/2 directional control and the other a pressure control valve. Schematic diagram of pressure sequence vavle is shown in Figure 1.17

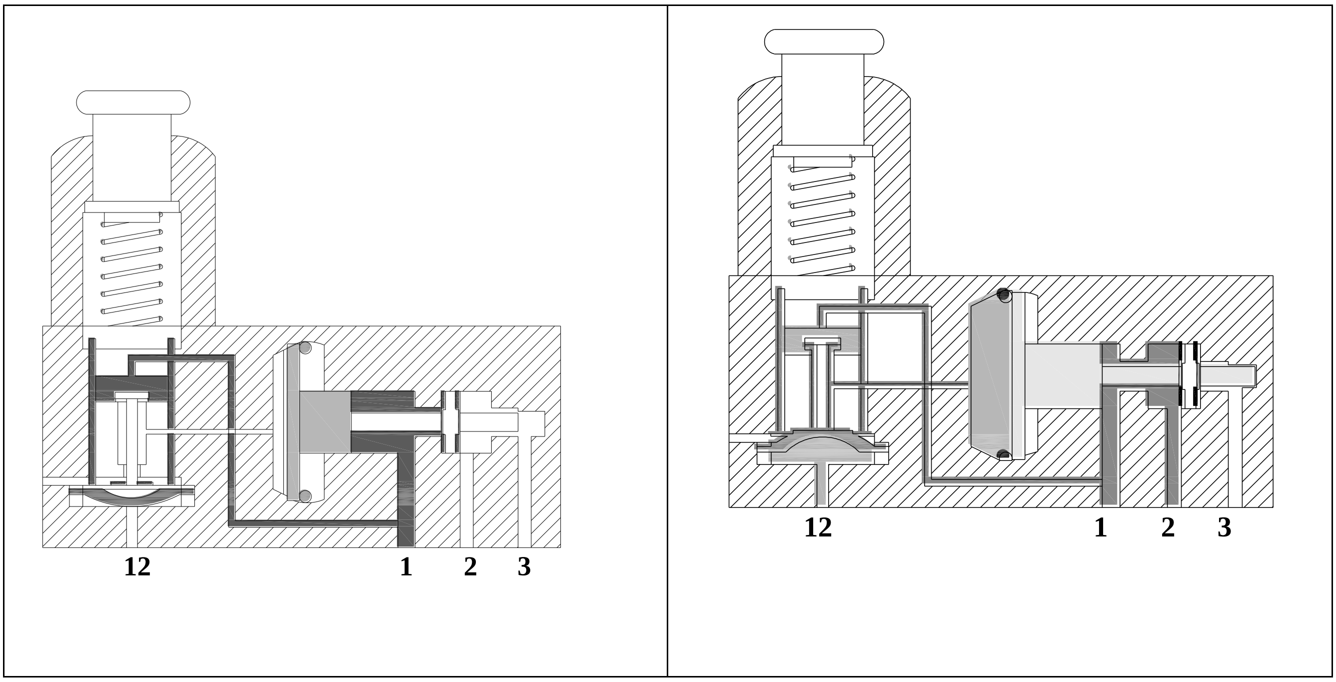


Figure 1.17: Pressure dependant control valve

Sensing pressure signal is introduced at port 12. Manual adjustment of pressure setting is done with the help of a cap screw/knob which is spring loaded. Clock wise rotation of knob results setting for higher pressure setting and anticlockwise rotation of knob results in lower pressure setting. The right section is basically a 3/2 directional control valve [NC] - pilot operated using pressure signal derived from left section.

The cross sectional views and symbols of a pressure sequence valve is shown in Figure. This valve consists of a 3/2 directional control valve and a pressure control valve. The principle of operation of a sequence valve is that outlet of sequence valve remains closed until pressure upstream of it builds up to a predetermined value. When a pre-set pilot pressure is reached within the valve at port 12(due to the building up of the line pressure after the cylinder piston reaches the end of its stroke), the spring loaded piston is unseated. The resulting pilot pressure actuates the integrated 3/2 directional control valve of the pressure sequence valve and generates an output signal which connects port 1 to 2. Note that when the piston is seated, the piston area in the 3/2 directional valve, that is exposed to the pilot pressure, is held to a minimum. This helps reduce the restraining force. When the pressure, however, unseats the piston, a large piston area is exposed to pressure, and the piston is held wide open. The adjusting screw on the top of the valve sets the pressure.

Example 11: A double acting cylinder is used to press together glued components (Estimated pressure is around 4 bar). Upon pressing a push button, the clamping cylinder is to extend and trip the roller valve. One the fully extended position of the cylinder has been reached and sufficient clamping force has been developed, the cylinder is to retract to the initial position, develop a pneumatic circuit using a pressure sequence valve.

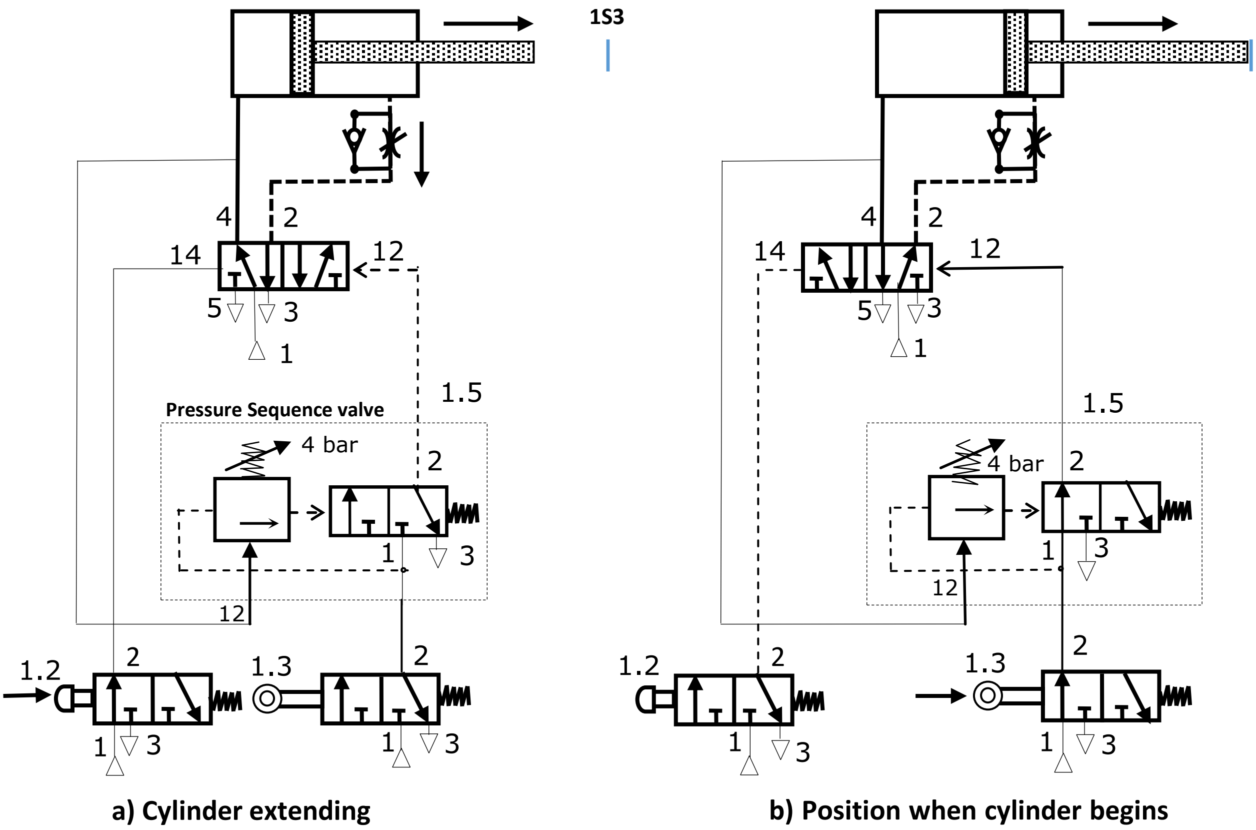


Figure 1.18 Pressure dependant control valve

The two position of the pneumatic circuit for the control task, when the cylinder is extending and when the cylinder is fully extended, are shown in Figure 1.18. The pressure in the sequence valve is set to working pressure of 4 bar, and the signal input to the pressure sequence valve is tapped from the power line from port 4 of valve 1.1 to the cylinder to gauge the pressure on the piston side of the cylinder. As shown in the Figure 1.18 valve 1.2 initiates the forward motion of the cylinder. While the cylinder is moving forward, the pressure in the power line from port 4 of valve 1.1 to the cylinder will not built up to the working pressure. Only after the cylinder is fully extended, will the maximum pressure in the line built up resulting in sufficient pressure to glue. When the set pressure in the sequence valve is reached, the integrated 3/2 directional control valve is actuated, generating an output signal. This signal is used to reset the final control element 1.1 and thus causing the return motion of the cylinder.

1.7 TIME DEPENDENT CONTROLS

Pneumatic timers are used to create time delay of signals in pilot operated circuits. Available as normally closed timers and normally open timers. Usually pneumatic timers are on delay timers. Delay of signals is very commonly experienced in applications such as bonding of two pieces. Normally open pneumatic timers are also used in signal elimination. Normally open pneumatic timers are used as safety device in two hand blocks

Time delay valve is a combination of a pneumatically actuated 3/2 direction control valve, an air reservoir and a throttle relief valve. The time delay function is obtained by controlling the air flow rate to or from the reservoir by using the throttle valve. Adjustment of throttle valve permits fine control of time delay between minimum and maximum times. In pneumatic time delay valves, typical time delays in the range 5-30 seconds are possible. The time delay can be extended with the addition of external reservoir.

Pneumatic timer can be classified as

1. On –delay timer
2. Off – delay timer

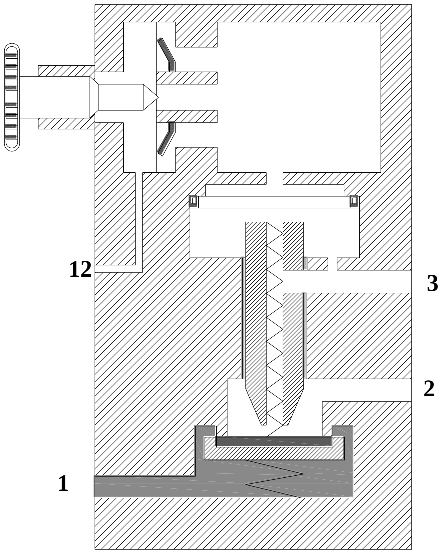
In on-delay timer, the 3/2 DCV is actuated after a delay with reference to the application of pilot signal and is rest immediately on the application of the pilot signal. In off delay timer, the 3/2 DCV is actuated immediately on the application of the pilot signal and is reset only after a delay with reference to the release of the pilot signal.

Pneumatic timers can also be classified according to type of pneumatically actuated 3/2 DCv as:

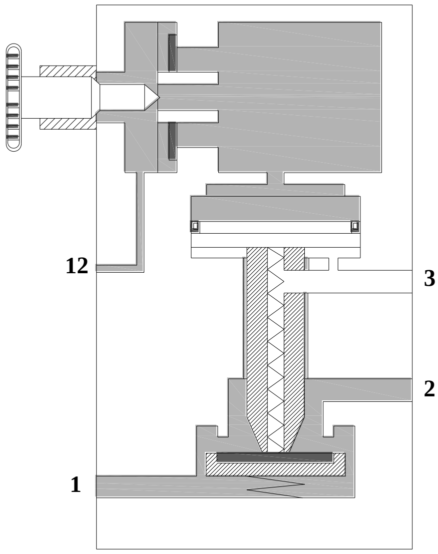
1) Time delay valve, NC type
2) Time delay valve, NO type.

Time delay valve, NC type. The constructions of an on-delay timer (NC) type in the normal and actuated are shown in Figure 1.19 It can be seen that 3/2 DCV operates in the on delay mode permanently. But, in some designs, the valve can be operated in the off-delay mode by connecting the check valve in reverse direction. For this purpose, the ports of the throttle check valve should be brought out.

Time delay valve, NO type. The construction and function of an on-delay timer (NO) type is similar to that of an on-delay timer (NC) type except for the type of 3/2 DCV valve. In the on-delay valve (NO) type, a 3/2 DCV (NO) type is used whereas in the on-delay timer (NC) type, a 3/2 DCV (NC) type is used. Timing diagrams for all four type of Pneumatic delay valve is given in Table 1.1



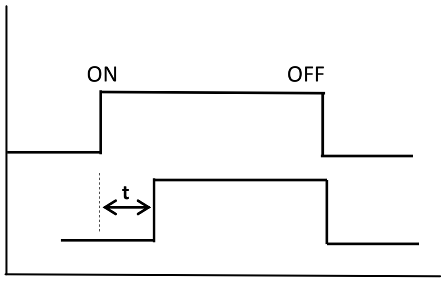
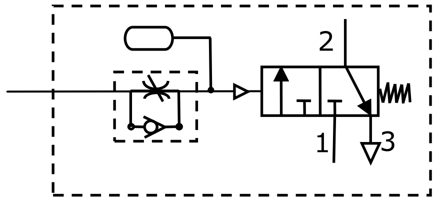
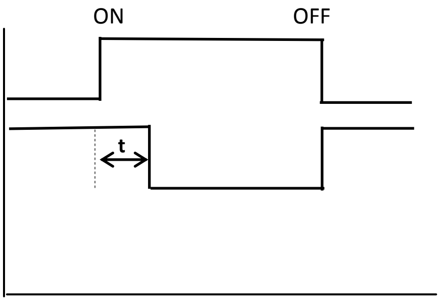
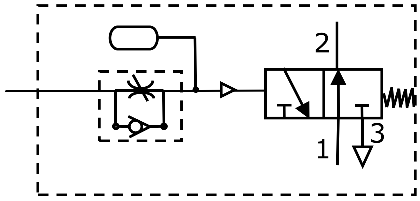
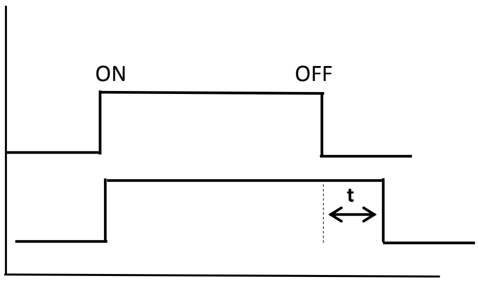
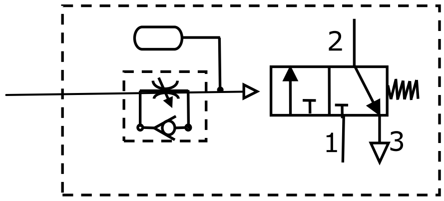
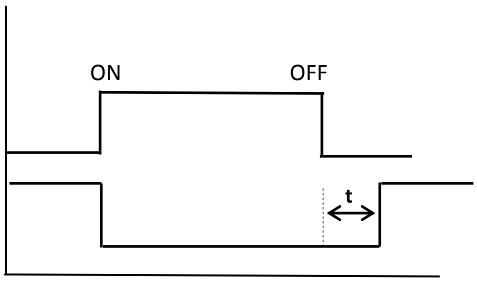
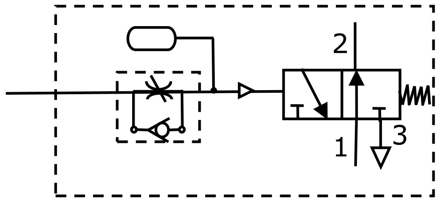
(a) Normal Position



(b) Actuated Position

Figure 1.19

Table 1.1 Timing diagrams for all four type of Pneumatic delay valve

Timing diagram	Symbol
	
	
	
	

Example 12: A double acting cylinder is used to press together glued components . Upon operation of a press button, the clamping cylinder slowly advances. Once the fully extended position is reached , the cylinder is to remain for a time of t = 6 seconds and then immediately retract to the initial position. A new start cycle is only possible after the cylinder has fully retracted and after a delay of 5 seconds. During this delay the finished part is manually removed and replaced with new parts for gluing. The retracting speed should be fast, but adjustable.

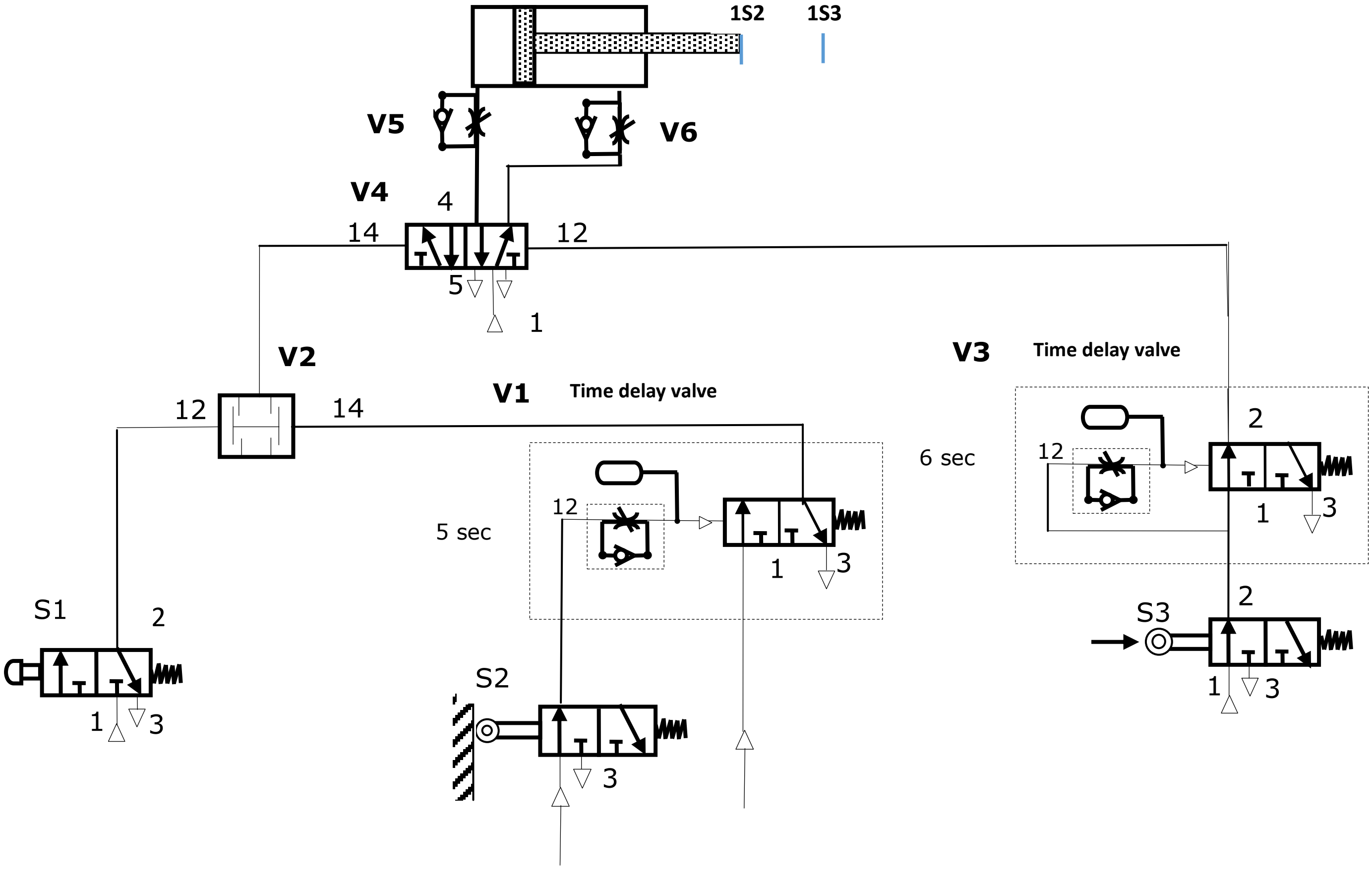


Figure 1.20

If the push button S1 is actuated for a sufficiently long time period (t = 5 second) then the air reservoir of time delay valve V1 is filled and corresponding 3/2 valve is switched, following which a signal is applied at input 1 of the dual pressure valve V2.

If the push button S1 is actuated, the AND condition at the dual pressure valve is met. A signal is applied at the control port 12 of the control element V4. The valve V4 switches, pressure is applied to the piston side of the cylinder 1A and the piston rod advances. After as short advancing distances, the limit switch S2 is released, pressure is reduced in the air reservoir of the time delay valve V1 via the roller lever valve S2, and the integrated 3/2 way valve switches back to its initial position. The

AND condition at the dual pressure valve is now no longer met. Actuation of the push button S1 becomes ineffective.

Upon reaching the advancing position, the piston rod actuates the roller lever S3. The pressure line to the time delay valve V3 is now released and pressure in the air reservoir is increased. The rate of pressure increase is adjustable via the integrated flow control valve. When the switching pressure has been reached, the integrated 3/2 way valve switches and a signal is applied at the control port 12 of the final control element V4. The valve V4 reverses and the piston rod retracts. Upon release of the limit switch S3, the time delay valve V3 Switches to its initial position again.

The limit switch S2 is actuated, when the piston rod reaches its initial position, the pressure in the air reservoir of the time delay V1 starts to increase until the switching pressure has been reached after t = 5 seconds. The integrated 3/2 way valve switches. The initial status of the system is now reached again and a new cycle can be started. The piston rod speed is set at the restrictors of the one way flow control valves V5 and V6.

Objective Type Questions

1. For relatively small single acting and double acting cylinder ------- type of control is preferred.

2. A single acting pneumatic cylinder or a non-reversing air motor can be controlled by ---- way valve as the signal output unit. When a four way control valve is used one outlet port is -----------

3. Double acting cylinders can be controlled by two –three way valve or ....way valve.

4. Throttle relief valve or throttle check valve is used to control the ---------------- in pneumatic circuits.

5. In supply air throttling, air ----------------- the cylinder is throttled

6. Shuttle valve or double check valve can be used as logic ----- Valve.

1. In ------------ pneumatic timer, the 3/2 DC valve is actuated after a delay with reference to the application of the pilot signal and is reset immediately after the release of the pilot signal.

8. The pressure sequence valve is a combination of two valves, that is adjustable pressure ---------- valve and a 3/2 direction control valve.

9. As per ISO 5599 pilot lines ports are designated using 12 or -------------

10. The most important threads used on pneumatic valves are metric thread(M), National pipe thread (NPT) , Unified fine thread(UNF) and -------- (BSP~G)

State True or False

1. For large single acting or double acting cylinder we prefer to use direct control

2. Supply air throttling is used only for the speed control of single acting cylinders and small volume cylinders.

3 Exhaust air throttling is practically used in all double acting cylinder due to double cushioning effect.

4. Double piloted 4/2 and 5/2 valves can be used as memory valves in pneumatic circuit design

5. There is no possibility of signal conflict in memory valves.

6. Two pressure valve can be used as not valve in pneumatic circuit design

7. Return speed of the double acting cylinder can be increased by means of quick exhaust valve.

8. In off time delay timer, the 3/2 DC valve is actuated immediately on the application of the pilot signal and is reset only after a delay with reference to the release of the pilot signal.

9. There are four possible combinations of pneumatic delay timer using NO and NC control.

10. ISO 5599 systems recommend alpha numeric port designation for direction control valves

Review Questions

1. Where are three way and four way valves used?

2. List the basic five rules that are important in design of pneumatic circuits.

3. State three types of signal processing elements used in pneumatic control

4. Explain the working of NOT valve, give one application of it.

5. Mention few applications of AND and OR valve

6. What are the functions of pneumatic timer delay valves

7. How are timer delay valves classified

8. Differentiate between ON time delay and OFF time delay with help of symbols

9. Explain with the help of neat sketch the construction and working of pressure sequence valve

10. Give two applications of pressure sequence valve and time delay valve.

11. Compare and contrast two way valve and shuttle valve. Mention its application.

12 Differentiate between supply air throttling and exhaust air throttling

13. Why exhaust air throttling is practically used for speed control of the double acting cylinder.

14. What is the advantage of internal pilot valve in pneumatic valves.

15. State the difference between the final control element and signal element

Answers

Fill in the Blanks

1. direct
2. three / plugged
3. four/five
4. flow(speed)
5. entering
6. OR
7. On time delay
8. Relief
9. 14
10. British standard pipe thread (BSP~G)

State True or False

1. False
2. True
3. True
4. True
5. False
6. False
7. True
8. True
9. True
10. False

Lecture 40

MULTI ACTUATOR CIRCUITS

Learning Objectives

Upon completion of this chapter, Student should be able to

- Differentiate between single actuator and multi actuator circuit
- List the various methods available for pneumatic circuit design
- Explain the signal conflict in double piloted memory valve
- Draw motion and displacement diagram for multi cylinders
- List and explain various signal elimination methods
- Explain the working of idle roller valves and its use
- Design multi cylinder circuits using cascade method and step counter method

1.1 SINGLE ACTUATOR CIRCUIT VERSUS MULTI ACTUATOR CIRCUITS

In the previous chapter, we have learnt about the various means and ways to control a single actuator circuits, both for single acting and double acting cylinders. Implementation of logic gates along with use of pressure sequence valve and time delay was systematically presented.

Most of the practical pneumatic circuits use multi cylinders. They are operated in specific sequence to carry out the desired task. For example, to drill a wooden component first we need to clamp and then drill. We can only unclamp the cylinder, if and only if the drill is withdrawn away from the workpice. Here sequencing of movement of clamp cylinders and cylinder which carries the drill is important. This sequencing is carried out by actuation of appropriate final control valves like directional control valves. The position of the cylinders is sensed by the sensors like limit switches, roller or cam operated valves.

Multi cylinder pneumatics circuits can be designed in various methods. There is no universal circuit design method that suits all types of circuits. Some methods are commonly used for compound circuits but would be too expensive for simple circuits. There are five common methods used by engineering and they are given below

- Classic method or Intuitive method
- Cascade method
- Step counter method
- Karnaugh–veitch method
- Combinational circuit design

In this chapter Classic method, cascade method and step counter methods are discussed. Chapter 8 deals with Boolean algebra, KV mapping method and combinational circuit methods. Double piloted 4/2 and 5/2 directional control valves are susceptible to signal conflicts. Cascading and Step counter method are more systematic methods than intuitive methods. Signal conflict can be eliminated by using cascade and stepper counter method.

1.2 CLASSIC METHOD OR INTUITIVE METHOD

In intuitive method, circuit design is done by use of general knowledge of pneumatics following the sequence through intuitively. In general, steps involves

- Write down sequence and draw motion diagrams
- Draw in cylinders and control valves
- Complete circuits intuitively.

1.2.1 Coordinated and sequential motion control.

In majority of the pneumatic applications more than one cylinder is used. The movement of these cylinders are coordinated as per the required sequence. Sensors are used for confirming the cylinder position and the resultant actuation of the final control element. Normally limit switches are used. The activation of limit switches of different cylinders will provide set or reset signal to the final control valves for further controlling the movement of various cylinders. The limit switches have to be arranged in the proper location with the help of motion diagram

1.2.2 Demonstration of Classic method

In order to develop control circuitry for multi cylinder applications, it is necessary to draw the motion diagram to understand the sequence of actuation of various signal input switches-limit switches and sensors. Motion diagram represents status of cylinder position -whether extended or retracted in a particular step

Example 1 : In a press shop, stamping operation to be performed using a stamping machine. Before stamping, workpice has to be clamped under stamping station. Then stamping tool comes and performs stamping operation. Work piece must be unclamped only after stamping operation.

Step 1: Write the statement of the problem:

Let A be the clamping cylinder and B be the stamping cylinder as shown in the Figure xxx. First cylinder A extends and brings under stamping station where cylinder B is located. Cylinder B then extends and stamps the job. Cylinder A can return back only cylinder B has retracted fully.

Step 2: Draw the positional layout. (Figure 1.1)

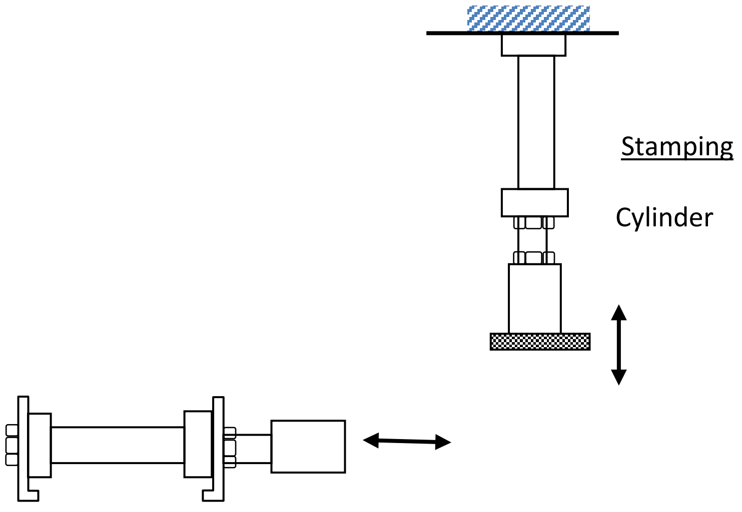


Figure 1.1 Positional layout

Step3: Represent the control task using notational form

Cylinder **A** advancing step is designated as **A+**
Cylinder **A** retracting step is designated as **A-**
Cylinder **B** advancing step is designated as **B+**
Cylinder **B** retracting step is designated as **B-**

Therefore, given sequence for clamping and stamping is A+B+B-A-

Step 4 Draw the Displacement –step diagram (Figure 1.2)

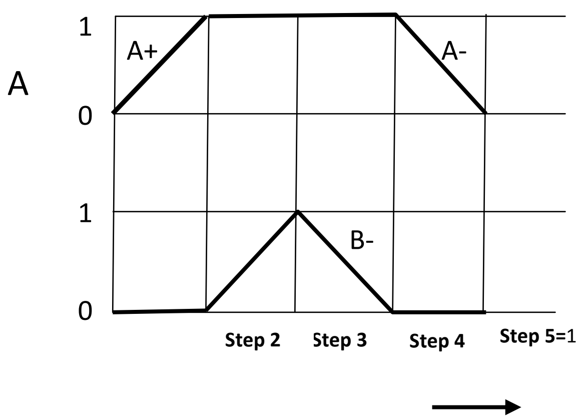


Figure 1.2 Displacement step diagram

Step 5 Draw the Displacement –time diagram (Figure 1.3)

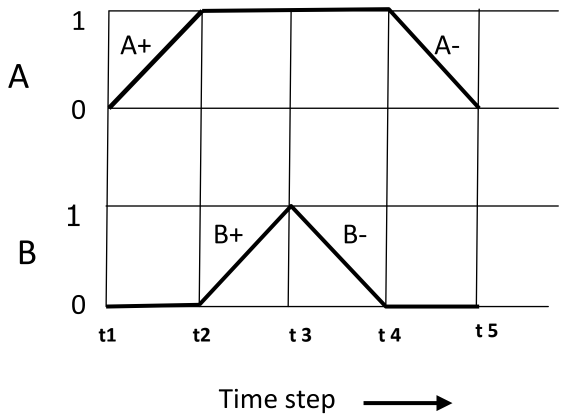


Figure 1.3 Displacement time diagram

Step 6: Analyse and Draw Pneumatic circuit.

Step 6.1 Analyse input and output signals.

Input Signals

Cylinder A – Limit switch at home position ao
Limit switch at home position a1
Cylinder B - Limit switch at home position bo
Limit switch at home position b1

Output Signal

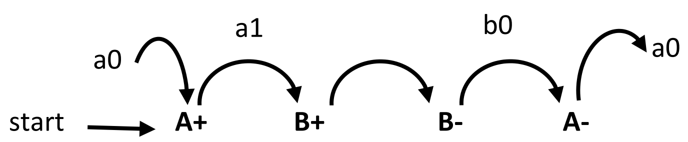
Forward motion of cylinder A (A+)
Return motion of cylinder A (A-)
Forward motion of cylinder B(B+)
Return motion of cylinder B(B+)

Step 6.2 Using the displacement time/step diagram link input signal and output signal.(Figure 1.4)

Usually start signal is also required along with a0 signal for obtaining A+ motion.

1. A+ action generates sensor signal a1, which is used for B+ motion
2. B+ action generates sensor signal b1, which is used for B- motion
3. B- action generates sensor signal b0, which is used for A- motion
4. A- action generates sensor signal a0, which is used for A+ motion

Above information (given in 6.2) is shown below graphically



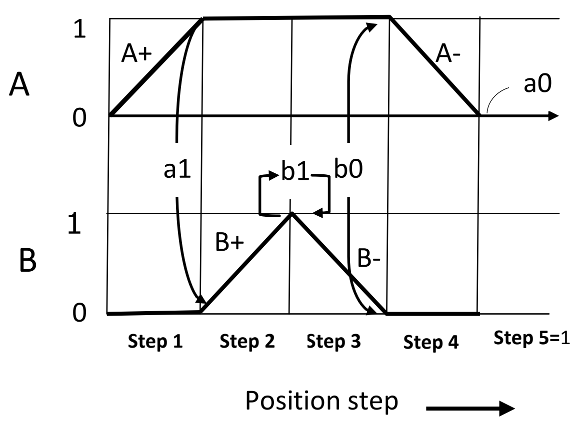


Figure 1.4 Input/output signal flow

Step 7 Draw the power circuit (Figure 1.5)

i) Draw the cylinders A(1.0) and B(2.0).

ii) Draw the DCVs 1.1 and 2.1 in unactuated conditions

iii) Mark the limit switch positions for cylinders A(1.0) and B(2.0).

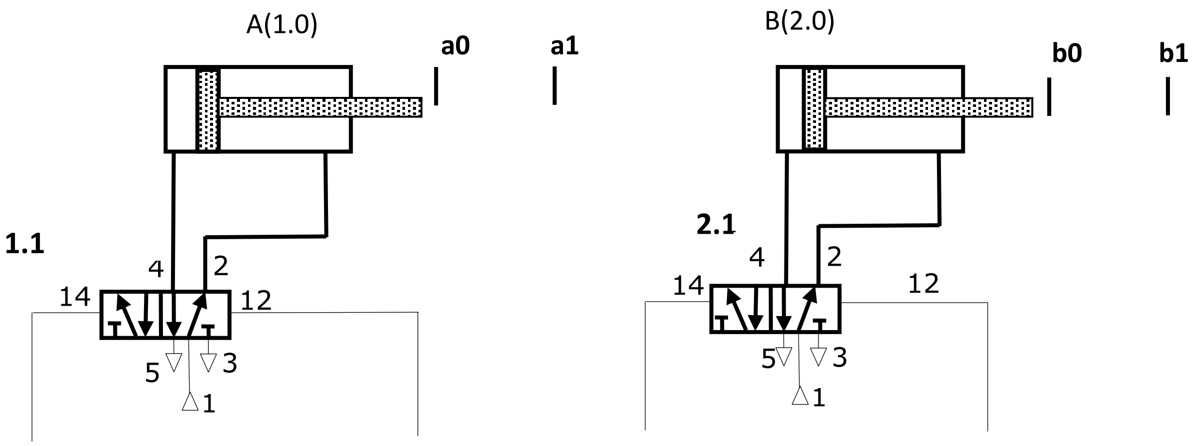


Figure 1.5 Power circuit diagram

Step 8 Draw the control circuit (Figure 1.6)

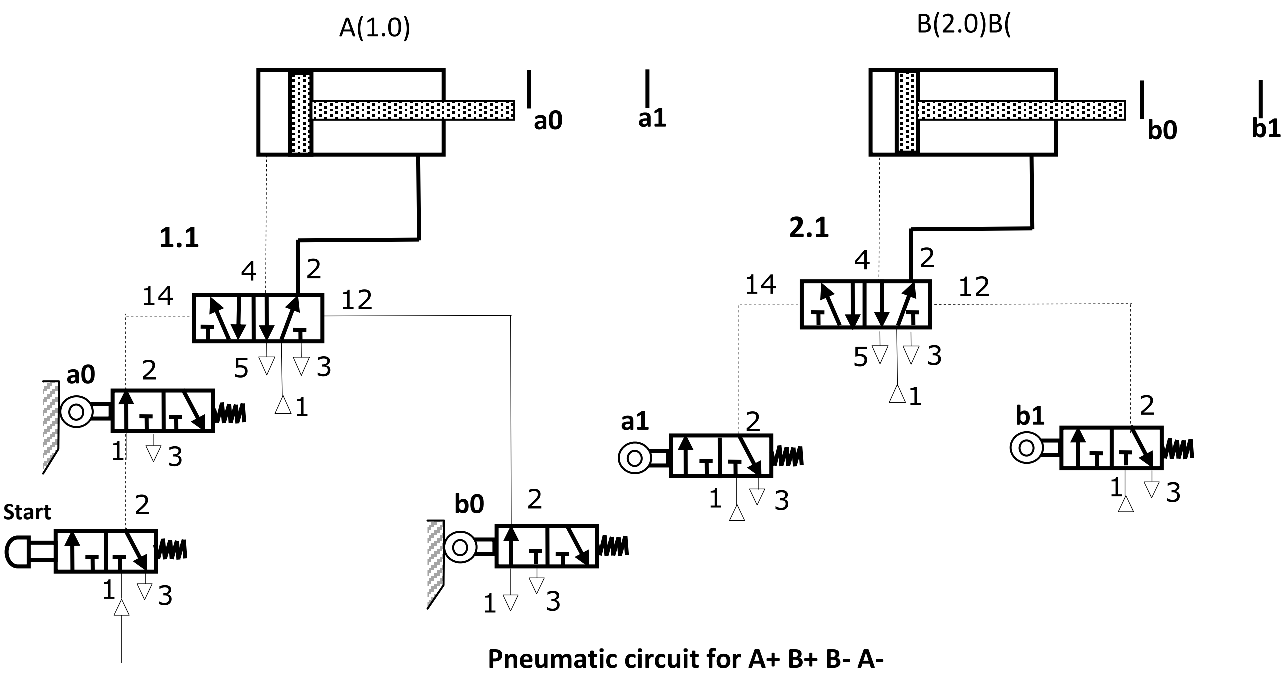


Figure 1.6 Control circuit diagram

Step 9 Analysis of pneumatic circuit

1.When the start button is pressed, the signal appears at port 14 of valve 1.1 through limit switch signal a0.

2. Check for the presence of the signal at the other end (12) of valve 1.1. Notice that the signal is also present at port 12 of valve 1.1. (because bo is also pressed). This results in signal conflict and valve 1.1 is unable to move.(Figure 1.7)

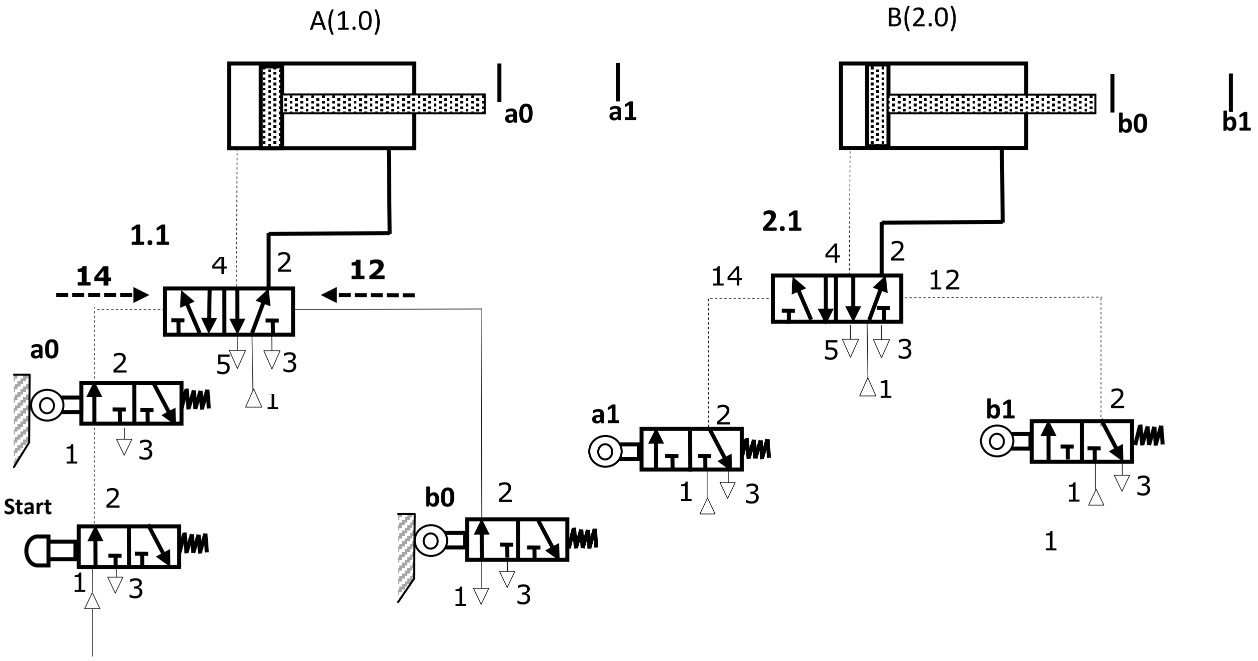


Figure 1.7 Signal conflict at valve 1.1

3. Let us assume for time being, bo is somehow disengaged so that valve 1.1 can switch over and consequently cylinder A can extend. When the start button is pressed. (Figure 1.8)

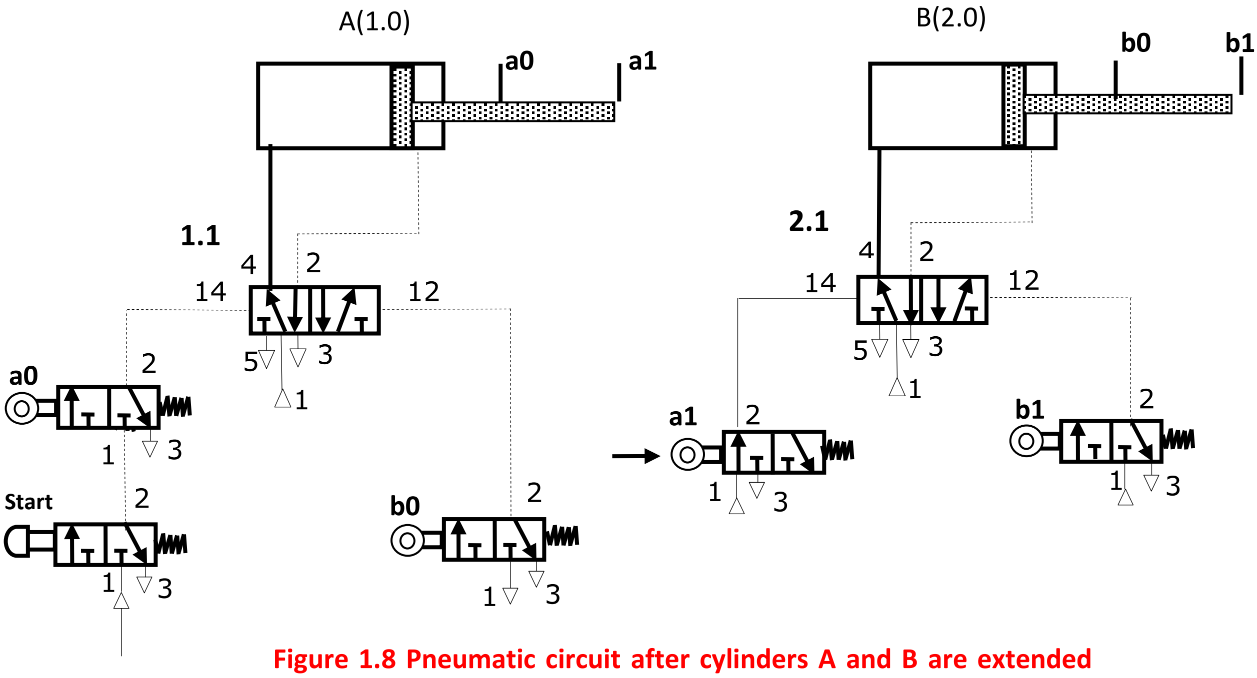


Figure 1.8 Pneumatic circuit after cylinders A and B are extended

4. When cylinder A fully extends, it generates a limit switch signal a1, which is applied to port 14 of the valve 2.1.

5. Check for the presence of the signal at the other end (12) of valve 2.1. Signal is not present at port 12 of valve 2.1 and hence there is no signal conflict

6. Signal applied to port 14 of the valve 2.1 causes the shifting of DCV 2.1 and cylinder B extends.

7. When cylinder B fully extends, it generates a limit switch signal b1, which is applied to port 12 of valve 2.1

1. Check for the presence of the signal at the other end of 14 of valve 2.1. It can be seen that signal is also present at the port 14 of valve 2.1(because a1 is also pressed). This results in signal conflict and valve 2.1 is unable to move (Figure 1.9)

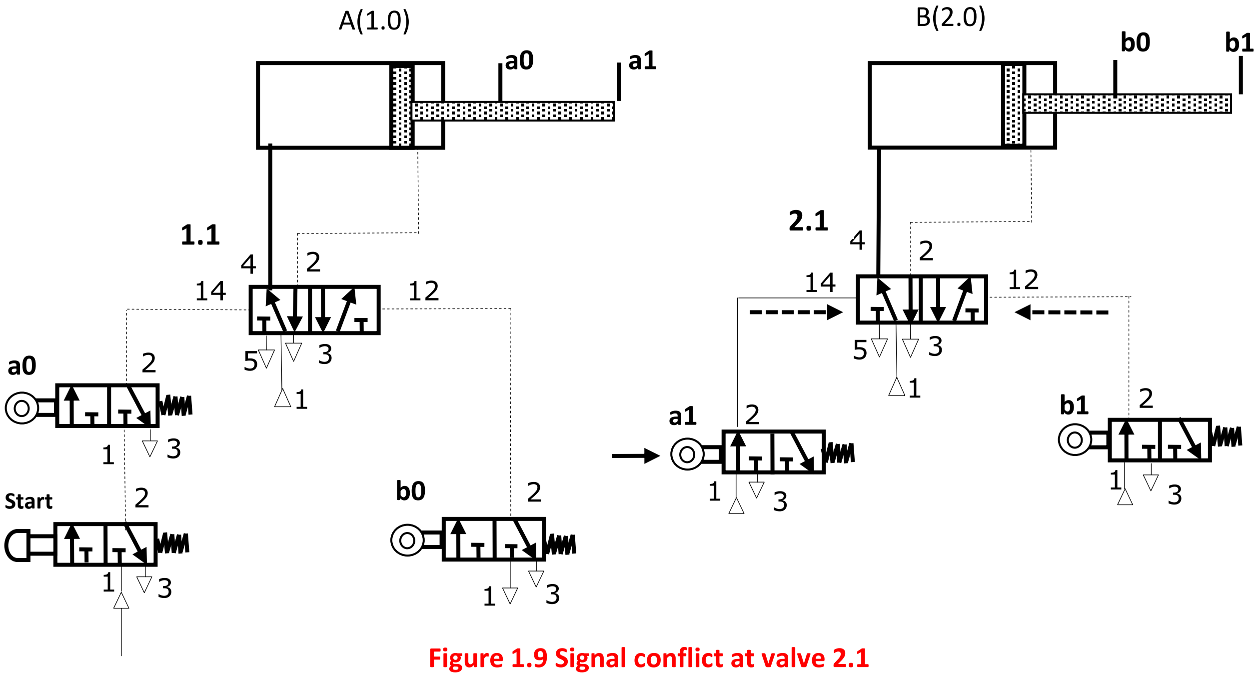


Figure 1.9 Signal conflict at valve 2.1

9. Let us assume for time being, b1 is somehow disengaged so that valve 2.1 can switch over and consequently cylinder B can retract.(Figure 1.10)

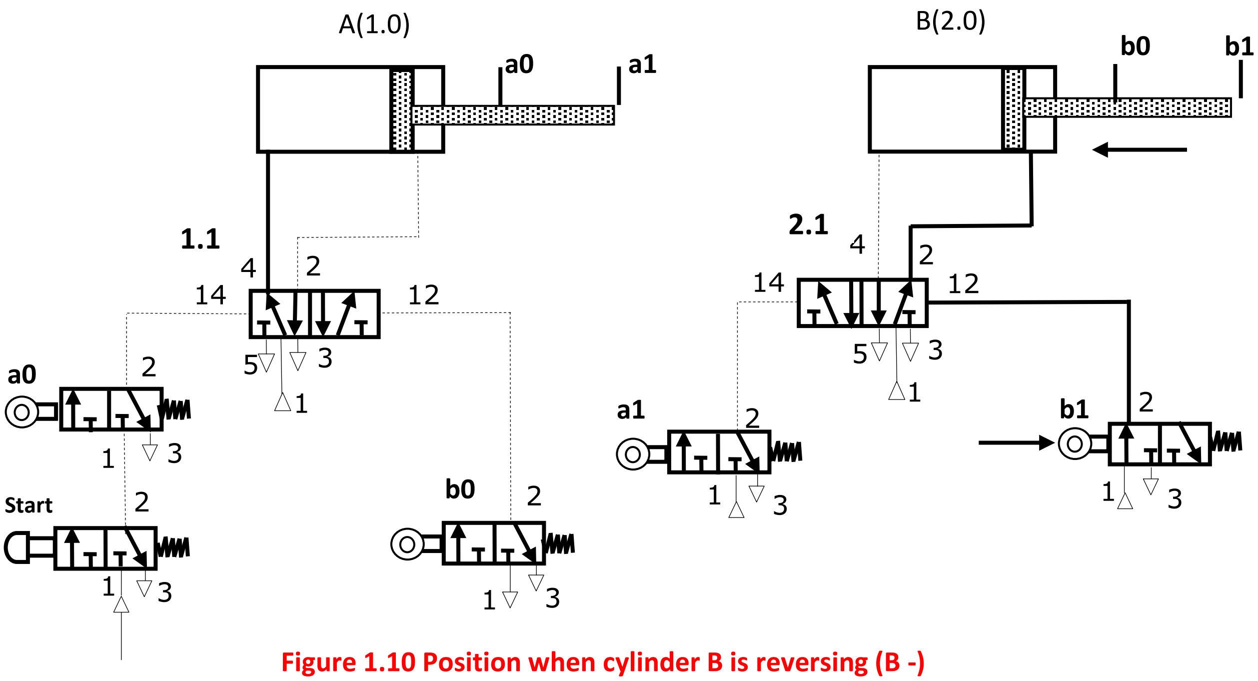


Figure 1.10 Position when cylinder B is reversing (B -)

10. When the cylinder B is fully retracted, it generates a limit switch signal b0, which is applied to port 12 of the valve 1.1. (Figure 1.11)

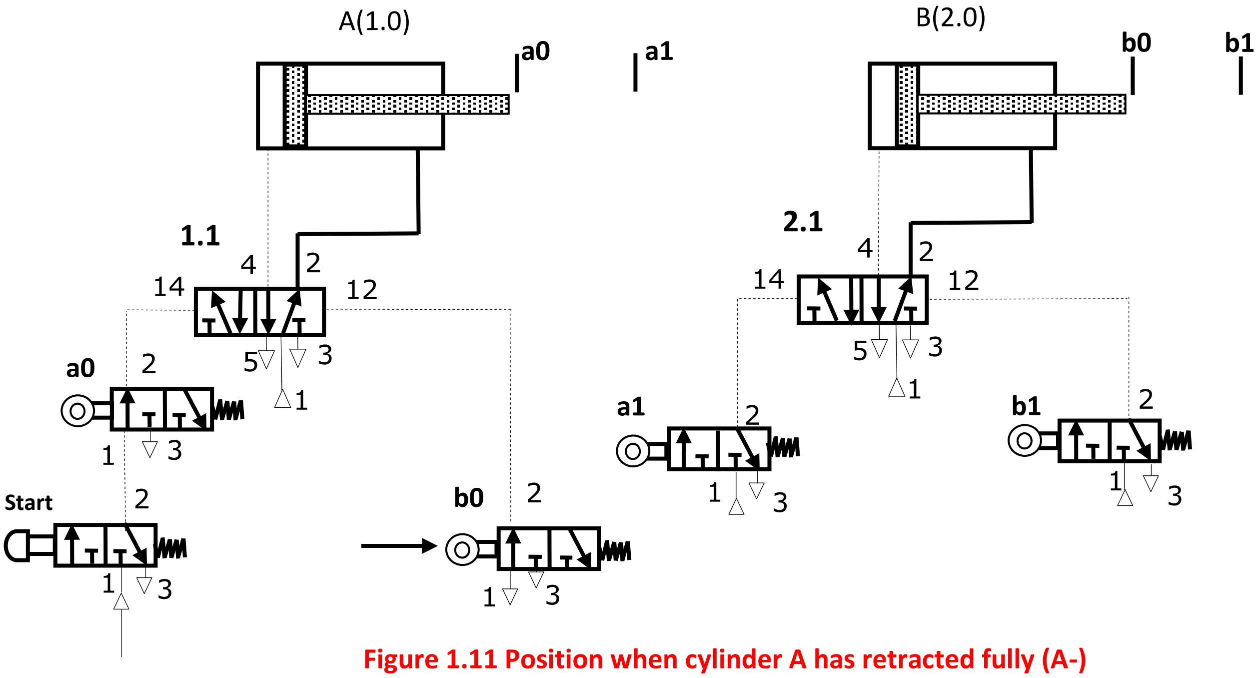


Figure 1.11 Position when cylinder A has retracted fully (A-)

11. Check for the signal at the other end 14 of the valve 1.1 Notice that signal is not present at port 14 of the valve 1.1 and hence there is no signal conflict. So valve 1.1 can switch over and Cylinder A can retract.

Example 2 : Two cylinders are used to transfer parts from a magazine onto a chute (Figure 1.12). When a push button is pressed, the first cylinder extends. Pushing the part from the magazine and positions it in preparation for transfer by the second cylinder onto the out feed chute. Once the part is transferred, the first cylinder retracts, followed by the second. Confirmation of all extended and retracted positions are required.

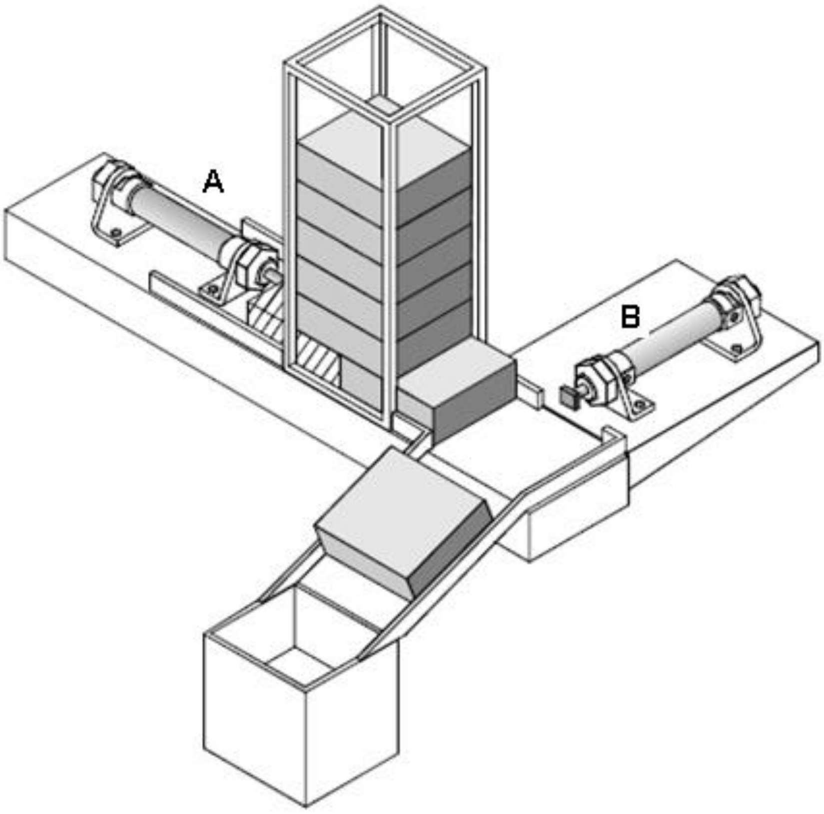


Figure 1.12 Positional diagram

Step 1: Write the statement of the problem:

Let A be the first cylinder (Pushing) and B be second cylinder (feeding) as shown in the Figure xxx. First cylinder A extends and brings under stamping station where cylinder B is located. Cylinder B then extends and stamps the job. Cylinder A can return back only cylinder B has retracted fully.

Step 2: Draw the positional layout. (Figure 1.13)

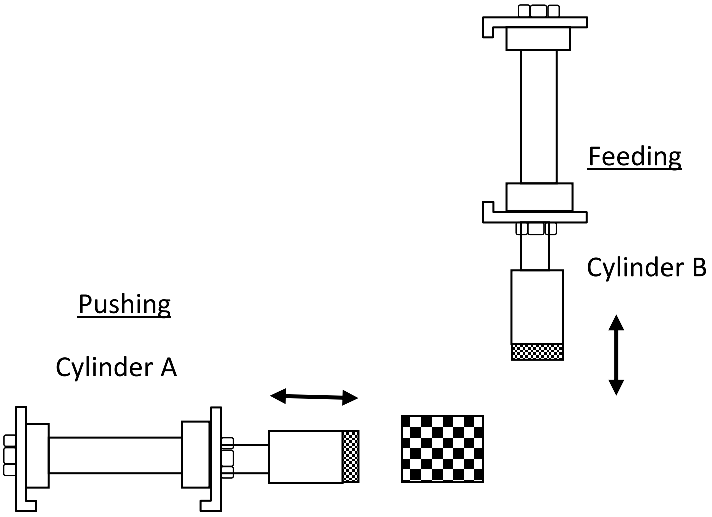


Figure 1.13 Positional diagram

Step3: Represent the control task using notational form

Cylinder **A** advancing step is designated as **A+**
Cylinder **A** retracting step is designated as **A-**
Cylinder **B** advancing step is designated as **B+**
Cylinder **B** retracting step is designated as **B-**

Therefore, given sequence for clamping and stamping is A+B+A-B-

Step 4 Draw the Displacement –step diagram (Figure 1.14)

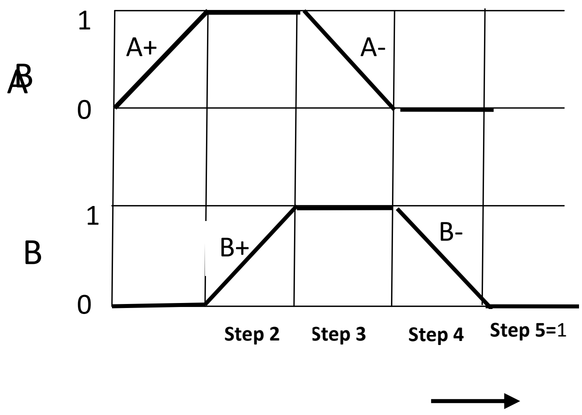


Figure 1.14 Displacement step diagram

Step 5 Draw the Displacement –time diagram (Figure 1.15)

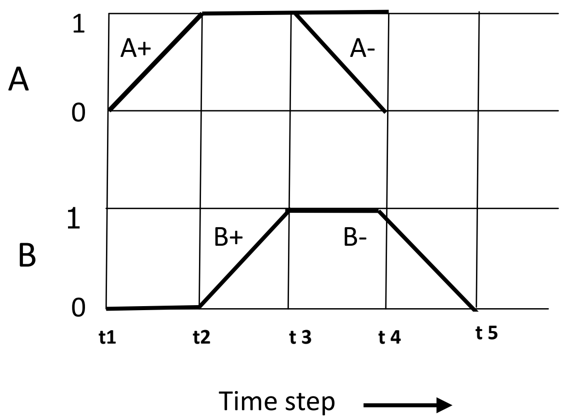


Figure 1.15 Displacement time diagram

Step 6: Analyse and Draw Pneumatic circuit.

Step 6.1 Analyse input and output signals.

Input Signals

Cylinder A – Limit switch at home position ao
Limit switch at home position a1
Cylinder B - Limit switch at home position bo
Limit switch at home position b1

Output Signal

Forward motion of cylinder A (A+)
Return motion of cylinder A (A-)
Forward motion of cylinder B(B+)

Return motion of cylinder B(B+)

Step 6.2 Using the displacement time/step diagram link input signal and output signal. (Figure 1.16)

Usually start signal is also required along with b0 signal for obtaining A+ motion.

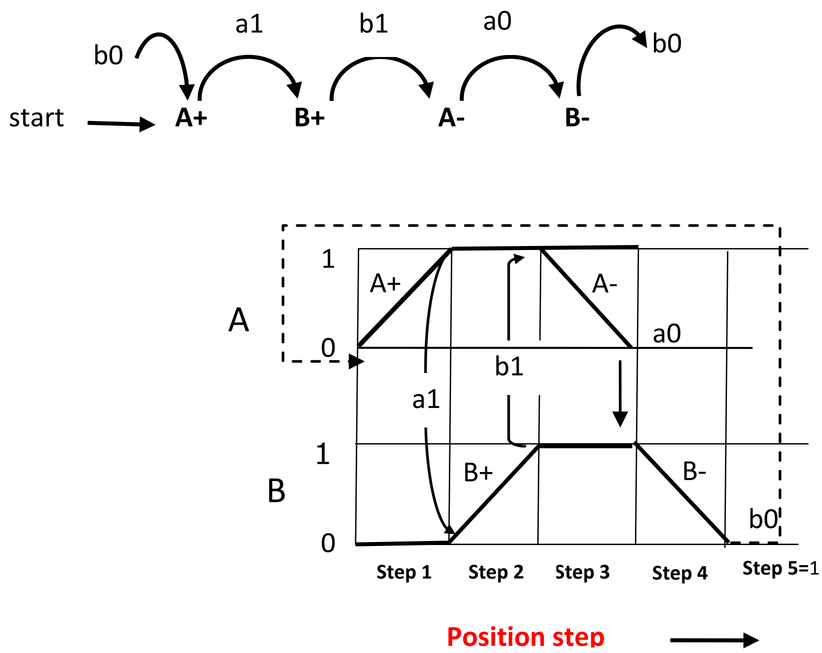
1. A+ action generates sensor signal a1, which is used for B+ motion

2. B+ action generates sensor signal b1, which is used for A- motion

3. A- action generates sensor signal a0, which is used for B- motion

4. B- action generates sensor signal b0, which is used for B- motion

Above information (given in 6.2) is shown below graphically



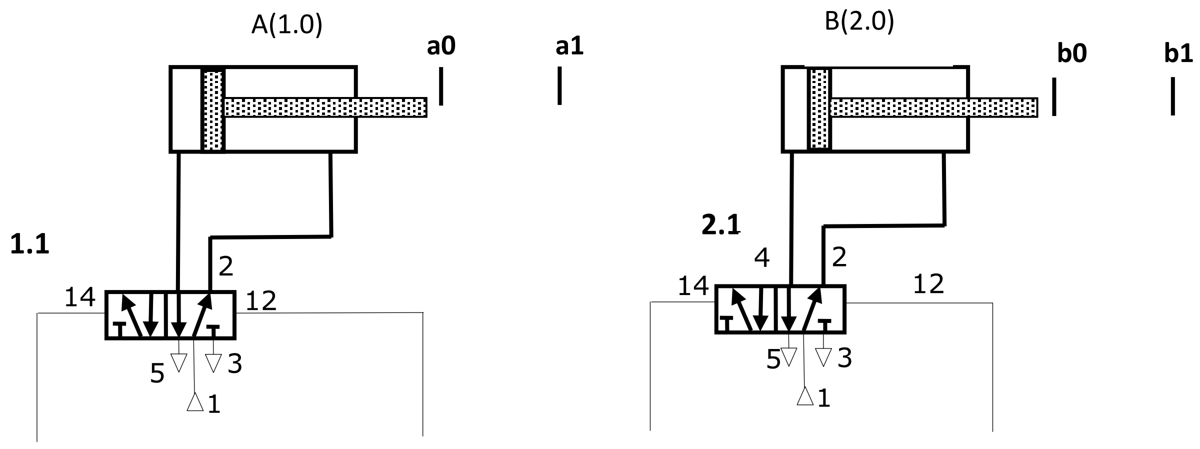
1.16 Input output signal flow

Step 7 Draw the power circuit (Figure 1.18)

i) Draw the cylinders A(1.0) and B(2.0).

ii) Draw the DCVs 1.1 and 2.1 in unactuated conditions

iii) Mark the limit switch positions for cylinders A(1.0) and B(2.0).



Step 8 Draw the control circuit (Figure 1.19)

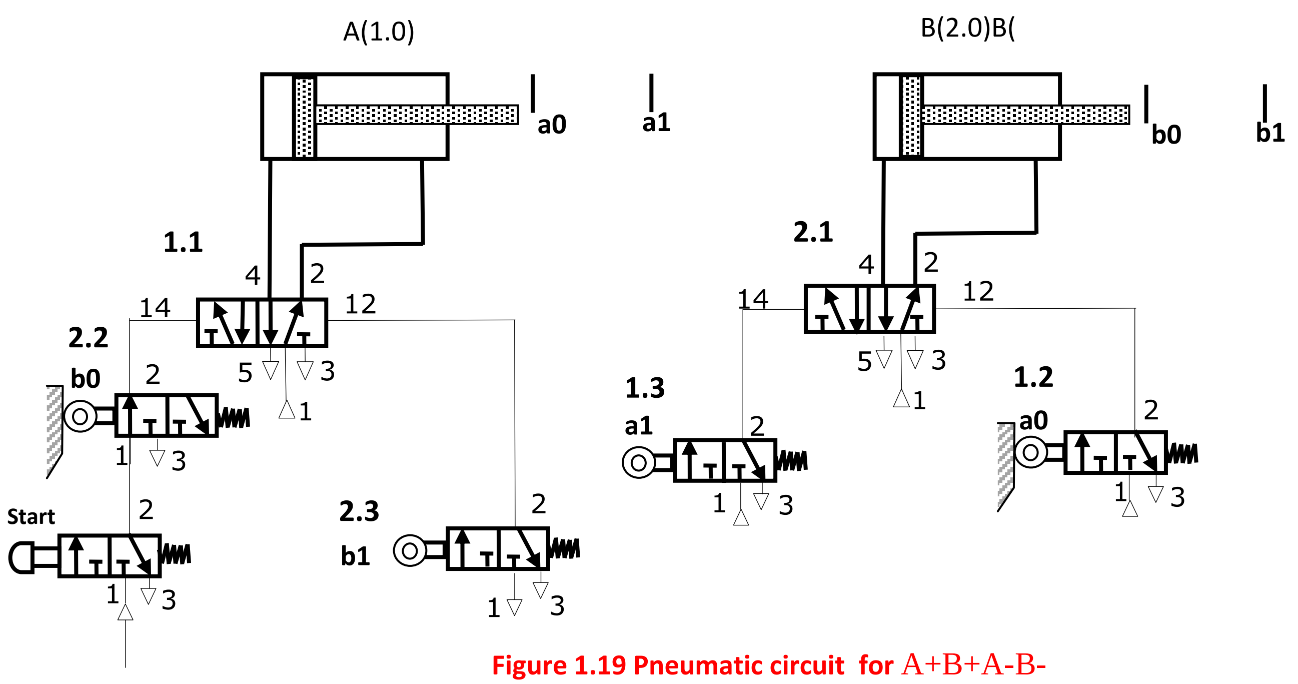


Figure 1.19 Pneumatic circuit for A+B+A-B-

Step 9 Analysis of pneumatic circuit

1. When the start button is pressed, the signal appears at port 14 of valve 1.1 through limit switch signal b0.

2. Check for the presence of the signal at the other end (12) of valve 1.1. Notice that the signal is not present at port 12 of valve 1.1. (Because b1 is not pressed). There is no signal conflict and valve 1.1 is able to move. So A advances to forward position.

3. When cylinder A fully extends, it generates a limit switch signal a1, which is applied to port 14 of the valve 2.1. Cylinder B advances to forward position.

5. Check for the presence of the signal at the other end (12) of valve 2.1. Signal is not present at port 12 of valve 2.1 (because a0 is not pressed, A is already in extended position now) and hence there is no signal conflict

6. Signal applied to port 14 of the valve 2.1 causes the shifting of DCV 2.1 and cylinder B extends.

7. When cylinder B fully extends, it generates a limit switch signal b1, which is applied to port 12 of valve 1.1 . Cylinder A returns and ao is pressed. There is no signal conflict, as ao and a1 are mutually exclusive signals.

10. When the cylinder A is fully retracted, it generates a limit switch signal a0, which is applied to port 12 of the valve 2.1. Cylinder B retracts.

All five sequence of operations are shown in Figure 1.20 to Figure 1.24

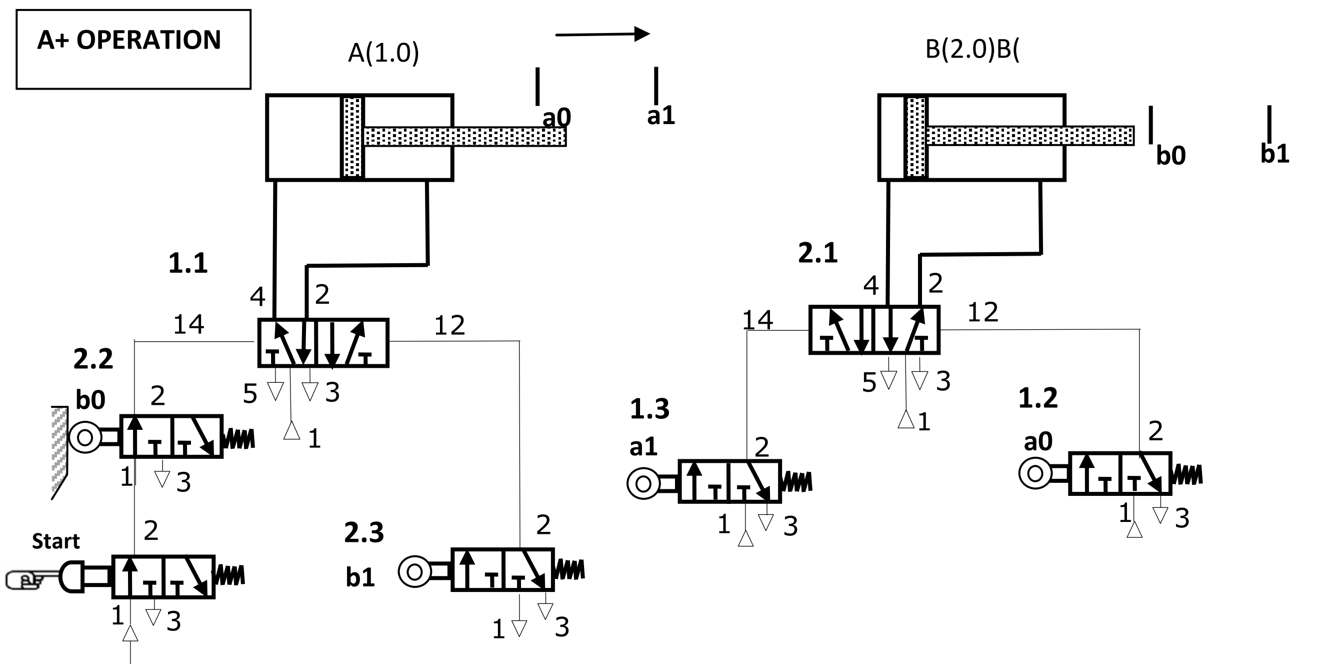


Figure 1.20 Pneumatic circuit for A+B+A-B- (A+ operation)

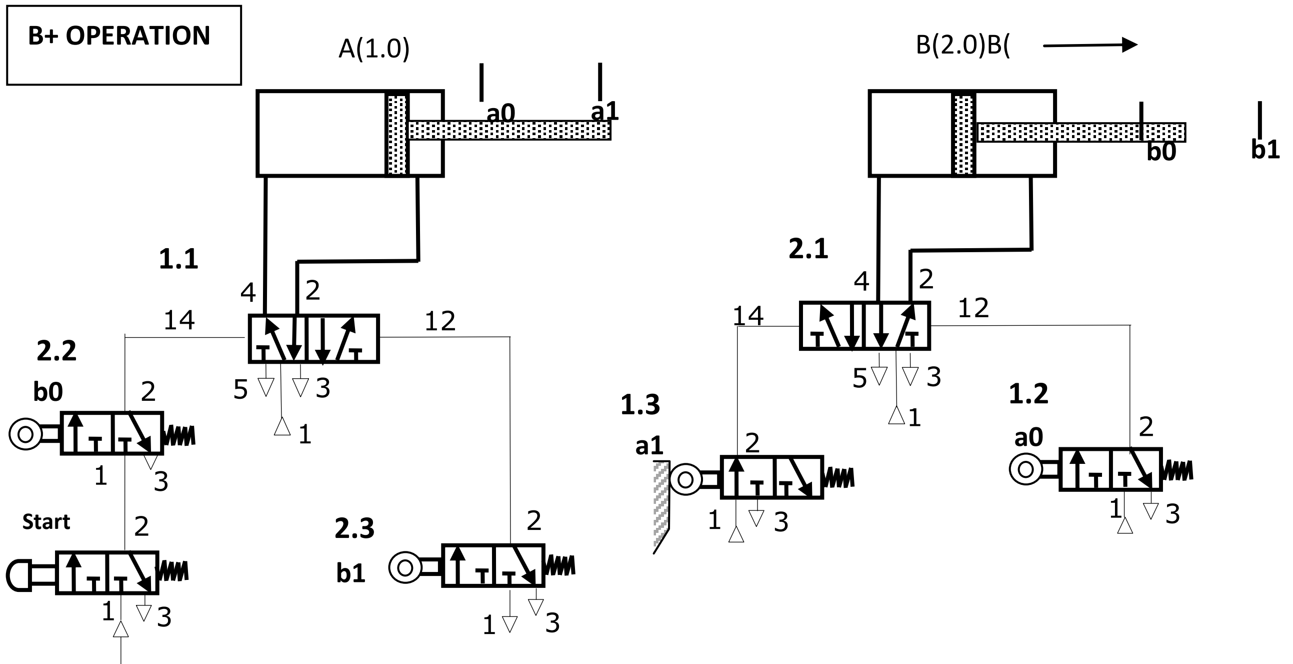


Figure 1.21 Pneumatic circuit for A+B+A-B- (B+ operation)

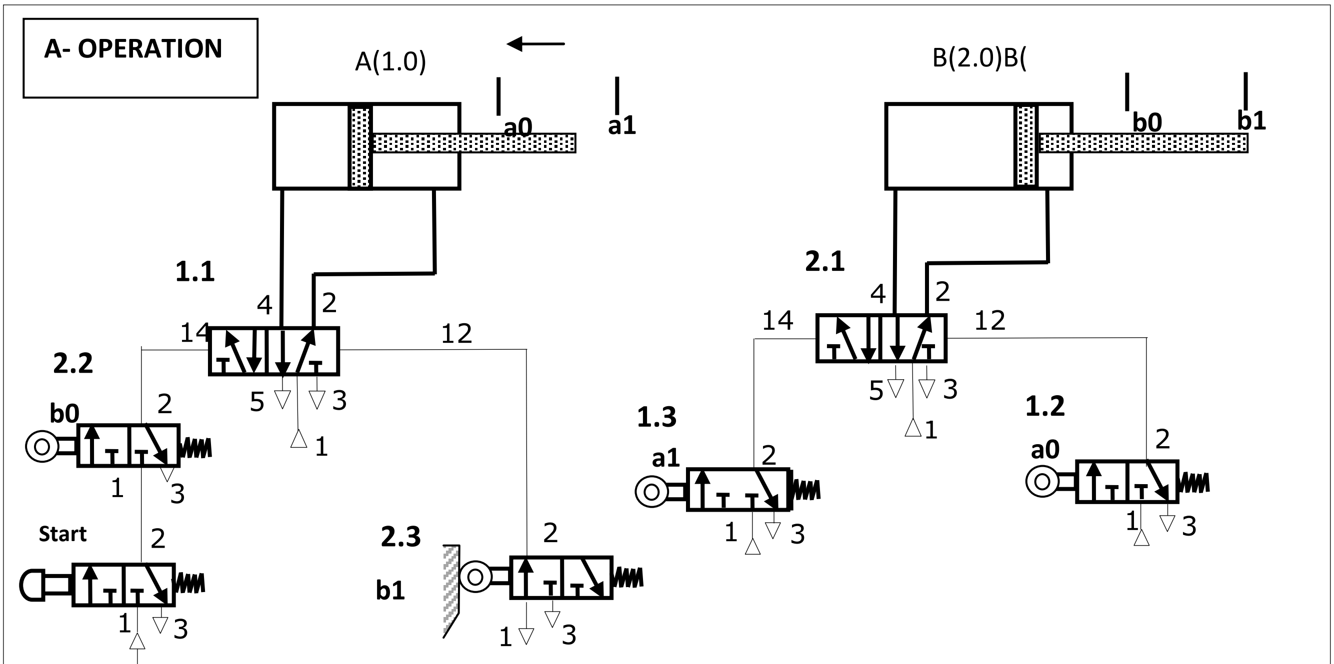


Figure 1.22 Pneumatic circuit for A+B+A-B- (A- operation)

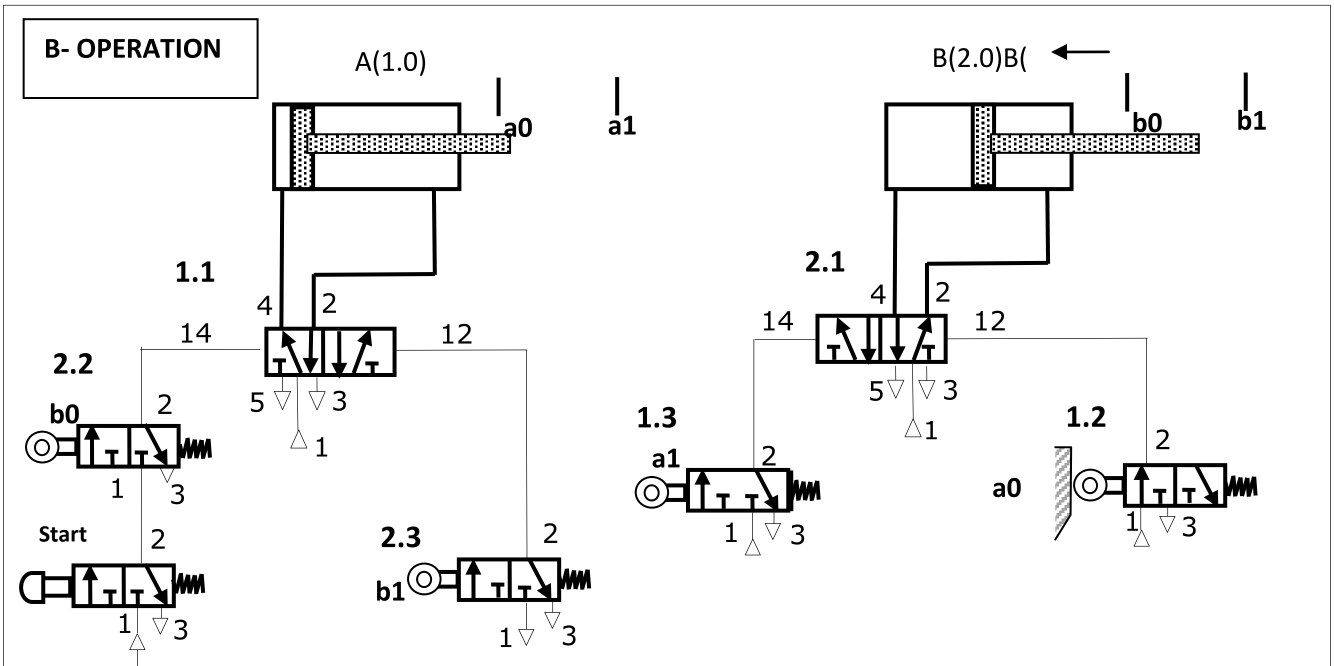


Figure 1.23 Pneumatic circuit for A+B+A-B- (B- operation)

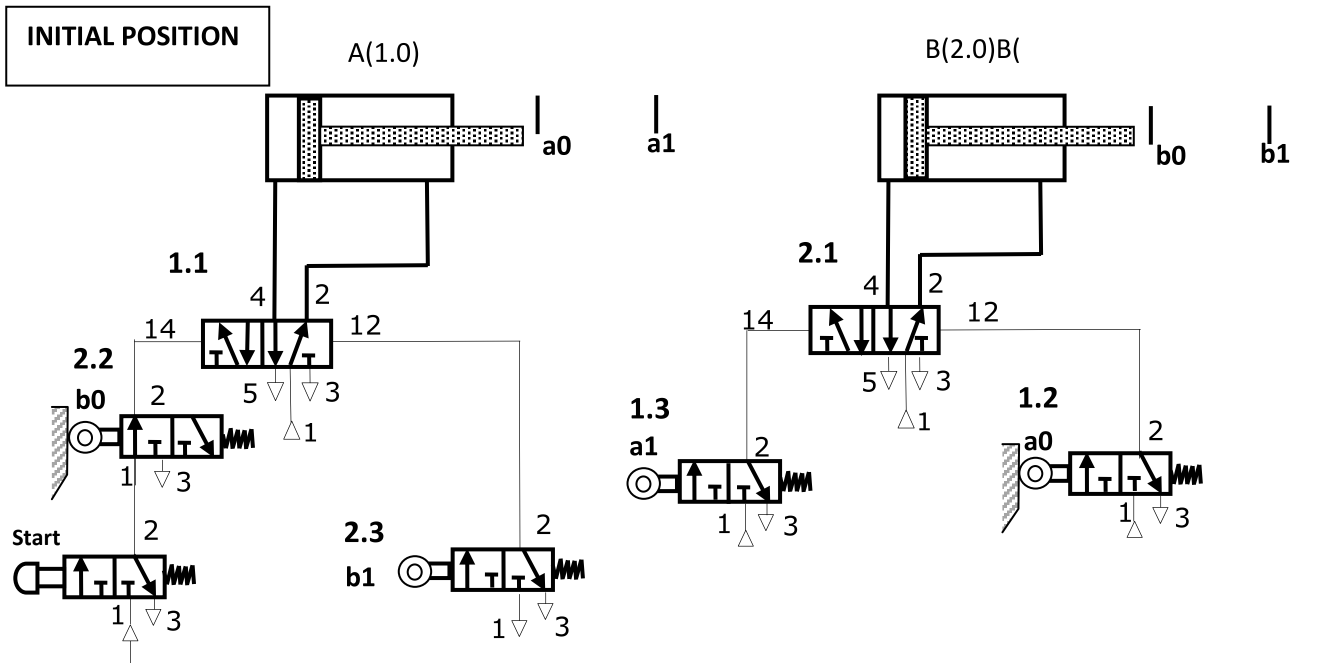


Figure 1.24 Pneumatic circuit for A+B+A-B- (initial position)

1.2.3 Elimination of Signal Conflict

Various methods are used to solve problem of signal conflicts in multi cylinder circuits.

a) Idle return roller
b) Reversing valves (memory valves)
c) Modules as combination of valves

Cascading method uses the revering valves (also known group changing valves) and Step counter method uses modular valves. Both methods are discussed in subsequent section in this chapter.

1.2.3.1 Use of Idle Return Rollers.

An idle-return roller valve consists of a 3/2 DCV fitted with an idle return roller mechanism. The two designs of the idle roller is shown in Figure 1.25

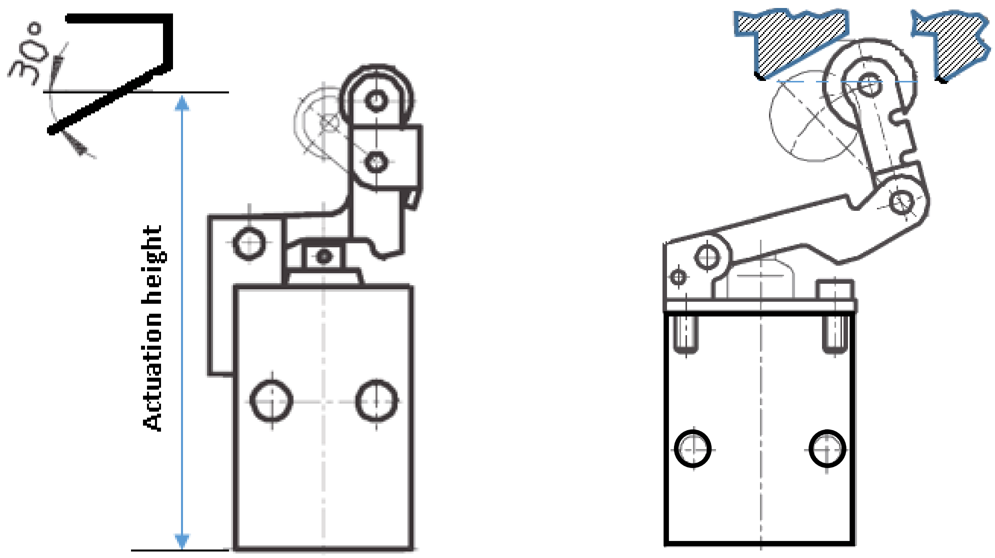


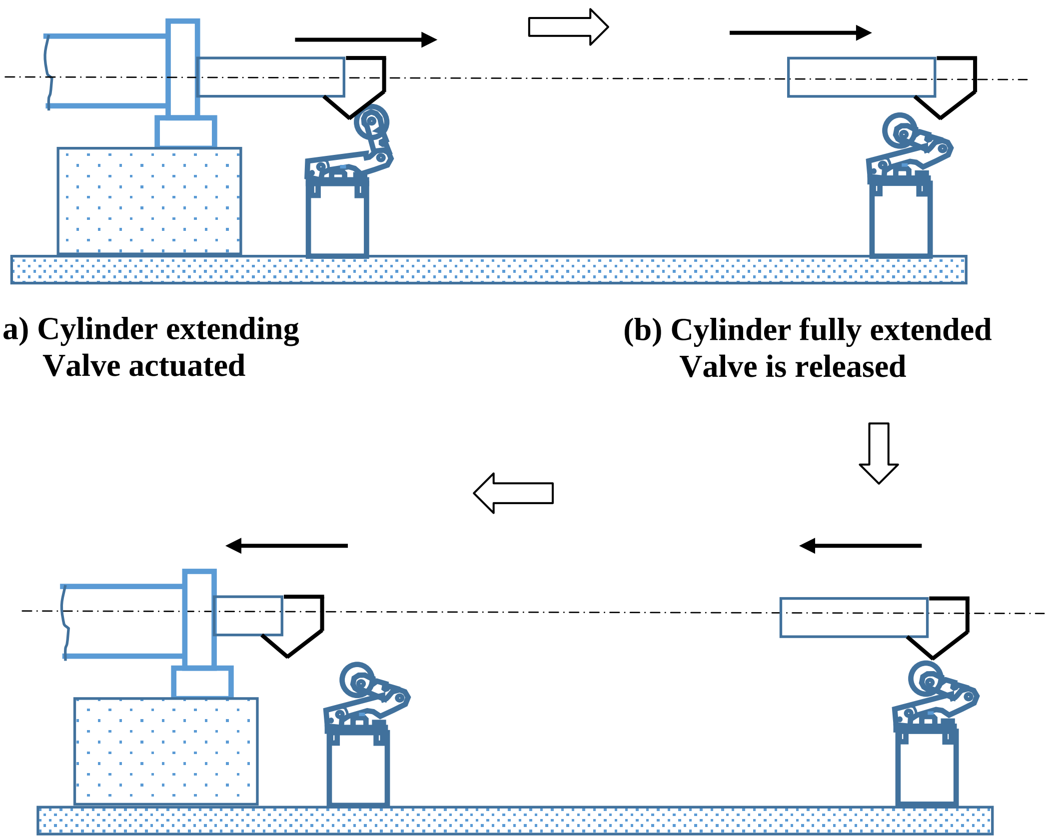
Figure 1.25 Two designs of Idle return rollers

The action of the idle return roller valve can be understood using the Figure 1.26 The idle return roller may be positioned in the control system so that when the cylinder extends, the piston passes over the idle – roller mechanism of the valve, thus activating the valve. (Figure 1.26a), but also permitting the valve to be deactivated immediately when the piston moves to the extreme end position (Figure 1.26b). As a result, the valve generates a short output pulse during the forward motion of the cylinder. The idle return mechanism also allows the cylinder to retract without re-activating the valve (Figure 1.26c and Figure 1.26d). Hence, in the end position or during the return motion of the piston, the valve does not gets actuated, and no output signal is produced. For the generation of short output pulse by the idle-return roller valve during the return motion of the cylinder, this valve may be positioned in the opposite direction as compared to the case during the forward motion of the cylinder.

In the previous sequence problem, we have identified that roller valves b1 and a1 are responsible for signal conflicts. To eliminate the problem of signal conflicts the roller valve b1 and a1 to be replaced by idle return rollers.

Drawbacks of idle –return rollers.

1. This method is not reliable
2. End position cannot be sensed accurately
3. Fast control system cannot be set up.



a) Cylinder extending
Valve actuated

(b) Cylinder fully extended
Valve is released

(d) Cylinder fully retracted
Valve is NOT actuated

(c) Cylinder retracting
Valve is NOT actuated

Figure 1.26 Actions of Idle return rollers

Example 3 : Develop Pneumatic circuit for A+ B+ B- A- sequence. Avoid signal conflict using idle – return roller.

Figure 1.27 shows the circuit for getting the control sequence A+ B+ B- A- using the idle –return rollers at the position bo and a1. The roller valves at position a0 and b1 need be replaced with the idle return rollers as these valves do not cause signal conflicts for the given sequence circuit.

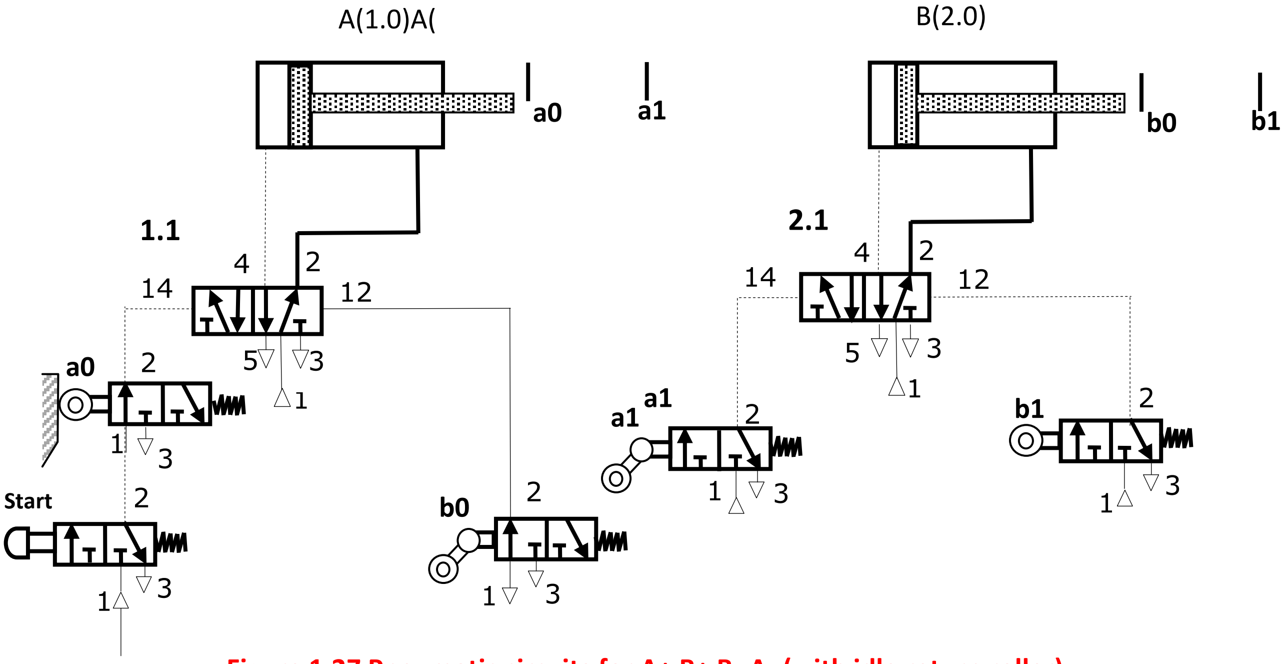


Figure 1.27 Pneumatic circuits for A+ B+ B- A- (with idle return roller)

1.3 CASCADE METHOD

A Bi-stable memory valve or reversing valve can be used to eliminate signal conflicts. Signal conflict is avoided by allowing the signal to be effective only at times when they are needed. Two of the possible designs are possible.

i) Cascade method

ii) Shift register method

1.3.1 Demonstration of Cascade method

In order to develop control circuitry for multi cylinder applications, as done before in classic method, it is necessary to draw the motion diagram to understand the sequence of actuation of various signal input switches-limit switches and sensors. Motion diagram represents status of cylinder position - whether extended or retracted in a particular step

Step 1: Write the statement of the problem:

First cylinder A extends and brings under stamping station where cylinder B is located. Cylinder B then extends and stamps the job. Cylinder A can return back only cylinder B has retracted fully.

Step 2: Draw the positional layout. (Figure 1.28)

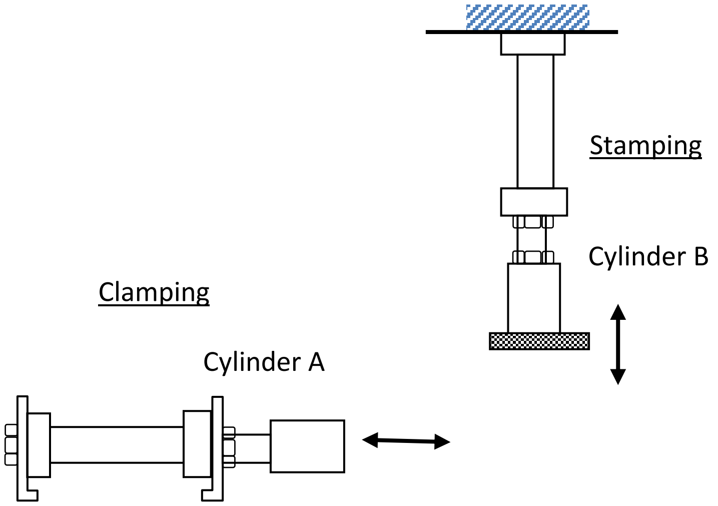


Figure 1.28 Positional diagram

Step3: Represent the control task using notational form

Cylinder **A** advancing step is designated as **A+**
Cylinder **A** retracting step is designated as **A-**
Cylinder **B** advancing step is designated as **B+**
Cylinder **B** retracting step is designated as **B-**
Given sequence for clamping and stamping is A+B+B-A-

Step 4 Draw the Displacement –step diagram (Figure 1.29)

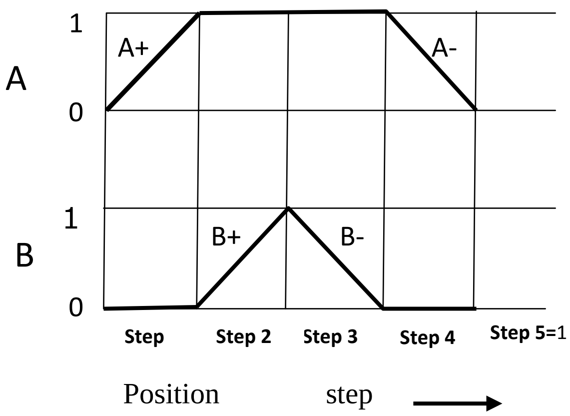


Figure 1.29 Displacement step diagram

Step 5 Draw the Displacement –time diagram (Figure 1.30)

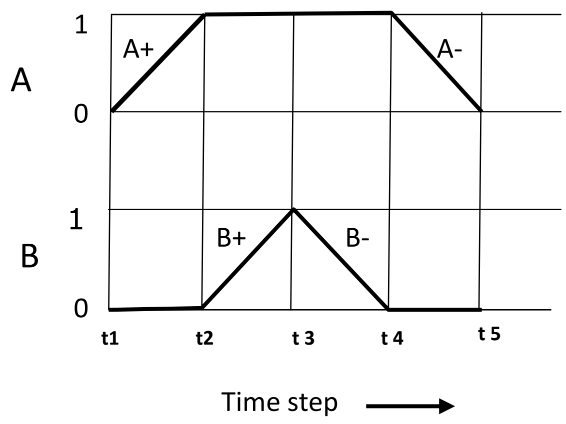


Figure 1.30 Displacement time diagram

Step 6: Analyse and Draw Pneumatic circuit.

Step 6.1 Analyse input and output signals.

Input Signals

Cylinder A – Limit switch at home position ao
Limit switch at home position a1
Cylinder B - Limit switch at home position bo
Limit switch at home position b1

Output Signal

Forward motion of cylinder A (A+)
Return motion of cylinder A (A-)
Forward motion of cylinder B (B+)
Return motion of cylinder B(B+)

Step 6.2 Using the displacement time/step diagram link input signal and output signal. (Figure 1.31)

Usually start signal is also required along with a0 signal for obtaining A+ motion.

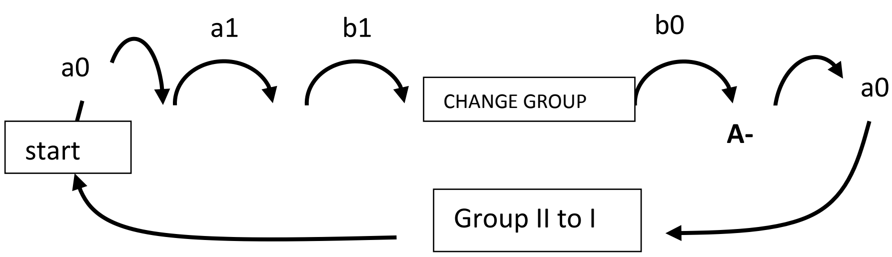
1. A+ action generates sensor signal a1, which is used for B+ motion

2. B+ action generates sensor signal b1, which is used group changing.

3. B- action generates sensor signal b0, which is used for A- motion

4. A- action generates sensor signal a0, which is used for group changing

Above information (given in 6.2) is shown below graphically



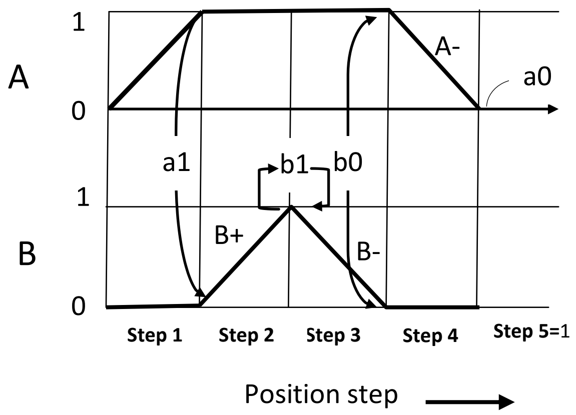


Figure 1.31 Displacement time diagram

Step 7 Draw the power circuit (Figure 1.32)

i) Divide the given circuits into groups. Grouping should be done such that there is no signal conflict. Do not put A+ and A- in the same group. Similarly B+ and B- should not be put in the same group. In other word A+ and A- should belong to different group to avoid signal conflict.

In our example of **A+ B+ B- A- we can group as**

A+ B+	B- A-
Group 1	Group 2

ii) Choose the number of **group changing valve** = no of groups -1

In our example, we have 2 groups so we need one group changing valve

Connect the group changing valve as follows. From the figure it is clear that when the control signals I and II are applied to group changing valve, the air (power) supply changes from Group 1(G1) to Group 2 (G2)

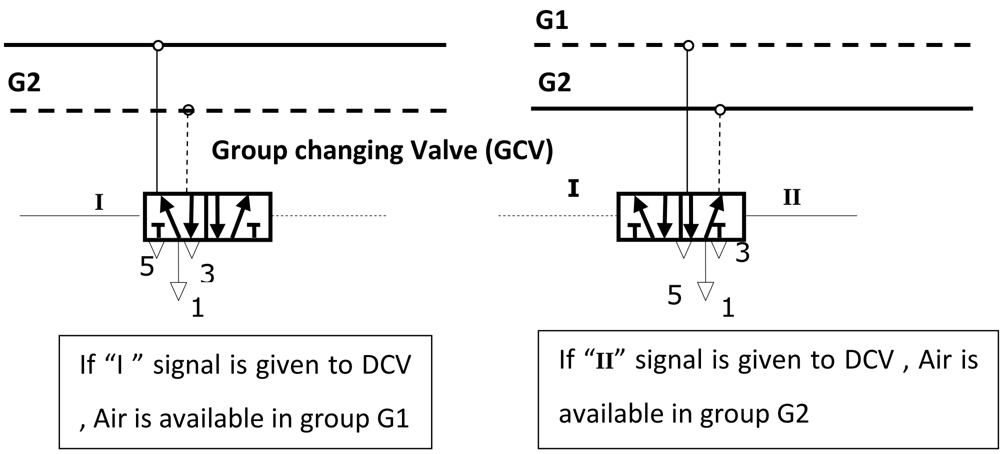


Figure 1.32

iii) Arrange the limit switch and start button as given below (Figure 1.33)

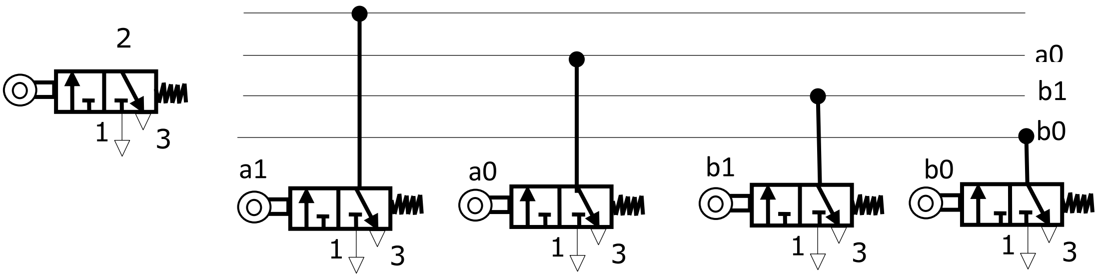


Figure 1.33

iv) Draw the power circuit (Figure 1.34)

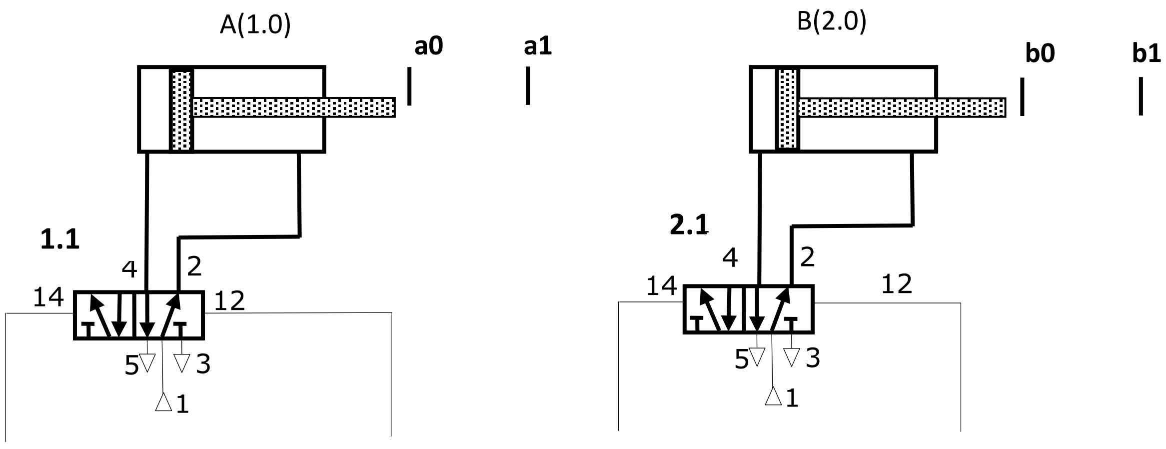


Figure 1.34

Step 8 Draw the control circuit (Figure 1.35)

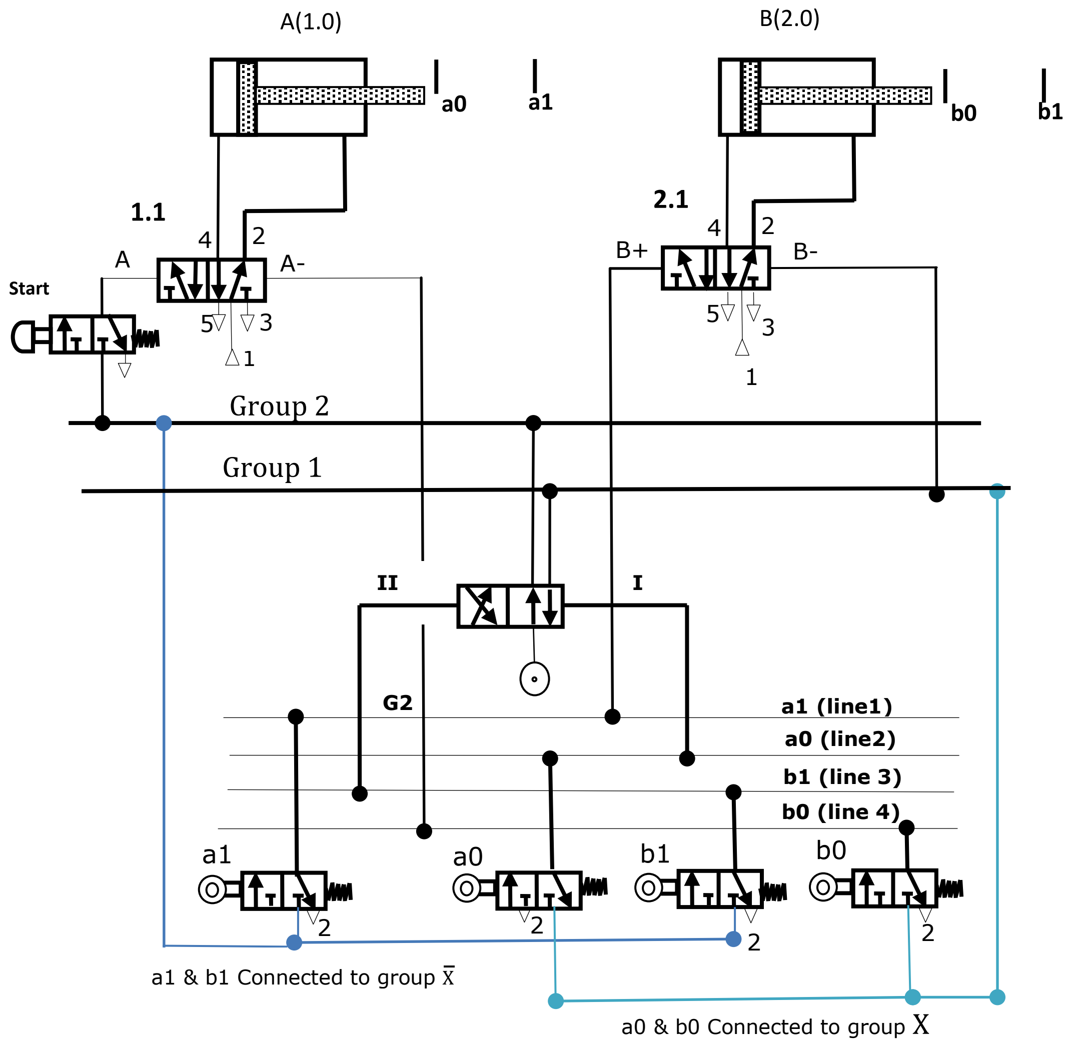


Figure 1.35 Pneumatic circuits for A+ B+ B- A-

Step 9 Analysis of pneumatic circuit

1. Assume that air is available in the line G2 to start with. (Say from last operation)

2. When the start button is pressed, Air supply from Group G2 is directed to line 2 through actuated limit switch a0. Now the air available in line 2, actuates the Group changing valve (GCV) to switch over to position I. This switching of the GCV causes air supply to change from G2 to G1.

3. Now the air is available in line G1. The air supply from group G1 is directed to port 14 of the valve 1.1. As there is no possibility of signal conflict here, valve 1.1 switches over causing the A+ action.

4. Sensor a1 is actuated as the result of A+ action, allowing the air supply from the Group G1 to reach to line 1 through a1. Now the air available reaches port 14 of valve 2.1. As there is no possibility of signal conflict here, valve 2.1 switches over, causing B+ action automatically.

5. Sensor b1 is actuated as result of B+ action, allowing the air supply in line 3. Air from line 3 allows the air to reach port 12 of Group changing valve (also called reversing valve). As a result, the Group changing valve switches over, causing the group supply to change from G1 to G2.

6. Now the air is available in G2. Air from G2 acts on port 12 of the Valve 2.1. As there is no possibility of signal conflict here, valve 2.1 switches over, causing B- action automatically.

7. Sensor is actuated as the result of B- action. Now the air is available in line 4.Air from line 4 reach port 12 of the valve 1.1 , As there is no possibility of signal conflict here, valve 2.1 switches over , causing A- action automatically.

The cascade system provides a straightforward method of designing any sequential circuit. Following are the important points to note:

a) **Present** – the system must be set to the last group for start-up
b) **Pressure drop**- Because the air supply is cascaded, a large circuit can suffer from more pressure drop.
c) **Cost** – Costly due to additional reversing valves and other hardware.

Objective Type Questions

1. Design of circuits using intuitive method is --------------- time consuming compared to step counter method.
2. When hard ware costs are not important but circuit design time must be minimal, then ...... method of the circuit design is used.
3. Where the hardware costs are paramount, then -------------- circuit methods are used.
4. When absolute fool proofing of circuits is required, ------------------ circuit methods are used.
5. --------------- are used to sense the end position of cylinder movements

State True or False

1. We have to draw all valves in their de-actuated , unpressurised rest position as in electrical switching components

2. In order to gain fully controlled sequence of all cylinders in a program , it is essential to install end position sensors at all movements end position.

3 4/2 or 5/2 double piloted valves are free from signal overlaps.

4. Step counter method is absolutely free from signal conflicts

5. Time delay and pressure delay functions cannot be used in multicylinder circuit design

Review Questions

1. Explain how signal conflict occurs using an example

2. What is the effect of signal conflict in multi actuator circuit design

3. What are the different ways to eliminate signal conflict in multi actuator circuit design

4. Explain the step displacement diagram for A+B+B-A- sequence.

5. List few disadvantages of using idle return rollers for overcoming signal conflicts

6. Explain the principle of cascade method with a suitable sequence example

7. Briefly explain the principles of step counter method with a suitable example

1. Draw a group changing cascade circuit for two groups, three groups, four groups and five groups.

9. Explain with the help of neat sketch the construction and working of pressure sequence valve

10. What are the different ways to sense the end position and movement of cylinders

Answers

Fill in the Blanks

1. More
2. Step counter
3. logic
4. step counter
5. limit switches

State True or False

1. True
2. True
3. False
4. True
5. False

Lecture 41

ELECTRO – PNEUMATIC CONTROL

Learning Objectives

Upon completion of this chapter, Student should be able to

- Explain the various steps involved in electro pneumatics
- List seven basic electrical devices used in electro pneumatics
- Describe the constructional details of solenoid valves
- Explain the operations of control devices like limit switches, sensors, timers , counters and pressure switches
- Differentiate between capacitive and inductive proximity sensors
- Differentiate between dominant on and off latching circuits
- Design single actuator electro pneumatic circuits
- Design a sequence circuits using two and three cylinders

1.1 INTRODUCTION

Electro pneumatics is now commonly used in many areas of Industrial low cost automation. They are also used extensively in production, assembly, pharmaceutical, chemical and packaging systems. There is a significant change in controls systems. Relays have increasingly been replaced by the programmable logic controllers in order to meet the growing demand for more flexible automation.

Electro-pneumatic control consists of electrical control systems operating pneumatic power systems. In this solenoid valves are used as interface between the electrical and pneumatic systems. Devices like limit switches and proximity sensors are used as feedback elements.

Electro Pneumatic control integrates pneumatic and electrical technologies, is more widely used for large applications. In Electro Pneumatics, the signal medium is the electrical signal either AC or DC source is used. Working medium is compressed air. Operating voltages from around 12 V to 220 Volts are often used. The final control valve is activated by solenoid actuation

The resetting of the valve is either by spring [single Solenoid]or using another solenoid [Double solenoid Valve] . More often the valve actuation/reset is achieved by pilot assisted solenoid actuation to reduce the size and cost of the valve

Control of Electro Pneumatic system is carried out either using combination of Relays and Contactors or with the help of Programmable Logic Controllers [PLC]. A Relay is often is used to convert signal input from sensors and switches to number of output signals [either normally closed or normally open] .Signal processing can be easily achieved using relay and contactor combinations

A Programmable Logic Controller can be conveniently used to obtain the out puts as per the required logic, time delay and sequential operation.. Finally the out put signals are supplied to the solenoids activating the final control valves which controls the movement of various cylinders. The greatest advantage of electro pneumatics is the integration of various types of proximity sensors [electrical] and PLC for very effective control. As the signal speed with electrical signal, can be much higher, cycle time can be reduced and signal can be conveyed over long distances.

In Electro pneumatic controls, mainly three important steps are involved:

- **Signal input devices** -Signal generation such as switches and contactor, Various types of contact and proximity sensors
- **Signal Processing** – Use of combination of Contactors of Relay or using Programmable Logic Controllers
- **Signal Out puts** – Out puts obtained after processing are used for activation of solenoids, indicators or audible alarms

1.2 SEVEN BASIC ELECTRICALDEVICES

Seven basic electrical devices commonly used in the control of fluid power systems are

1. Manually actuated push button switches
2. Limit switches
3. Pressure switches
4. Solenoids
5. Relays
6. Timers
7. Temperature switches

Other devices used in electro pneumatics are

1. Proximity sensors
2. Electric counters

1.2.1 Push button switches

A push button is a switch used to close or open an electric control circuit. They are primarily used for starting and stopping of operation of machinery. They also provide manual override when the emergency arises. Push button switches are actuated by pushing the actuator into the housing. This causes set of contacts to open or close.

Push buttons are of two types

i) Momentary push button

ii) Maintained contact or detent push button

Momentary push buttons return to their unactuated position when they are released. Maintained (or mechanically latched) push buttons has a latching mechanism to hold it in the selected position.

The contact of the push buttons, distinguished according to their functions,

i) Normally open (NO) type

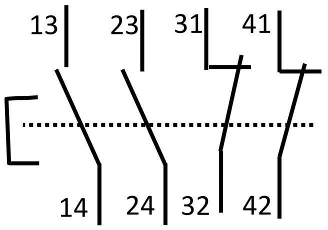
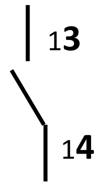
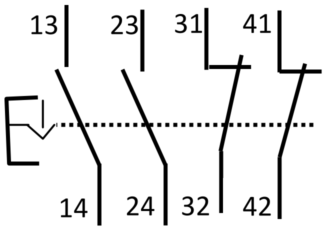
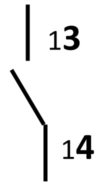
ii) Normally closed (NC) type

iii) Change over (CO) type.

The cross section of various types of push buttons in the normal and actuated positions and their symbols are given in the Figure 1.1 In the NO type, the contacts are open in the normal position, inhibiting the energy flow through them. But in the actuated position, the contacts are closed, permitting the energy flow through them. In the NC type, the contacts are closed in the normal position, permitting the energy flow through them. And, the contacts are open in the actuated position, inhibiting the energy flow through them. A changeover contact is a combination of NO and NC contacts.

Type of devices	***Terminal Numbers***	
	Normally closed contacts	**Normally open contacts**
Push buttons and Relays	1 and 2	3 and 4
Timers and Counters	5 and 6	7 and 8

Designation of the pushbuttons

Type of contact	Designation
Momentary contact type PB station (2 NO +2 NC) 	First digit indicates the function of contact. Second digit represents a serial ordering. 3 and 4 represents **NO** contacts and 1 is the serial no. 
Maintained Contact type PB station (2NO+2 NC) 	First digit indicates the function of contact. Second digit represents a serial ordering. 3 and 4 represents **NO** contacts and 1 is the serial no. 

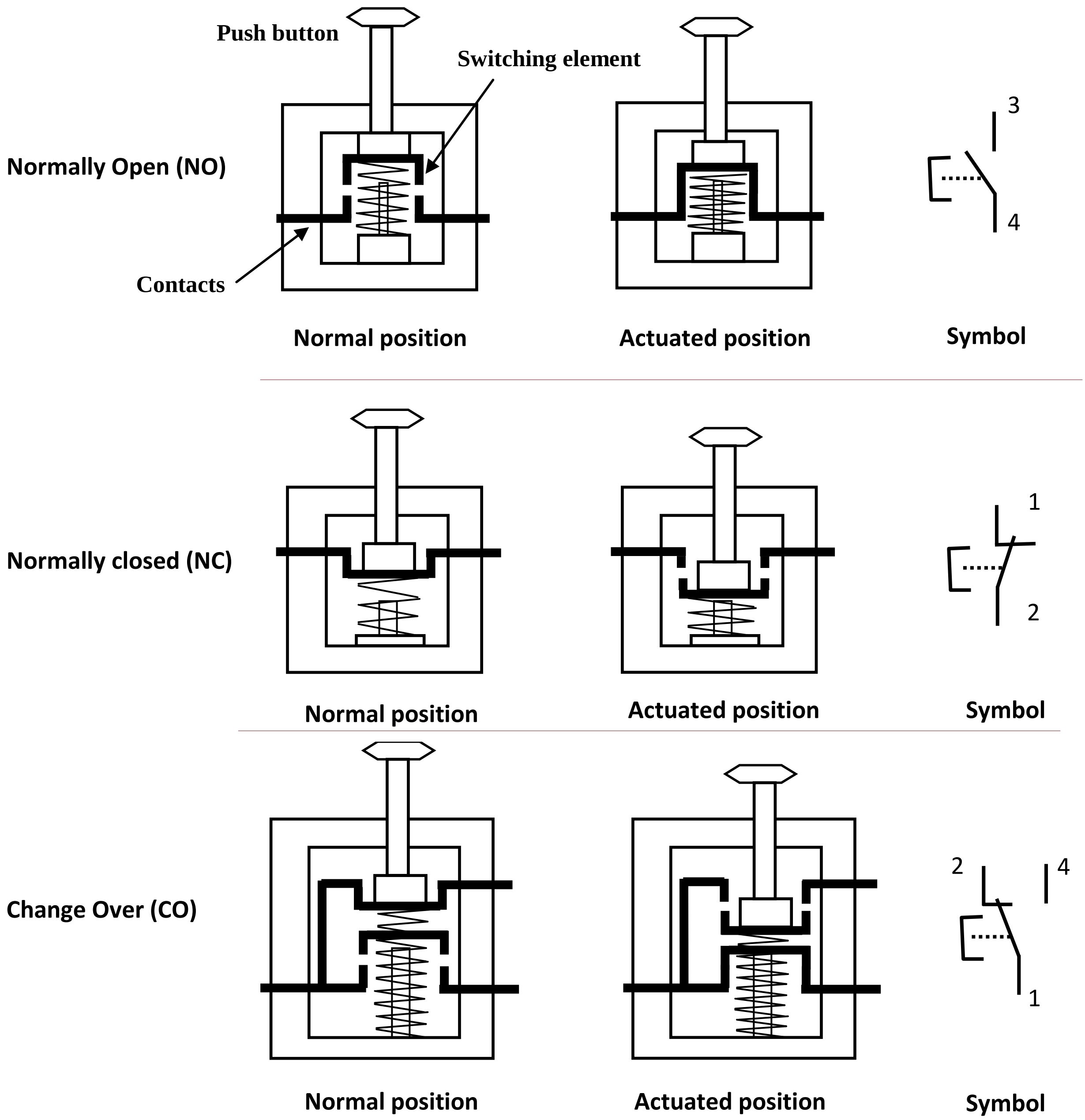


Figure 1.1 : Pushbuttons and their symbols

1.2.2 Limit switches

Any switch that is actuated due to the position of a fluid power component (usually a piston rod or hydraulic motor shaft or the position of load is termed as limit switch. The actuation of a limit switch provides an electrical signal that causes an appropriate system response.

Limit switches perform the same function as push button switches. Push buttons are manually actuated whereas limit switches are mechanically actuated.

There are two types classification of Limit switches depending upon method of actuations of contacts

a) Lever actuated contacts
b) Spring loaded contacts

In lever type limit switches, the contacts are operated slowly. In spring type limit switches, the contacts are operated rapidly. **Figure 1.2** shows a simplified cross sectional view of a limit switch and its symbol.

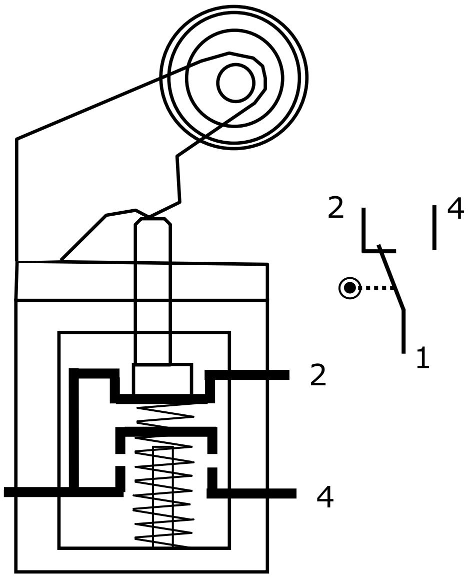


Figure 1.2: Cross sectional view of a limit switch

1.2.3 Pressure switches

A pressure switch is a pneumatic-electric signal converter. Pressure switches are used to sense a change in pressure, and opens or closes an electrical switch when a predetermined pressure is reached. Bellow or diaphragm is used to sense the change of pressure. Bellows or Diaphragm is used to expand or contract in response to increase or decrease of pressure. **Figure 1.3** shows a diaphragm type of pressure switch. When the pressure is applied at the inlet and when the pre-set pressure is reached, the diaphragm expands and pushes the spring loaded plunger to make/break contact.

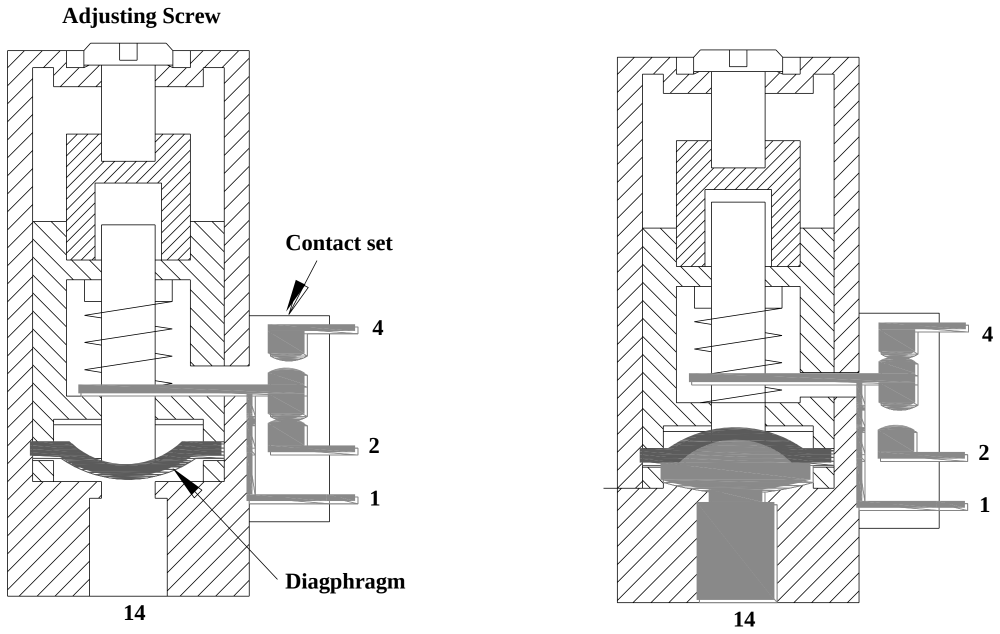


Figure 1.3: Cross sectional view of a pressure switch

1.2.4 Solenoids

Electrically actuated directional control valves form the interface between the two parts of an electro-pneumatic control. The most important tasks of electrically actuated DCVs include.

i) Switching supply air on or off
ii) Extension and retraction of cylinder drives

Electrically actuated directional control valves are switched with the aid of solenoids. They can be divided into two groups:

i) Spring return valves only remain in the actuated position as long as current flows through the solenoid
ii) Double solenoid valves retain the last switched position even when no current flows through the solenoid.

In the initial position, all solenoids of an electrically actuated DCVs are de-energised and the solenoids are inactive. A double valve has no clear initial position, as it does not have a return spring. The possible voltage levels for solenoids are 12 V DC, 12V AC, 12 V 50/60 Hz, 24V 50/60 Hz, 110/120V 50/60 Hz, 220/230V 50/60 Hz.

a) 3/2 Way single solenoid valve, spring return.

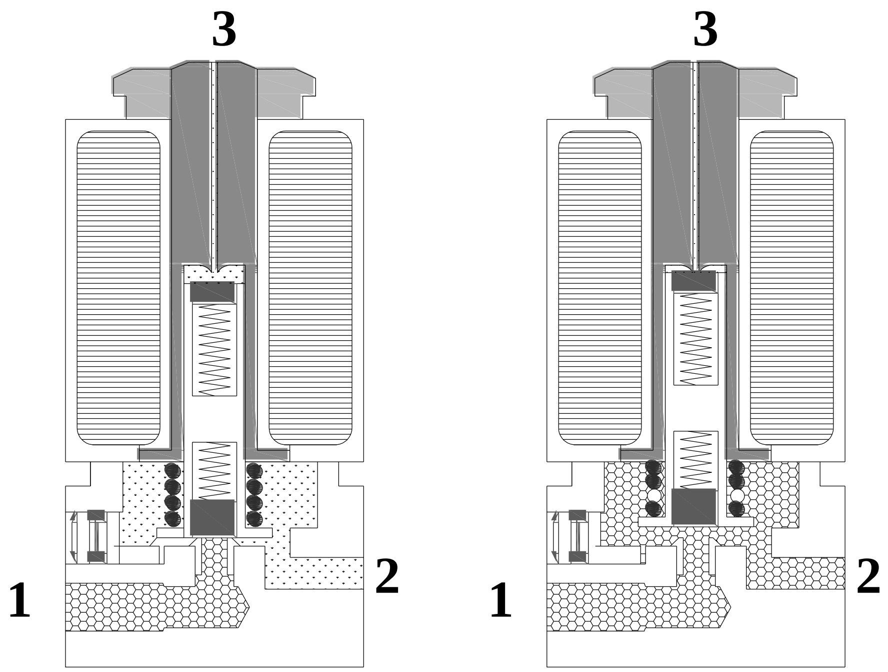


Figure 1.4: Cross sectional view of a 3/2 single solenoid valve

The cross sectional view of 3/2 way single solenoid valve in the normal and actuated positions are shown in **Figure 1.4**. In the normal position, port 1 is blocked and port 2 is connected to port 3 via back slot (details shown in the circle) When the rated voltage is applied to coil, armature is pulled towards the centre of the coil and in the process the armatures is lifted away from the valve seat. The compressed air now flows from port 1 to port 2, and ports 3 is blocked. When the voltage to the coil is removed, the valve returns to the normal position. **Figure 1.5** shows 2/2 solenoid operated valve

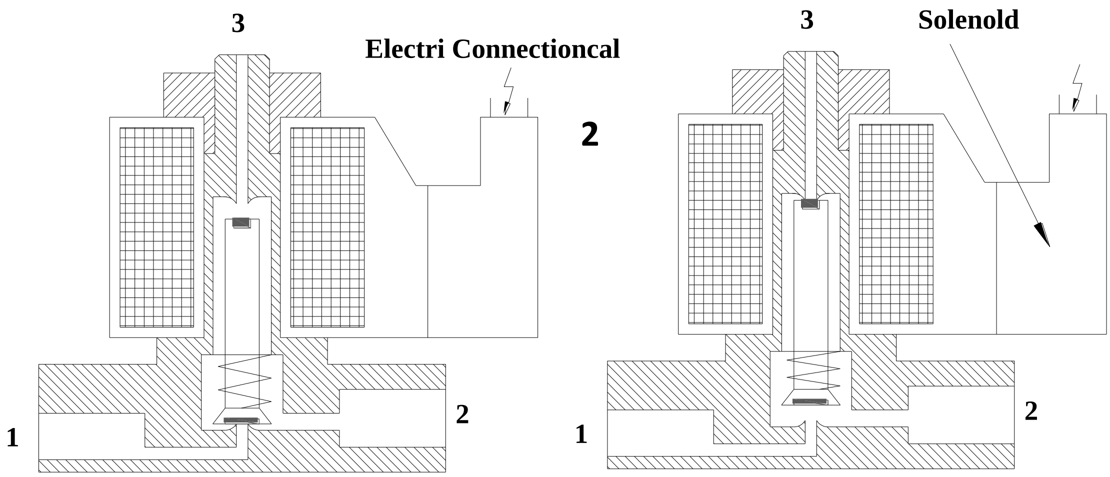


Figure 1.5: Cross sectional view of a 2/2 way solenoid operated valve

b) 5/2 Way single solenoid valve, spring return.

The cross section view of 5/2 way single solenoid in the normal and actuated positions are shown in **Figure 1.6.** In normal position, port 1 is connected to port 2, port 4 is connected to port 5, and port 3 is blocked. When the rated voltage is applied to coil 14, the valve is actuated through an internal pilot valve. In actuated position, port 1 is connected to port 4 , port 2 is connected to port 3, and port 5 is blocked. The valve returns to the normal position when the voltage to the armature coil is removed. This type of valves is normally used as final valve to control double acting cylinders.

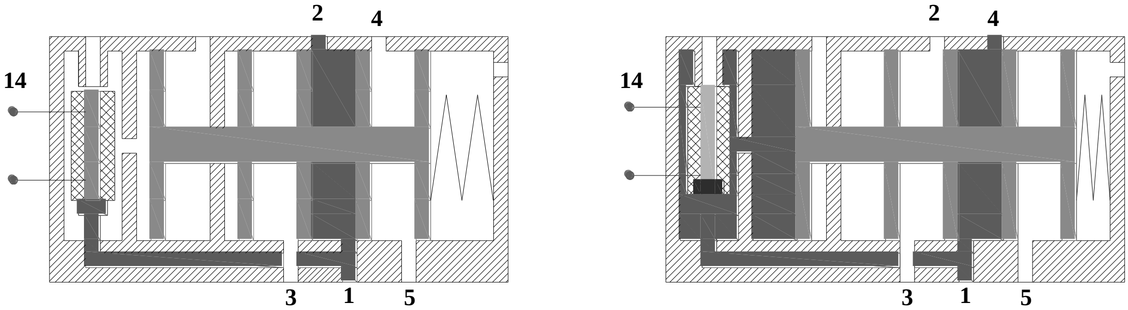


Figure 1.6: Cross sectional view of a 5/2 way solenoid operated valve

c) 5/2 Way single double solenoid valve

The cross section view of 5/2 way double solenoid in the normal and actuated positions are shown in the **Figure 1.7** when the rated voltage is applied to coil 14, the valve is actuated to a one switch in position with port 1 connected to port 4, port 2 connected to port 3, and port 5 blocked. When the rated voltage is applied to the coil 12, the valve is actuated to the other switching position with port 1 connected to port 2, port 4 connected to port 5 and port 3 blocked.

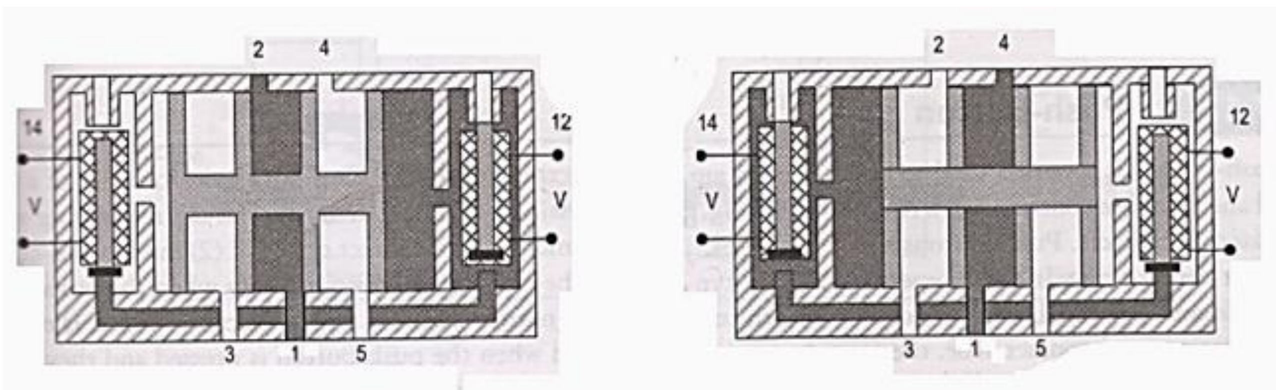
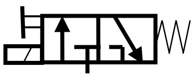
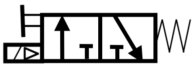
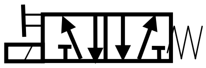
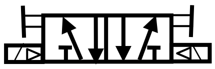


Figure 1.7: Cross sectional view of a 5/2 way double solenoid operated valve

The symbols for the various solenoid/pilot actuated valves are given in **Table 1.1**

Table 1.1 Various symbols for DCVs

Symbol	Details
	**3/2 way Single solenoid valve (spring return)**
	**3/2 way pilot operated single solenoid valve(spring return)**
	**5/2 way single solenoid Valve (spring return)**
	**5/2 way double solenoid valve.**
	**5/2 way piloted operated double solenoid valve.**

1.2.5 Relays

A relay is an electro magnetically actuated switch. It is a simple electrical device used for signal processing. Relays are designed to withstand heavy power surges and harsh environment conditions. When a voltage is applied to the solenoid coil, an electromagnet field results. This causes the armature to be attracted to the coil core. The armature actuates the relay contacts, either closing or opening them, depending on the design. A return spring returns the armature to its initial position when the current to the coil is interrupted. Cross sectional view of a relay is shown in **Figure 1.8**

A large number of control contacts can be incorporated in relays in contrast to the case of a push button station. Relays are usually designated as K1, K2, and K3 etc. Relays also possess interlocking capability that is an important safety feature in control circuits. Interlocking avoids simultaneous switching of certain coils.

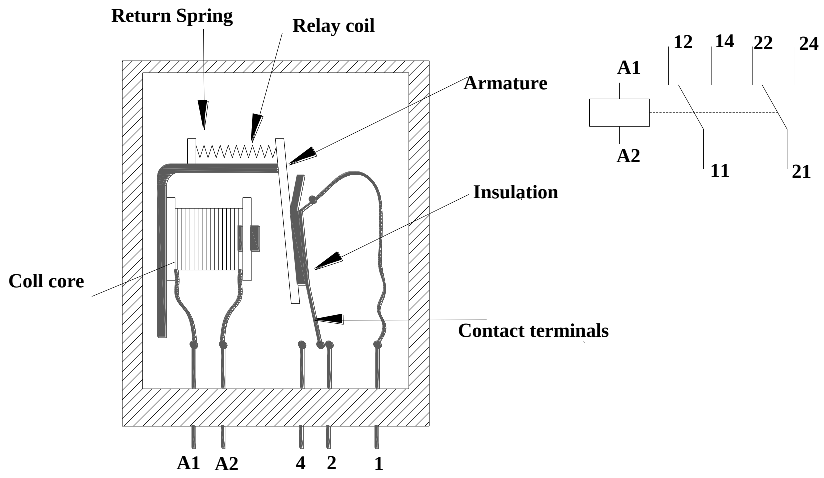


Figure 1.8: Cross sectional view of a relay

1.2.6 Timer Or Time delay relays

Timers are required in control systems to effect time delay between work operations. This is possible by delaying the operation of the associated control element through a timer. Most of the timers we use is Electronic timers. There are two types of time relay

i) Pull in delay (on –delay timer)

ii) Drop –out delay (off delay timer)

In the on-delay timer, shown in Figure cc, when push button PB is pressed (ON), capacitor C is charged through potentiometer R1 as diode D is reverse –biased. The time taken to charge the capacitor, depends on the resistance of the potentiometer (R1) and the capacitance(C) of the capacitor. By adjusting the resistance of the potentiometer, the required time delay can be set. When the capacitor is charged sufficiently, coil K is energised, and its contacts are operated after the set time delay. When the push button is released (OFF), the capacitor discharges quickly through a small resistance (R2) as the diode by passes resistor R1, and the contacts of relay (K) return to their normal position without any delay.

In the off-delay timer, the contacts are operated without any delay when the push button is pressed (ON). The contacts return to the normal position after the set delay when the push button is released (OFF).

The construction and symbols of the on-delay and off-relay timers are given in **Figure1.9**. Timing diagram is shown in **Table 1.2**

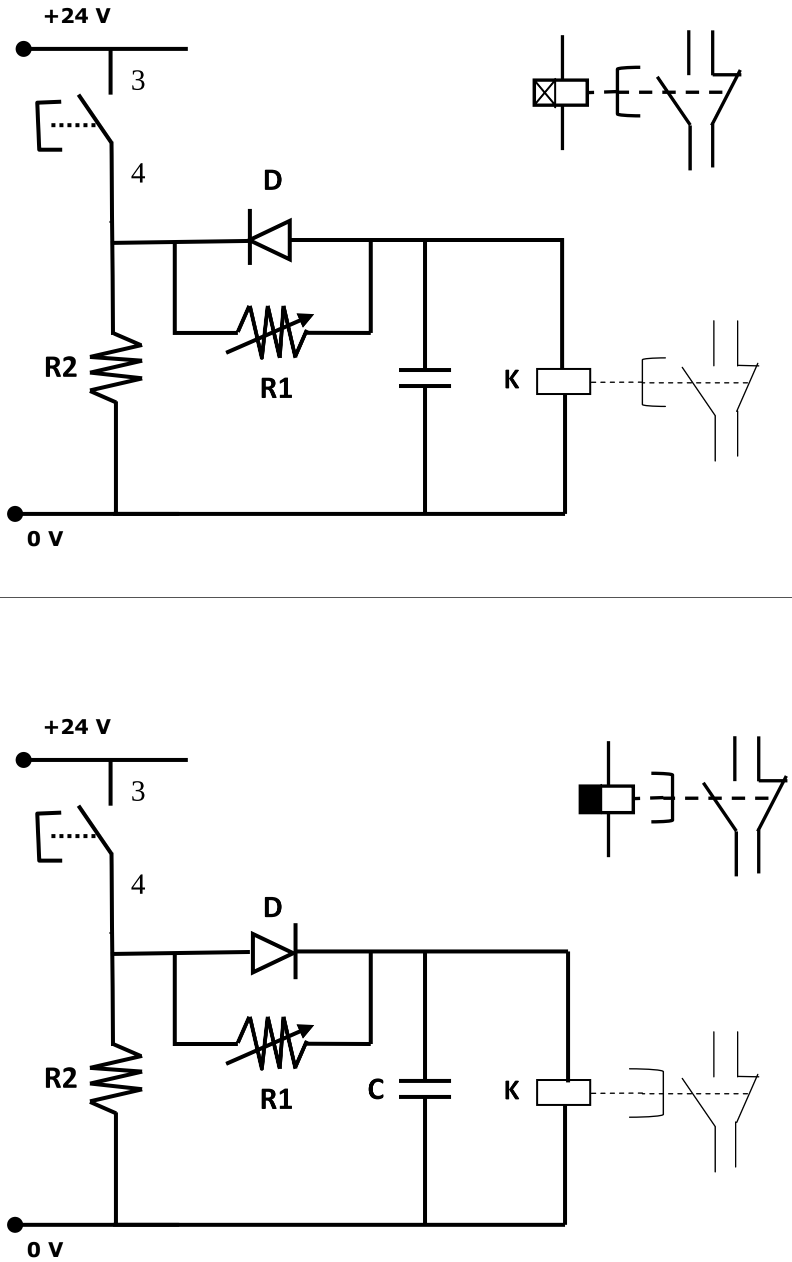
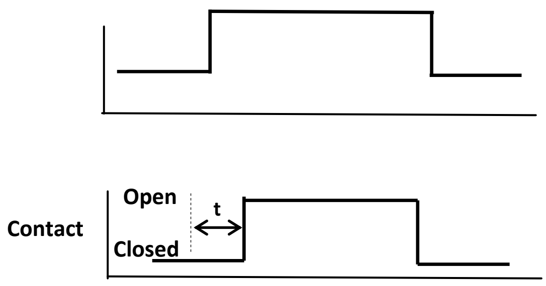
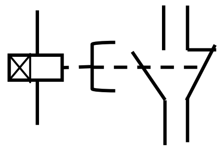
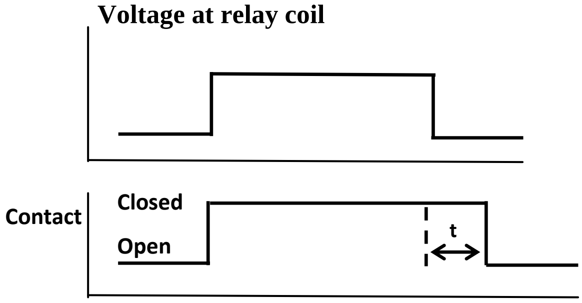
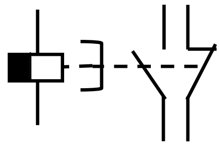


Figure 1.9: Construction features of timer and its symbols

Table 1.2: Timing diagram for on and off delay timer

Timing diagram	Symbol
Voltage at relay coil 	**On-delay timer circuit** 
Voltage at relay coil 	**Off-delay timer circuit** 

1.2.7 Temperature Switch

Temperature switches automatically senses a change in temperature and opens or closes an electrical switch when a predetermined temperature is reached. This switch can be wired either normally open or normally closed.

Temperature switches can be used to protect a fluid power system from serious damage when a component such as a pump or strainer or cooler begins to malfunction.

1.2.8 Reed Proximity switches

Reed switches are magnetically actuated proximity switches. Reed switches are similar to relays, except a permanent magnet is used instead of a wire coil. Schematic diagram of reed switch is shown in **Figure 1.1.** The reed switches comprise two ferromagnetic reeds placed with a gap in between and hermetically sealed in a glass tube. The glass tube is filled with inert gas to prevent the activation of the contacts. The surfaces of the reed contacts are plated with rhodium or iridium. Whole unit is encapsulated in epoxy resin to prevent mechanical damage to the switch. They are also provided with LED indicator to show its switching status.

When the magnet is away the switch is open, but when the magnet is brought near the switch is closed. The reed switch is operated by the magnetic field of an energized coil or a permanent magnet which induces north (N) and south (S) poles on the reeds. The reed contacts are closed by this magnetic attractive force. When the magnetic field is removed, the reed elasticity causes the contacts to open the circuit. The transfer type reed switch is normally ON, due to mechanical bias of the common (COM) lead, which is between the normally closed (N.C) reed contact and the normally open (N.O) reed contacts. When an external magnetic field is induced, the N.C blade is not affected because it is non-magnetic but the COM lead is attracted by the N.O lead and moves. When the magnetic field is removed, COM lead again moves to the N.C lead by mechanical.

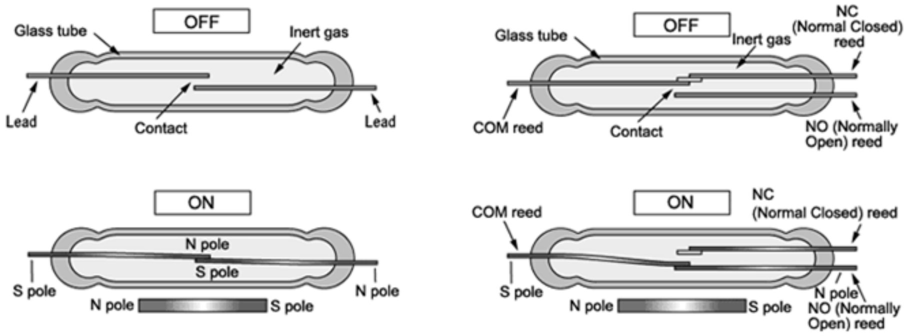


Figure 1.10: Construction features of reed switch

The two wire reed switch consists of two reeds. One of reed is connected to positive terminal of electric supply and other is connected to signal output. The three wire reed wire consists of three reed contacts. One is connected to positive terminal of electric supply. Second one is connected to negative terminal of the electric supply and third one is connected to the signal output. Symbol of the three wire reed switch and two wire reed switch is shown in Figure 1.11

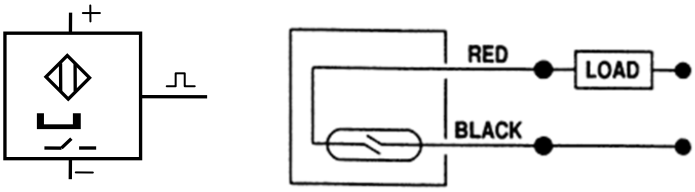


Figure 1.11: Symbol of 3 wire and 2 wire reed switch

Advantages of reed switches are

1. Reed switches are cheap.
2. They have long service life
3. They have shorter switching time (in the order of 0.2 to 0.3 millis seconds)
4. They are compact and maintenance frees

Disadvantages of reed switches are

1. They cannot be used in environments subjected to magnetic fields (like resistance welding machine)
2. Closing of contacts in reed switch is not bounce free

1.2.9 Electronic sensors

Inductive, Optical and capacitive proximity switches are electronic sensors. They normally have three electrical contacts. One contact for supply voltage , other for ground and third for output signal.

In these sensors, no movable contact is switched. Instead, the output is either electrically connected to supply voltage or to ground. There are two types of electronic sensors with regard to the polarity of output voltage.

Positive switching sensors: In this output voltage is zero if no part is detected in the proximity. The approach of a work piece or machine part leads to switch over of the output, applying the supply voltage.

Negative switching sensors: In this the supply voltage are applied to the output if no part is detected in the proximity. The approach of a work piece or machine part leads to switch over of the output, switching the output voltage to 0 volts.

a) Inductive sensors

Inductive sensor use currents induced by magnetic field to detect the nearby metal objects. The inductive sensor uses a coil or inductor to generate a high frequency magnetic field as shown in **Figure 1.12**. If there is a metal object near the changing magnetic field, current will flow in the object. This resulting current flow sets up a new magnetic field that opposes the original magnetic field. The net effect is that it changes the inductance of the coil in the inductive sensor. By measuring the inductance the sensor can determine when a metal have been brought nearby.

These sensors will detect any metals, when detecting multiple types of metal multiple sensors are used. In addition to metals, graphite also can be sensed. It is important to note that these work by setting up a high frequency field. If a target nears the field will induce eddy currents. These currents consume power because of resistance, so energy is in the field is lost, and the signal amplitude decreases . The detector examines filed magnitude to determine when is has decreased enough to switch.

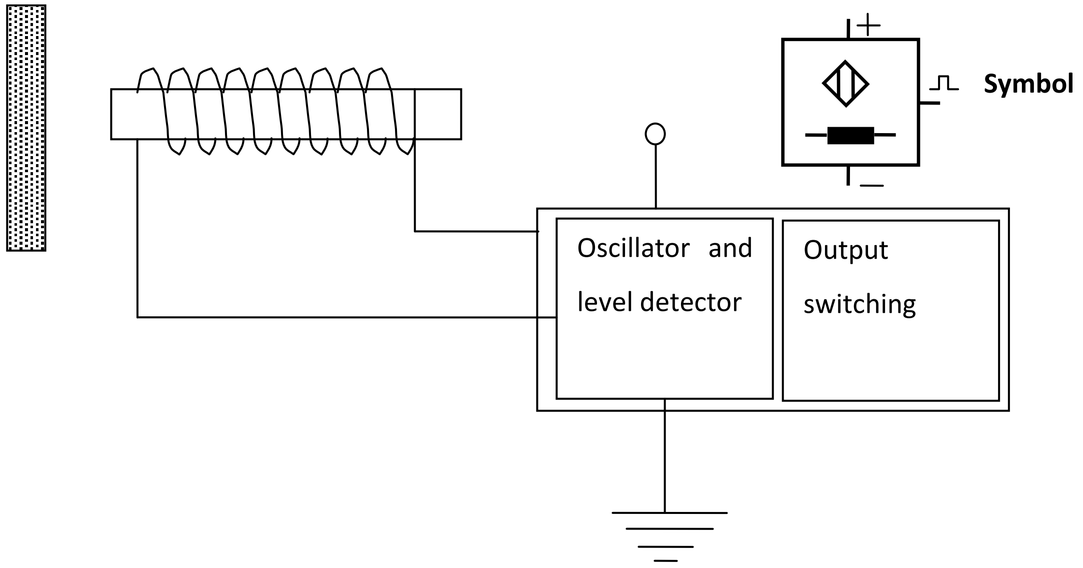


Figure 1.12: Inductive sensor

The sensors can detect objects a few centimetres away from the end. But, the direction to the object can be arbitrary as shown in **Figure 1.13**. The magnetic field of the unshielded sensors covers a large volume around the head of the coil. By adding a shield (a metal jacket around the sides of the coil) the magnetic field becomes smaller, but also more directed. Shields will often be available for inductive sensors to improve their directionality and accuracy.

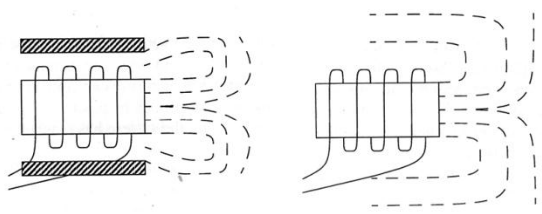


Figure 1.13: Shielded versus unshielded Inductive sensors

Advantages of proximity sensors are

1. They are self contained, rugged and extremely reliable
2. They have long service life
3. They have shorter switching time
4. They are compact and maintenance frees

Disadvantages of proximity sensors are

1. Like reed switches , they cannot be used in environments subjected to magnetic fields (like resistance welding machine)

Applications of proximity sensors

The proximity sensors can be used for various applications, These include:

- Sensing of end position of linear actuators like cylinders and semi rotary actuators
- They are used to detect metallic pieces on conveyor. That is presence or absence of work piece on conveyor
- They are used in press to detect the end position
- They are used to monitor drill breakage while drilling.
- They are also used as feed back devices in speed measuring devices

Factors influences the sensing distance

The switching distance of inductive sensors depends on the conductivity and permeability of the metal part whose presence or absence to be detected. This distance varies with the material composition of the target object, with mild steel takes as the material for standard reference (**Table 1.3**.) This is described by the reduction factor. The reduction factor is the factor by which the sensing range of the inductive sensor is reduced based on material composition of the objected to be sensed , compared to steel [FE 360] as the standard reference .

Table 1.3 Reduction factors for various materials

Material	**Reduction factor**
Stainless steel	0.80 to 0.85
Nickel steel	0.70 to 0.90
Aluminium and brass	0.35 -0.50
Copper	0.25 -0.40

Another factor which affects the sensing range of inductive sensors is the diameter of sensing coil. A small sensor with a coil diameter 0f 18mm has a typical range of 1mm, while a large sensor with core diameter of 75mm has a sensing range up to 50mm or even more

b) Capacitive sensors

Capacitive sensors are able to detect most materials at distances upto a few centimetres.

We know that

$$capacitance = \frac{Area\ of\ plates\ \times dielectric\ constant}{distance\ between\ plates\ (\ electrodes)}$$

In the sensor the area of the plates and distance between them is fixed. But, the dielectric constant of the space around them will vary as different material is brought near the sensor. An illustration of a capacitive sensor is shown **Figure 1.14** an oscillating field is used to determine the capacitance of the plates. When this changes beyond selected sensitivity the sensor output is activated.

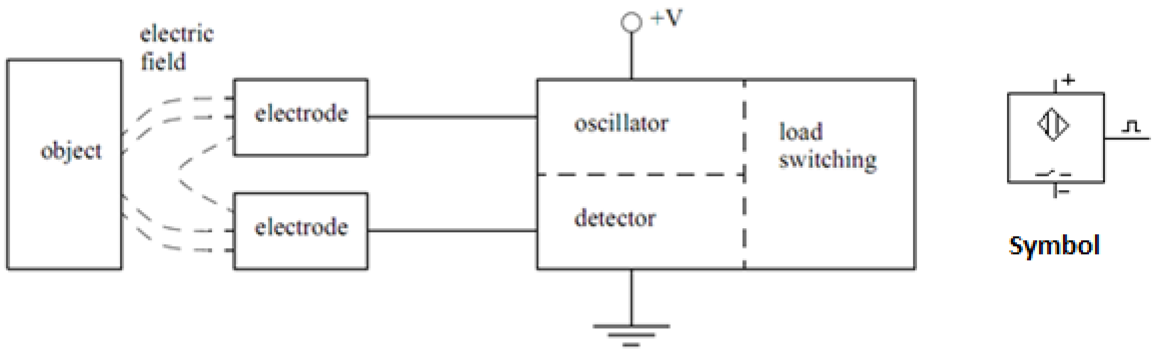


Figure 1.14: Capacitive sensors

For capacitive sensor the proximity of any material near the electrodes will increase the capacitance. This will vary the magnitude of the oscillating signal and the detector will decide when this is great enough to determine proximity.

These sensors work well for insulators (such as plastics) that tend to have high dielectric coefficients, thus increasing the capacitance. But, they also work well for metals because the conductive materials in the target appear as larger electrodes, thus increasing the capacitance as shown in **Figure 1.15**. In total the capacitance changes are normally in the order of pF.

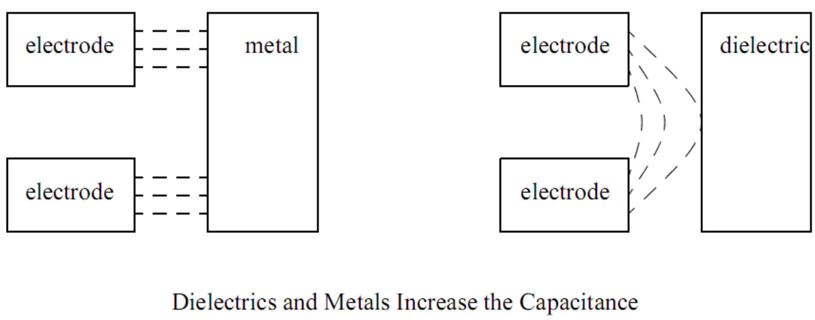


Figure 1.15: Capacitive sensors for metals and dielectrics

Advantages of proximity sensors are

1. They are widely used because of their ability to react with wide range of materials
2. They are suitable for detecting non metallic objects
3. They can be used to sense and monitor level in storage containers

Disadvantages of proximity sensors are

1. They are sensitive especially in humid environment
2. Without the compensator ring, the sensor would be very sensitive to dirt, oil and other contaminants that might stick to the sensor.

c) **Optical proximity sensors**

Light sensors have been used for almost a century - originally photocells were used for applications such as reading audio tracks on motion pictures. But modern optical sensors are much more sophisticated

Optical sensors require both a light source (emitter) and detector. Emitters will produce light beams in the visible and invisible spectrums using LEDs and laser diodes. Detectors are typically built with photodiodes or phototransistors. The emitter and detector are positioned so that an object will block or reflect a beam when present. A basic optical sensor is shown in **Figure 1.16**

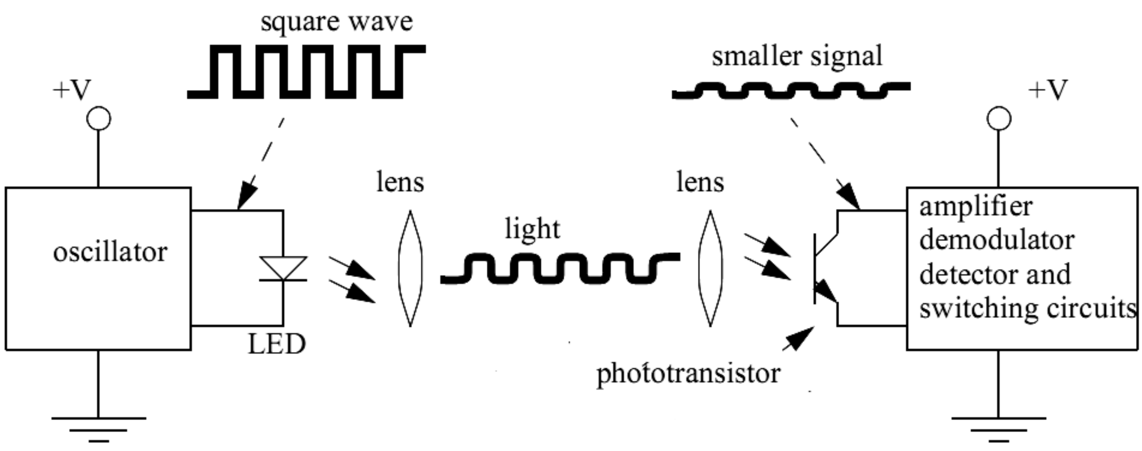


Figure 1.16: A Basic Optical sensor

In the figure the light beam is generated on the left, focused through a lens. At the detector side the beam is focused on the detector with a second lens. If the beam is broken the detector will indicate an object is present. The oscillating light wave is used so that the sensor can filter out normal light in the room. The light from the emitter is turned on and off at a set frequency. When the detector receives the light it checks to make sure that it is at the same frequency. If light is being received at the right frequency then the beam is not broken. The frequency of oscillation is in the KHz range, and too fast to be noticed. A side effect of the frequency method is that the sensors can be used with lower power at longer distances

An emitter can be set up to point directly at a detector, this is known as opposed mode. When the beam is broken the part will be detected. This sensor needs two separate components, as shown in **Figure 1.17** This arrangement works well with opaque and reflective objects with the emitter and detector separated by distances of up to hundreds of feet

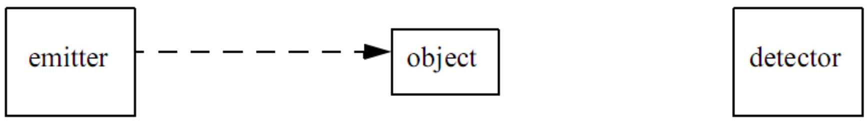


Figure 1.17: Opposed mode optical sensor

Having the emitter and detector separate increases maintenance problems and alignment is required. A preferred solution is to house the emitter and detector in one unit. But, this requires that light be reflected back as shown in **Figure 1.18**. These sensors are well suited to larger objects up to a few feet away.

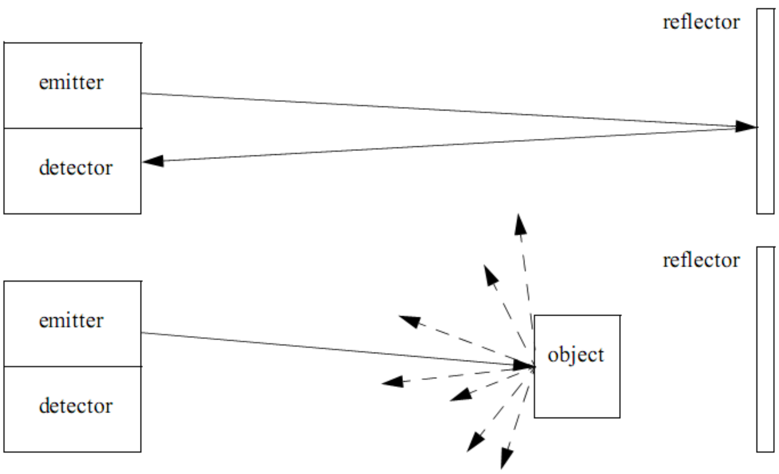


Figure 1.18: Emitter and detector in one unit

The reflector is constructed with polarizing screens oriented at 90 deg. angles. If the light is reflected back directly the light does not pass through the screen in front of the detector. The reflector is designed to rotate the phase of the light by 90 deg., so it will now pass through the screen in front of the detector.

In the figure, the emitter sends out a beam of light. If the light is returned from the reflector most of the light beam is returned to the detector. When an object interrupts the beam between the emitter and the reflector the beam is no longer reflected back to the detector, and the sensor becomes active. A potential problem with this sensor is that reflective objects could return a good beam. This problem is overcome by polarizing the light at the emitter (with a filter), and then using a polarized filter at the detector. The reflector uses small cubic reflectors and when the light is reflected the polarity is rotated by 90 degrees. If the light is reflected off the object the light will not be rotated by 90 degrees. So the polarizing filters on the emitter and detector are rotated by 90 degrees, as shown in **Figure 1.19** The reflector is very similar to reflectors used on bicycles

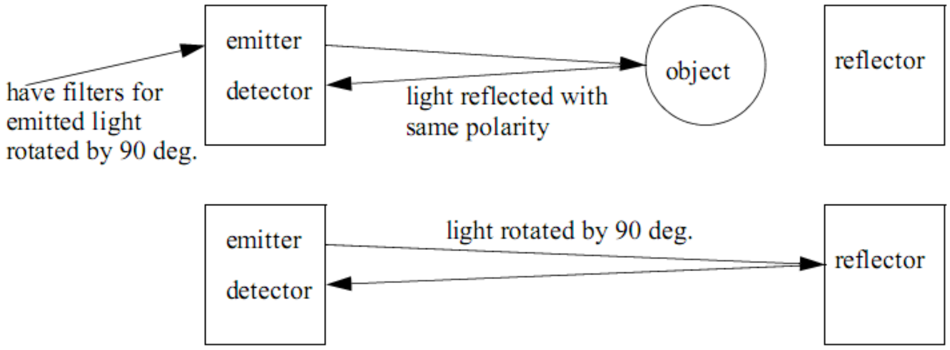


Figure 1.19: Polarized light in retroreflective sensors

For retro reflectors the reflectors are quite easy to align, but this method still requires two mounted components. A diffuse sensors is a single unit that does not use a reflector, but uses focused light as shown in **Figure 1.20**

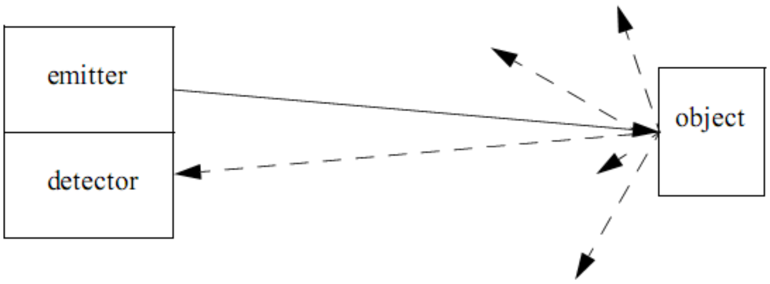


Figure 1.20: Diffuse sensor

With diffuse reflection the light is scattered. This reduces the quantity of light returned. As a result the light needs to be amplified using lenses

Diffuse sensors

Diffuse sensors use light focused over a given range, and a sensitivity adjustment is used to select a distance. These sensors are the easiest to set up, but they require well controlled conditions. For example if it is to pick up light and dark colored objects problems would result.

When using opposed mode sensors the emitter and detector must be aligned so that the emitter beam and detector window overlap, as shown in **Figure 1.21**. Emitter beams normally have a cone shape with a small angle of divergence (a few degrees of less). Detectors also have a cone shaped volume of detection. Therefore when aligning opposed mode sensor care is required not just to point the emitter at the detector, but also the detector at the emitter. Another factor that must be considered with this and other sensors is that the light intensity decreases over distance, so the sensors will have a limit to separation distance

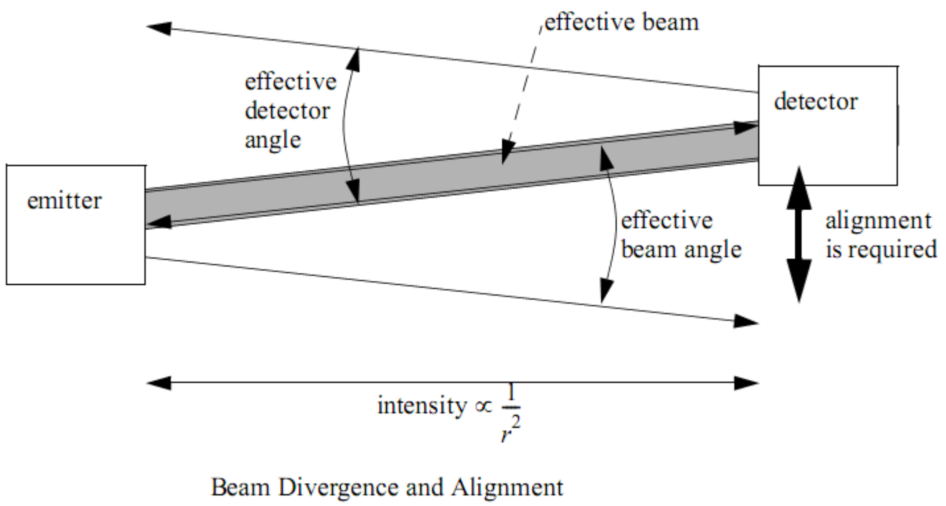


Figure 1.21: Beam divergence and alignment

If an object is smaller than the width of the light beam it will not be able to block the beam entirely when it is in front as shown in **Figure 1.22.** This will create difficulties in detection, or possibly stop detection altogether. Solutions to this problem are to use narrower beams, or wider objects. Fiber optic cables may be used with an opposed mode optical sensor to solve this problem, however the maximum effective distance is reduced to a couple feet

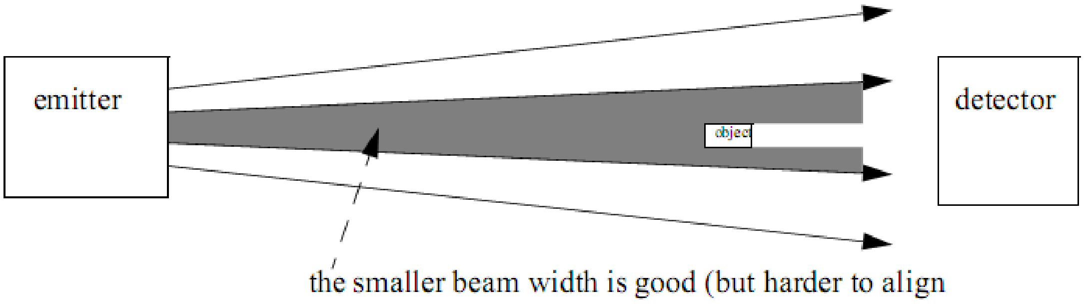


Figure 1.22: The relationship between beam width and object size

Separated sensors can detect reflective parts using reflection as shown in **Figure 1.23**. The emitter and detector are positioned so that when a reflective surface is in position the light is returned to the detector. When the surface is not present the light does not return

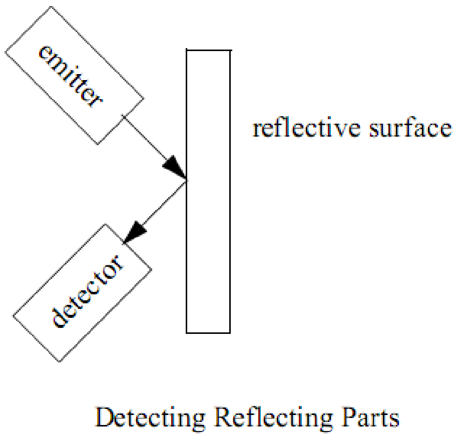


Figure 1.23: separated sensors

Other types of optical sensors can also focus on a single point using beams that converge instead of diverge.(**Figure 1.24**) The emitter beam is focused at a distance so that the light intensity is greatest at the focal distance. The detector can look at the point from another angle so that the two centerlines of the emitter and detector intersect at the point of interest. If an object is present before or after the focal point the detector will not see the reflected light. This technique can also be used to detect multiple points and ranges, as shown in **Figure 1.25** where the net angle of refraction by the lens determines which detector is used. This type of approach, with many more detectors, is used for range sensing systems

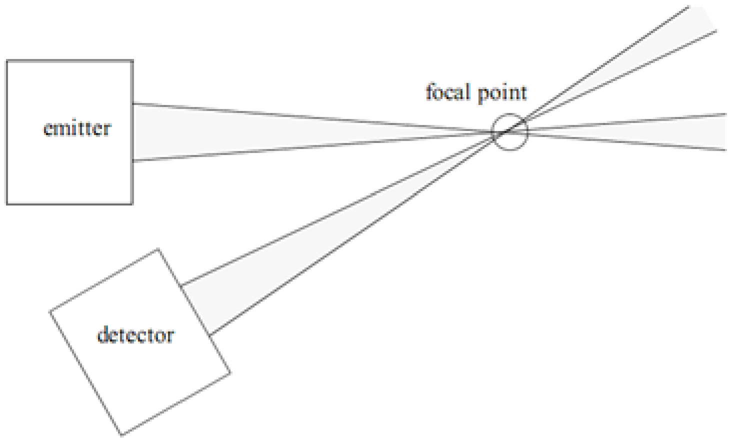


Figure 1.24: Point detection using focused optics

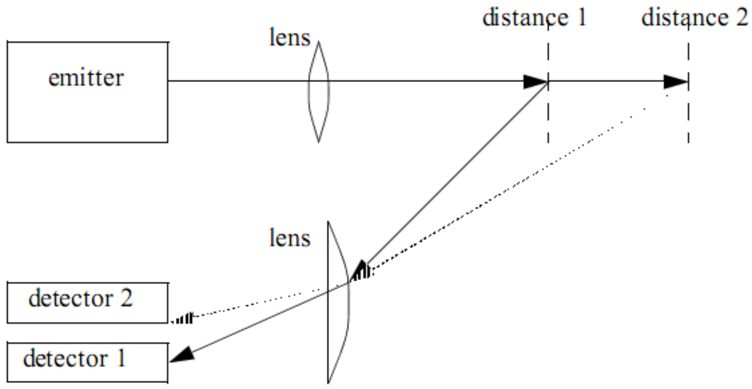


Figure 1.25: Multiple point detection using optics

Some applications do not permit full sized photo optic sensors to be used. Fiber optics can be used to separate the emitters and detectors from the application. Some vendors also sell photosensors that have the phototransistors and LEDs separated from the electronics.

1.2.10 Electric counters

An electric counter consist of a coil, associated circuits and contacts, a reset coil, manual reset, release button and a display window. Pressure the release button of the counter and entering the desired count valve set the pre-determining counter. The pre-determined count valve is displayed in the window. There are two types of counters

1. Up counter
2. Down counter

Up counter: An up counter counts electrical signal upwards from zero. For each electrical counting pulse input to an up-counter coil, the counter value is incremented by 1. When the predetermined valve has been reached, the relay picks up and the contact set is actuated.

Down counter: An down counter counts electrical signal downwards from preset valve. If the count valves of zero is reached the relay picks up and the contact set is actuated. The counter can be reset manually by pressing the reset button or electrically by applying a reset pulse to the reset coil. The pre-determined value is maintained when the counter is reset. The symbols of up counter and down counter is shown in **Figure 1.26(a) and Figure 1.26(b)**

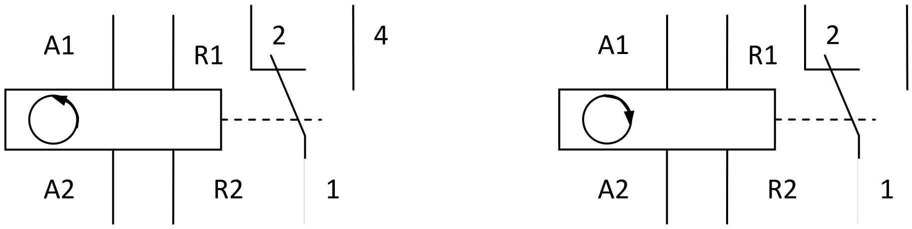


Figure 1.26: a) up counter b) down counter

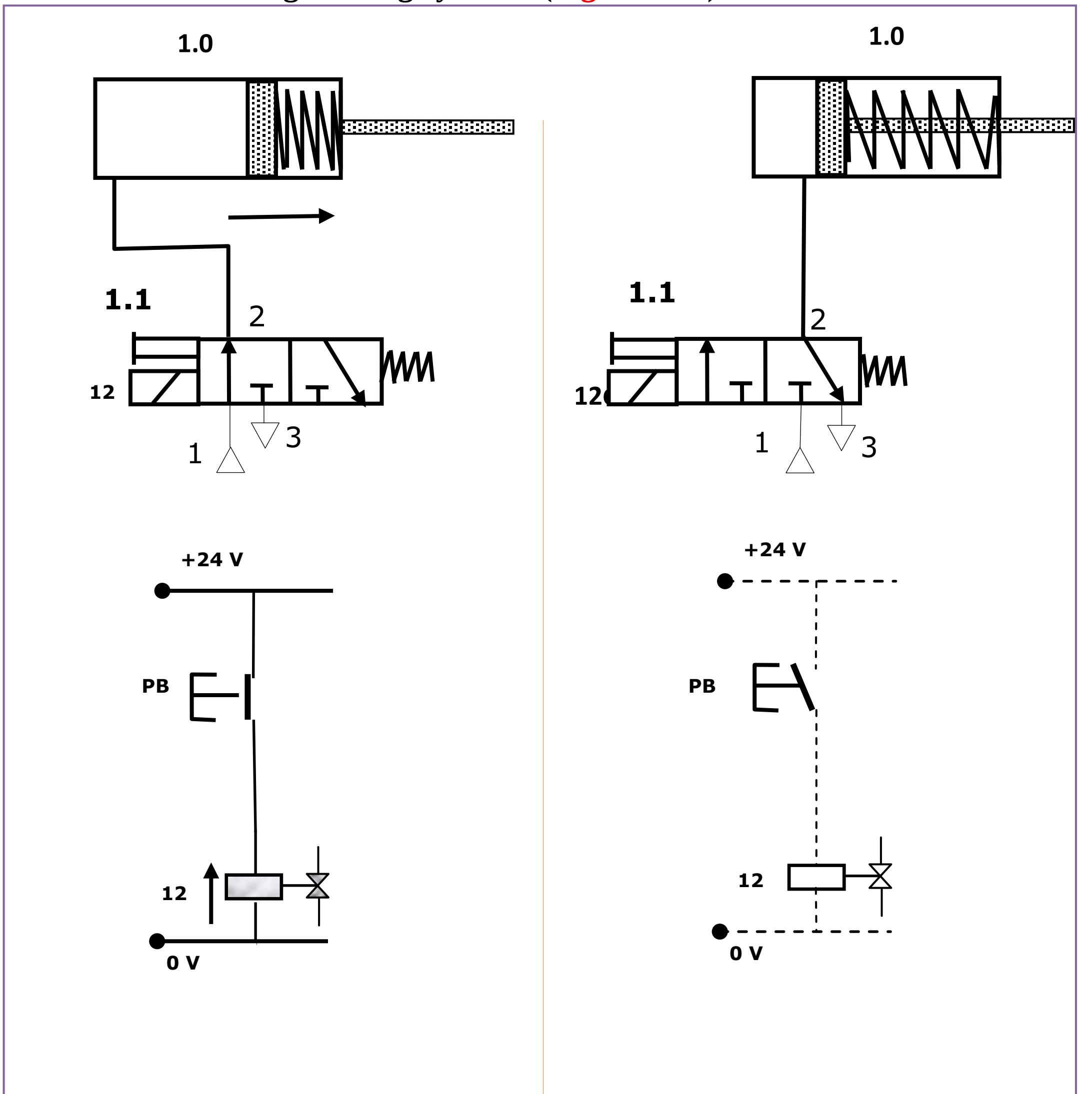
1.3 Electro pneumatics circuits for single actuator

Control of single cylinders using electro pneumatics

Forward stroke: The circuit is closed when push button PB closes. A magnetic field is produced in the coil Y. The armature in the coil opens the passage for the compressed air. The compressed air flows from 1 to 2 of the 3/2 DCV to cylinder, which travels to the final forward position.

Return stroke: When the push button PB is released, the circuit is interrupted. The magnetic field at coil Y collapses, the 3/2 way valve switches back to its original position as shown in **Figure 1.27.** The compressed air in the cylinder then exhausts through port 3 of the DCV and the cylinder travel to the final rear position.

10.3.1 Direct Control of single acting cylinder (Figure 1.27)



Position when cylinder is extended

Position when cylinder is retracted

Figure 1.27: Direct control of single acting cylinder

1.3.2. Indirect Control of single acting cylinder (Figure 1.28)

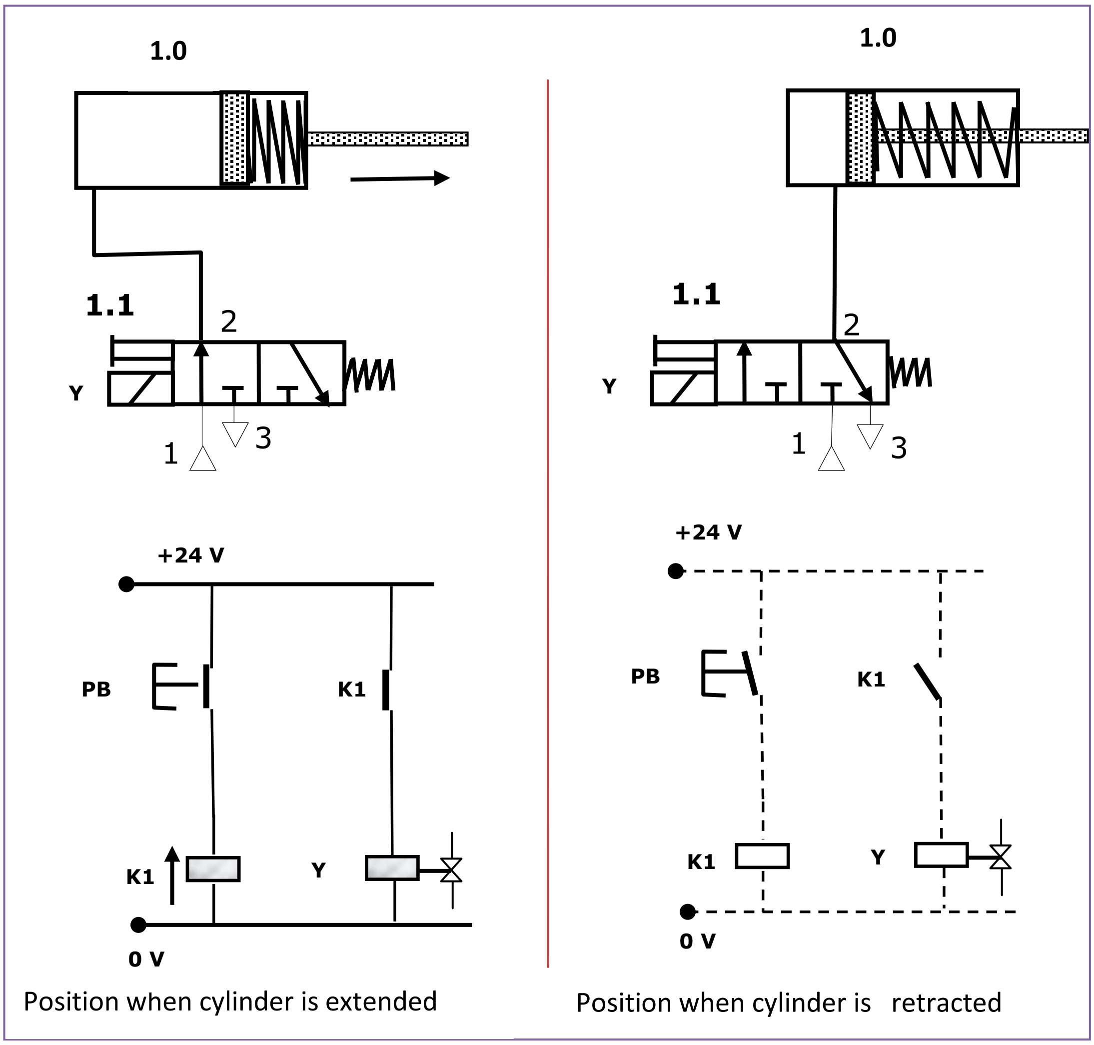


Figure 1.28: Indirect control of single acting cylinder

Forward stroke: The circuit is closed when push button PB closes. Closing of Push button PB energises a relay K1. The coil Y is energised via normally open contact K1 (indirect energising).
A magnetic field is produced in armature of the coil Y opens the passage for the compressed air. The compressed air flows from 1 to 2 of the 3/2 DCV to cylinder, which travels to the final forward position.

Return stroke: When the push button PB is released, the circuit is interrupted. Opening of Push button PB de-energises a relay K1. The magnetic field at coil Y is collapses due to the opening of contact K1 the 3/2 way valve switches back to its original position as shown in Figure 1.28. The compressed air in the cylinder then exhausts through port 3 of the DCV and the cylinder travel to the final rear position.

1.3.3 Direct Control of double acting cylinder (Figure 1.29)

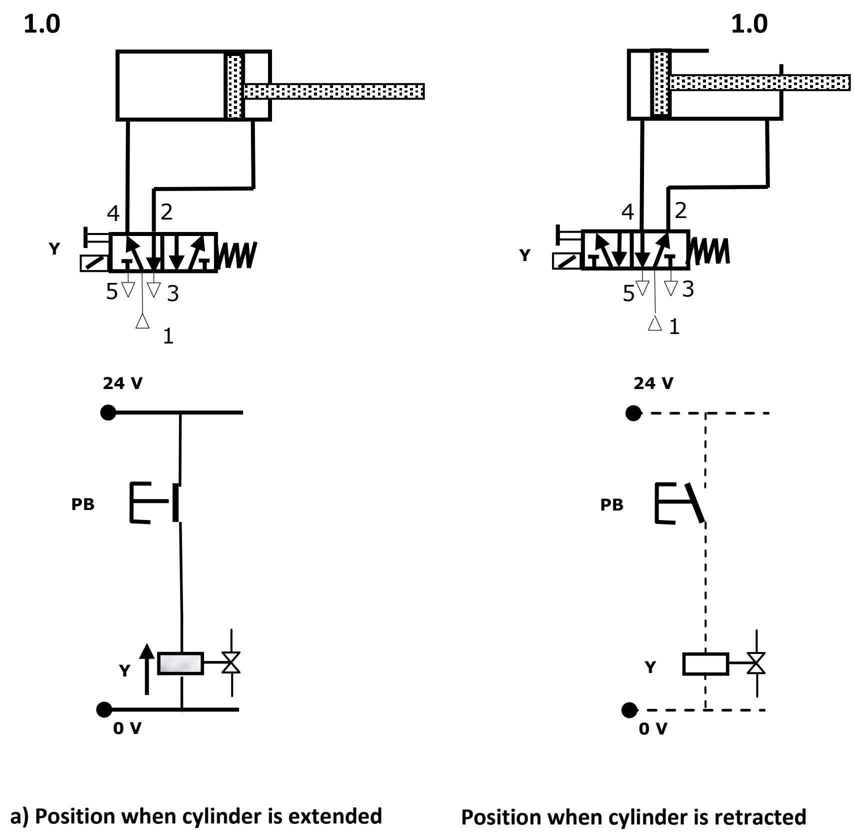


Figure 1.29: direct control of double acting cylinder

Forward stroke: The double acting cylinder is controlled by 5/2 way valve. When the pushbutton PB is pressed, coil Y is energised and the directional control valve is activated by compressed air via pilot control. The piston travels to the final forward position.

Return stroke: When the push button PB is released, the circuit is interrupted. The magnetic field at coil Y collapses, the return spring of 5/2 becomes active and the 5/2 way valve switches back to its original position as shown in **Figure 1.29**. The compressed air in the cylinder then exhausts through port 5 of the 5/2 DCV and the cylinder travel to the final rear position.

1.3.4 Indirect Control of double acting cylinder (using 5/2 way, single solenoid)

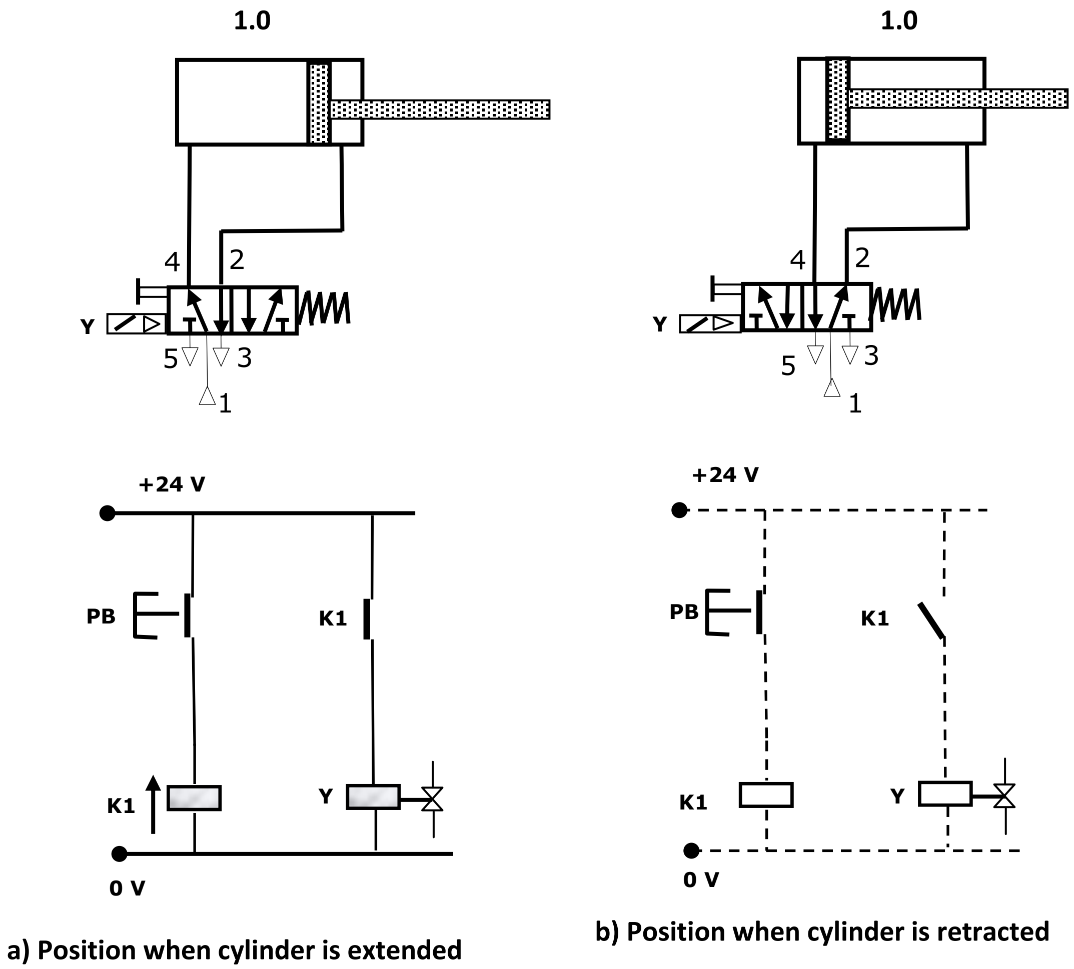
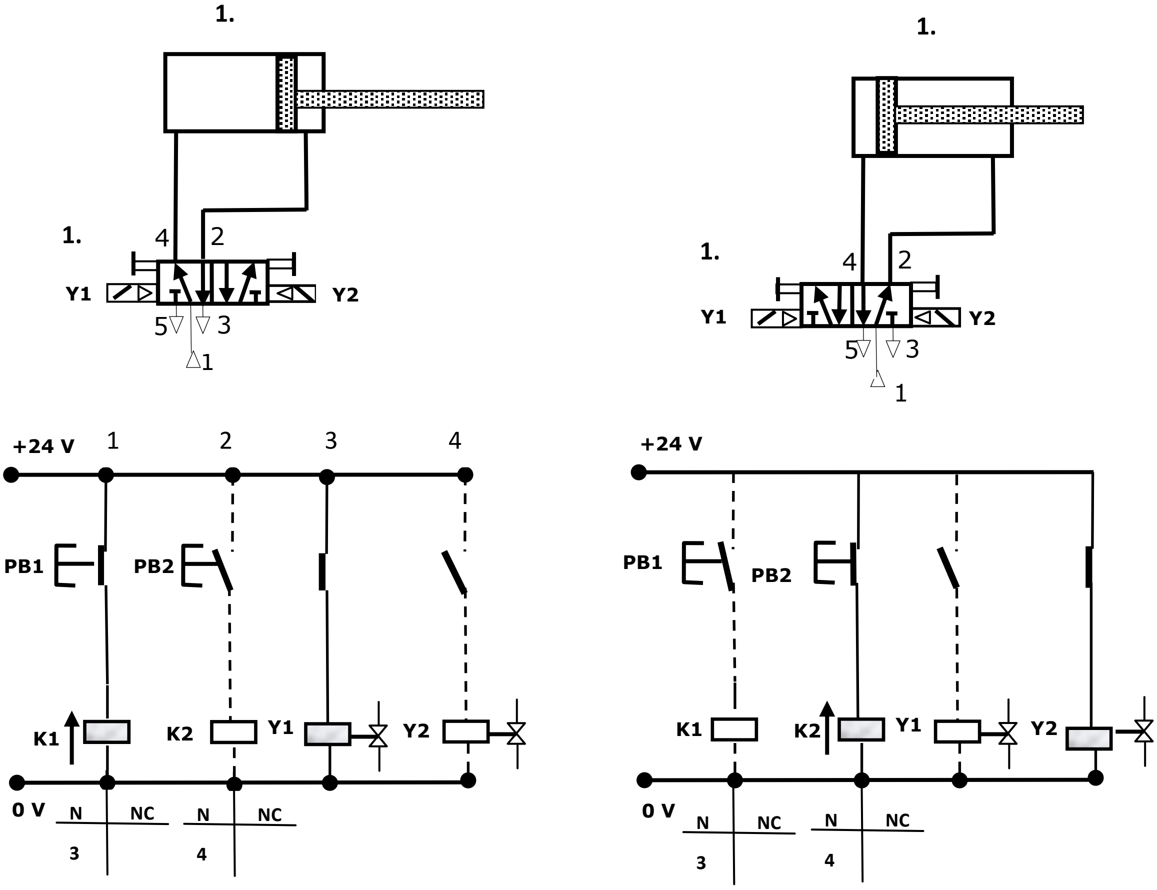


Figure 1.30: Indirect control of double acting cylinder

Forward stroke: The circuit is closed when push button PB closes. Closing of Push button PB energises a relay K1. The coil Y is energised via normally open contact K1 (indirect energising). A magnetic field is produced in armature of the coil Y opens the passage for the compressed air through internal pilot. The compressed air flows from 1 to 4 of the 5/2 DCV to cylinder, which travels to the final forward position.

Return stroke: When the push button PB is released, the circuit is interrupted. Opening of Push button PB de-energises a relay K1. The magnetic field at coil Y is collapses due to the opening of contact K1 the 5/2 way valve switches back to its original position as shown in **Figure 1.30**. The compressed air in the cylinder then exhausts through port 5 of the DCV and the cylinder travel to the final rear position.

1.3.5 Indirect Control of double acting cylinder (using 5/2 way, double solenoid) (Figure 1.31)



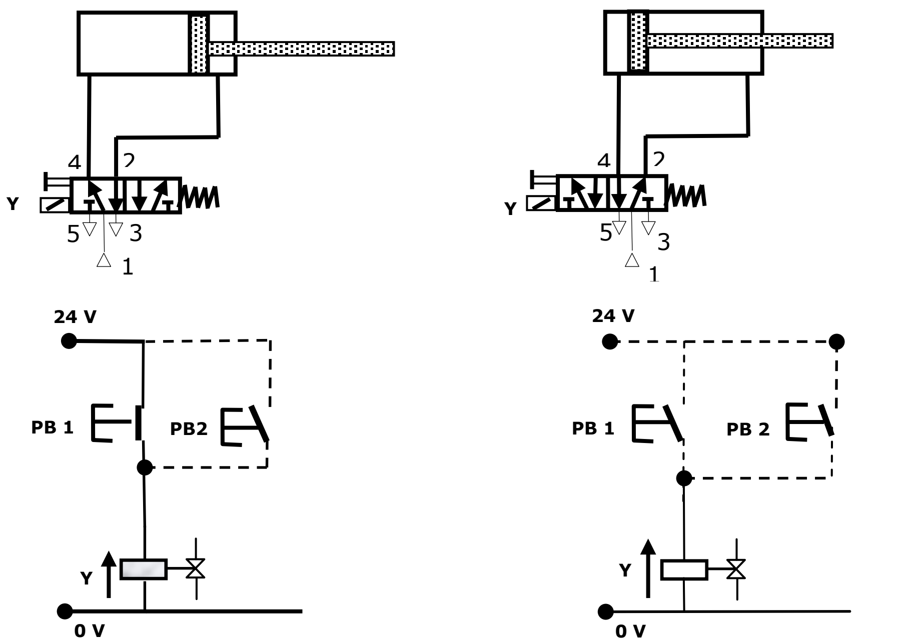
a) Position when cylinder is extended b) Position when cylinder is

Figure 1.31: Indirect control of double acting cylinder (using 5/2 way vavle)

Forward stroke: when push button PB1 is pressed, coil Y1 is energised and 5/2 way directional control valve changes over. The piston travels out and remains in the final forward position until a signal is applied to coil Y2. The 5/2 directional control valve will remain in the last position because it is double solenoid valve and has no return spring.

Return stroke: When the push button PB1 is released and PB2 is pressed. Opening of Push button PB1 de-energises a relay K1. The magnetic field at coil Y1 is collapses due to the opening of contact K1. Closing of PB2 energises Y2 and the piston returns to its original position as a result of changeover of the 5/2 way valve. The 5/2 way directional control valve will not switch over if there is an active opposing signal. For example, if Y1 is switched on and a signal is given to Y2, there will be no reaction as there would be an opposing signal

1.3.6 Control of double acting cylinder OR logic (Parallel circuit) (Figure 1.32)



a) Cylinder is extended using PB1

b) Cylinder is retracted (both PB released)b)

Figure 1.32: OR logic

The piston of a double acting cylinder is to travel out when either one of two pushbutton switch is pressed. It is to return when both are released. When push button PB1 or PB2 are pressed. Coil Y1 is energised. The directional control valve switches over and the piston travels to the final forward position. When both the push button switches are released, the signal is removed from Y1 and the cylinder travels back to its original position.

1.3.7 Control of double acting cylinder AND logic (Figure 1.33)

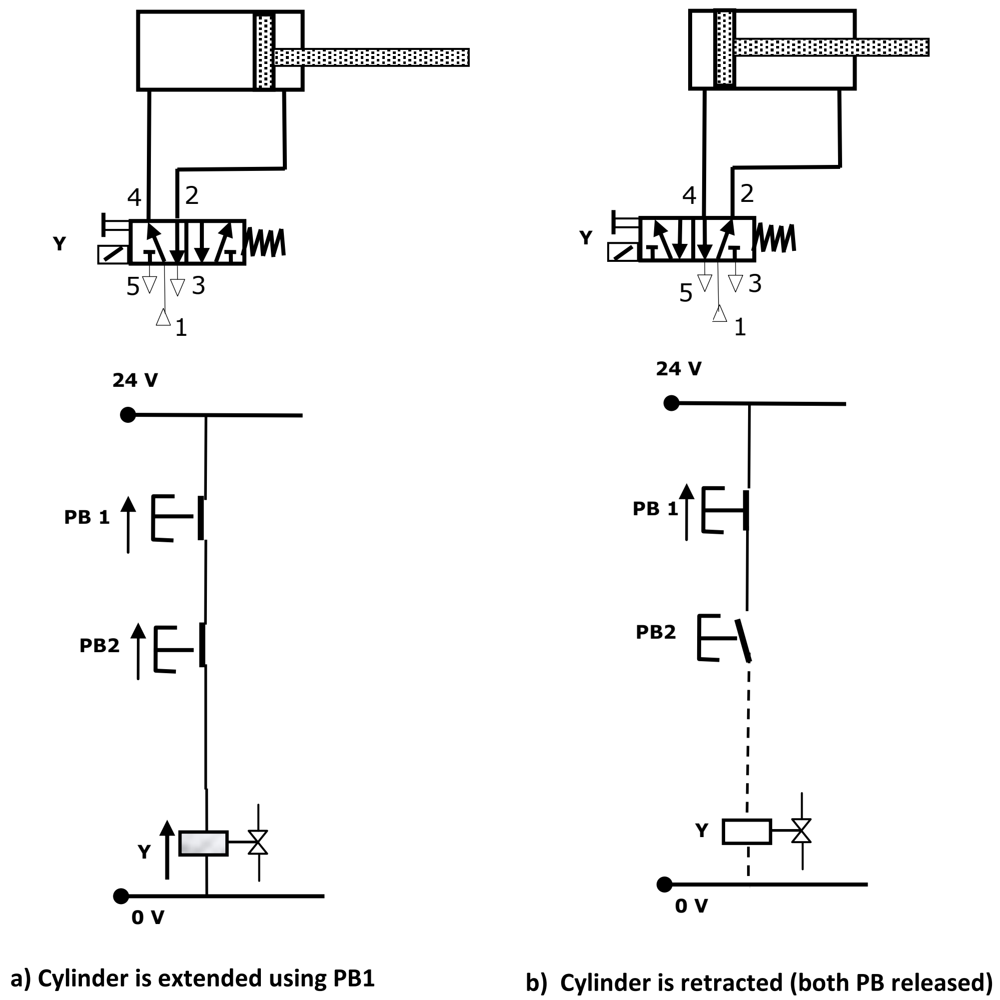


Figure 1.33: AND logic

1.3.8 Latching circuits

Double acting cylinder is to be controlled using 5/2 directional control valve , single solenoid, spring return. When push button PB1 is pressed, cylinder should extend and remains in that position when PB1 is released. The cylinder is to retract completely when PB2 is pressed. In addition, the cylinder is to remain in the retracted postion even when PB2 is released. Develop a Electro-pneumatic control circuit with an electrical latching with a) dominant off b) dominant On

Solution

In the following pneumatic circuit a double acting cylinder is controlled by 5/2 way valve.When Y1 is energised cylinder moves forward. When Y1 is deenergised cylinder retracts to its iniatil position.

We can construct the latching circui using the following electrical components

1. Use NO pustton button for ON or Start button control
2. Use NC push button for the OFF or stop control
3. Use a relay

Latching circuit can be dominatn ON or dominant Off. Dominnat position refer to stauts of relay coil (circuit) when both the start and stop signals are applied simultaneously

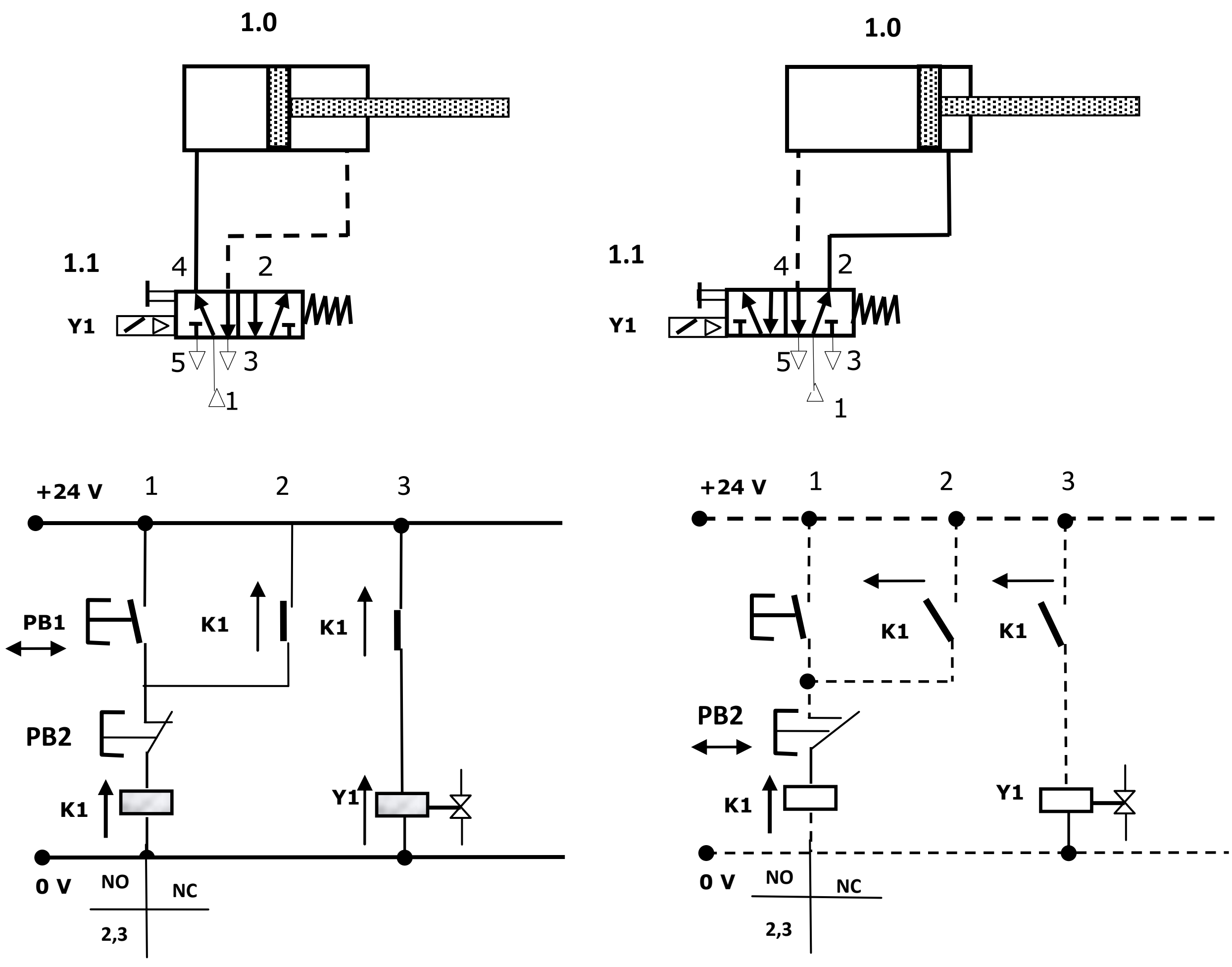
a) Latching circuit with Dominant OFF

When Start button (PB1) and Stop button (PB2) are pressed simultaneously, if the circuit goes to OFF position , then such a circuit is called Dominant OFF latching circuit. Refer to **Figure 1.34** ,

a) When we press START push button PB1 is pressed and released , following operations occurs:
   1. Relay coil K1 in branch 1 (vertical) is energised. All Contact K1 closes
   2. Notice that there is a NO contact of K1 in branch 2 , which is connected parallel to PB1. This NO contact of K1 latches the start push button. Therefore even if the PB1 is released, NO contact of K1 in branch 2 keeps coil K1 energised.
   3. There is another NO contact in branch 3, which is connected to Y1. When push button PB1 is pressed this also remain closed, as a result cylinder moves forward and remains there until stop button PB2 is pressed.
b) When we press STOP push button PB2 is pressed momentarily and released , following operations occurs:
   1. Relay coil K1 in branch 1 (vertical) is de-energised. All Contact K1 opens
   2. NO contact of K1 in branch 2 , which is connected parallel to PB1 is now open. This NO contact of K1 no more latches the start push button.

3. NO contact in branch 3 is also open now, which is denergises. As a resutl cylinder moves back to its home position and remains there until start button PB1 is pressed again.

When Start button (PB1) and Stop button (PB2) are pressed simultaneously, K1 contacts are open and the circuit goes to OFF position . That is why this circuit is called Dominant OFF latching circuit.



a) Circuit is in latched position

b) Circuit is in unlatched position

Figure 1.33: Dominant OFF ciruict

b) Latching circuit with Dominant ON

When Start button (PB1) and Stop button (PB2) are pressed simultaneously, if the circuit goes to ON position , then such a circuit is called Dominant ON latching circuit. Refer to **Figure 1.34** ,

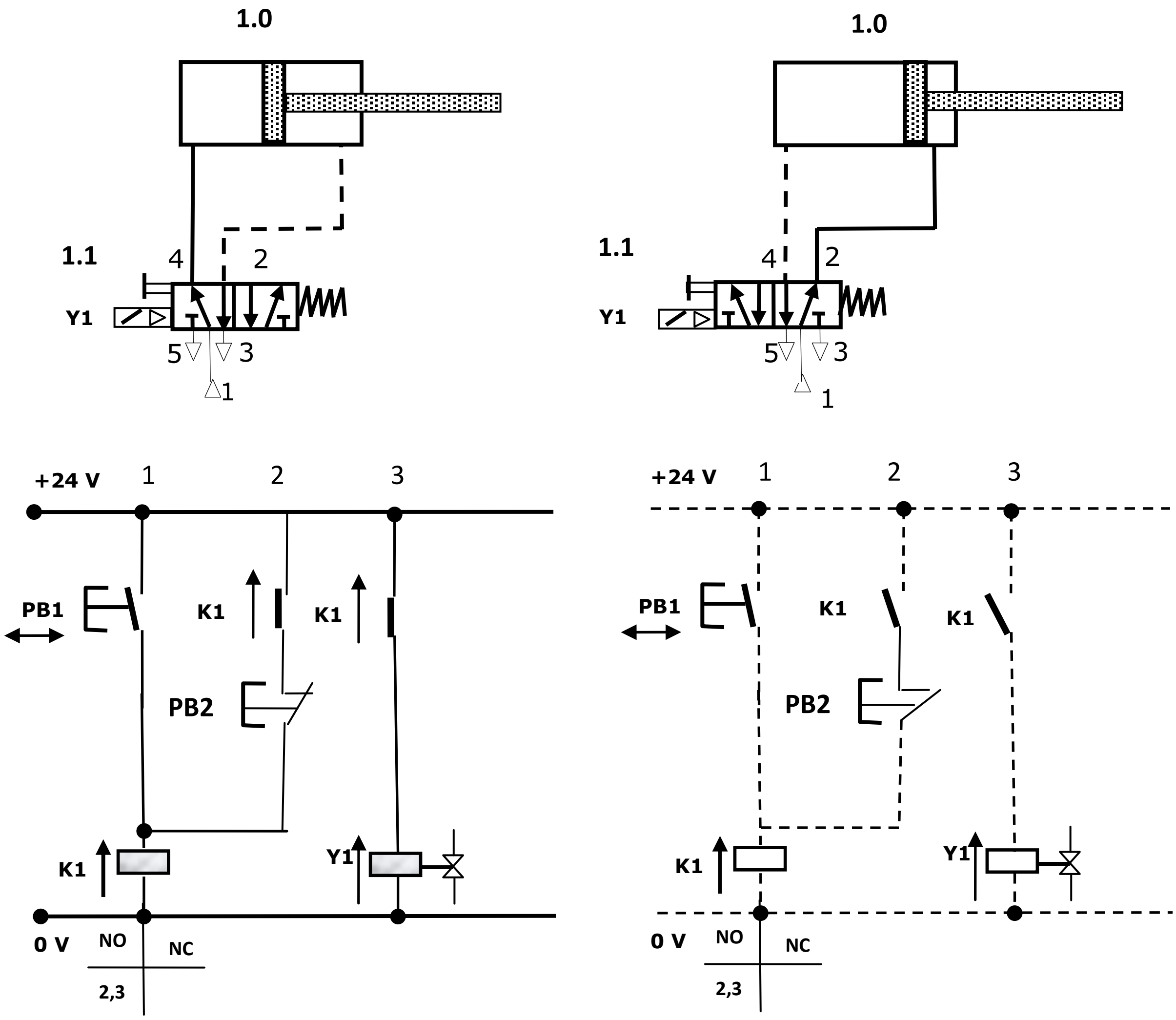
a) When we press START push button PB1 is pressed and released , following operations occurs:

   4. Relay coil K1 in branch 1 (vertical) is energised. All Contact K1 closes
   5. Notice that there is a NO contact of K1 in branch 2 , which is connected parallel to PB1 and in series with PB2. This NO contact of K1 latches the start push button. Therefore even if the PB1 is released, NO contact of K1 in branch 2 keeps coil K1 energised.
   6. There is another NO contact in branch 3, which is connected to Y1. When push button PB1 is pressed this also remain closed, as a result cylinder moves forward and remains there until stop button PB2 is pressed.

b) When we press STOP push button PB2 is pressed momentarily and released , following operations occurs:

   1. Relay coil K1 in branch 1 (vertical) is de-energised. All Contact K1 opens
   2. NO contact of K1 in branch 2 , which is connected parallel to PB1 is now open. This NO contact of K1 no more latches the start push button.
   3. NO contact in branch 3 is also open now, which is denergises. As a resutl cylinder moves back to its home position and remains in home position until start button PB1 is pressed again.

When Start button (PB1) and Stop button (PB2) are pressed simultaneously, K1 contacts are open and the circuit goes to OFF position . That is why this circuit is called Dominant OFF latching circuit.



a) Circuit is in latched position

b) Circuit is in unlatched position

Figure 1.34: Dominant ON circuit

1.3.9 Automatic return of a double acting cylinder (spring return) (Figure 1.35)

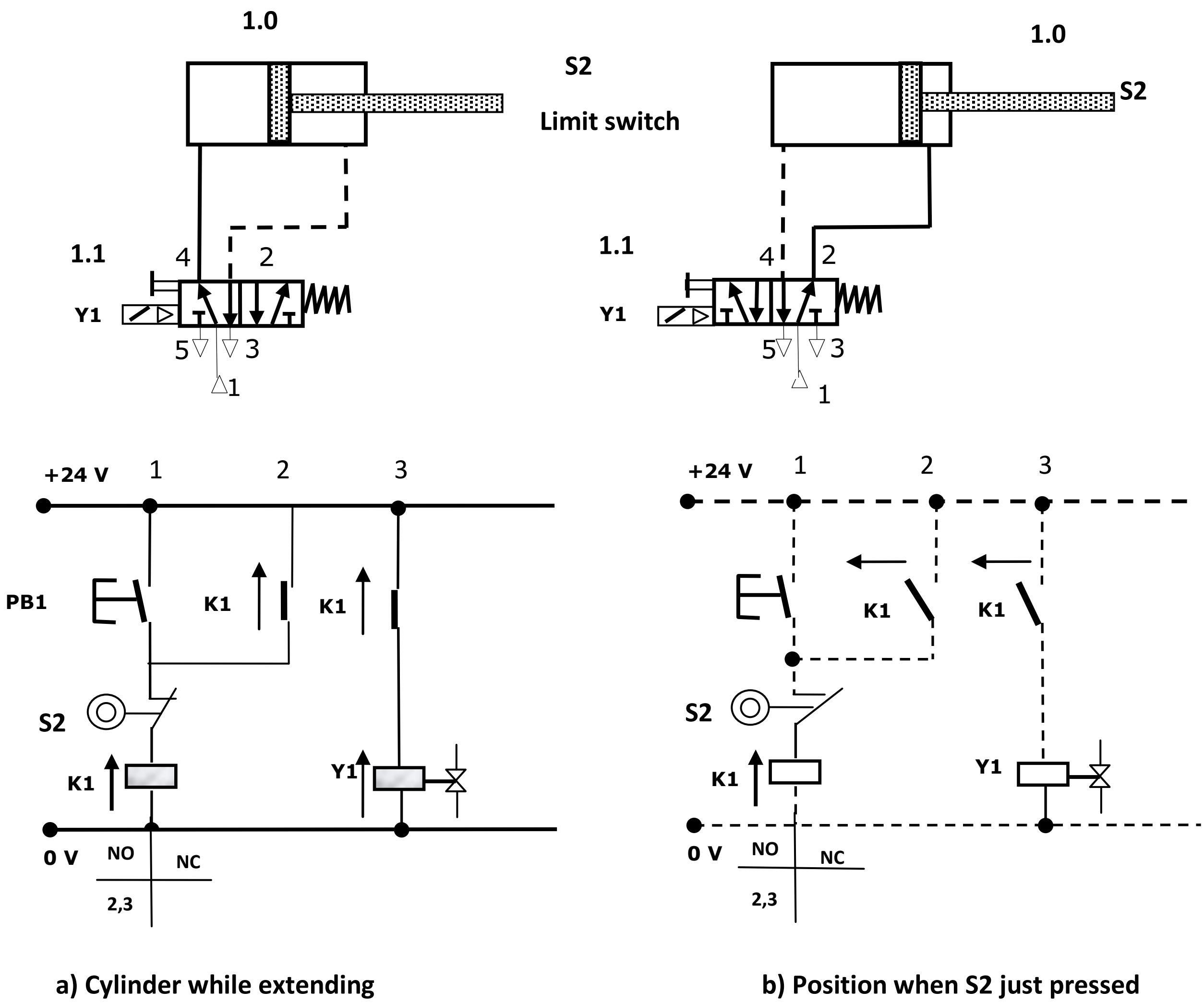


Figure 1.35: Automatic return of double acting cylinders using single solenoid

1.3.10 Direct control of automatic return of a double acting cylinder (double solenoid) (Figure 1.36)

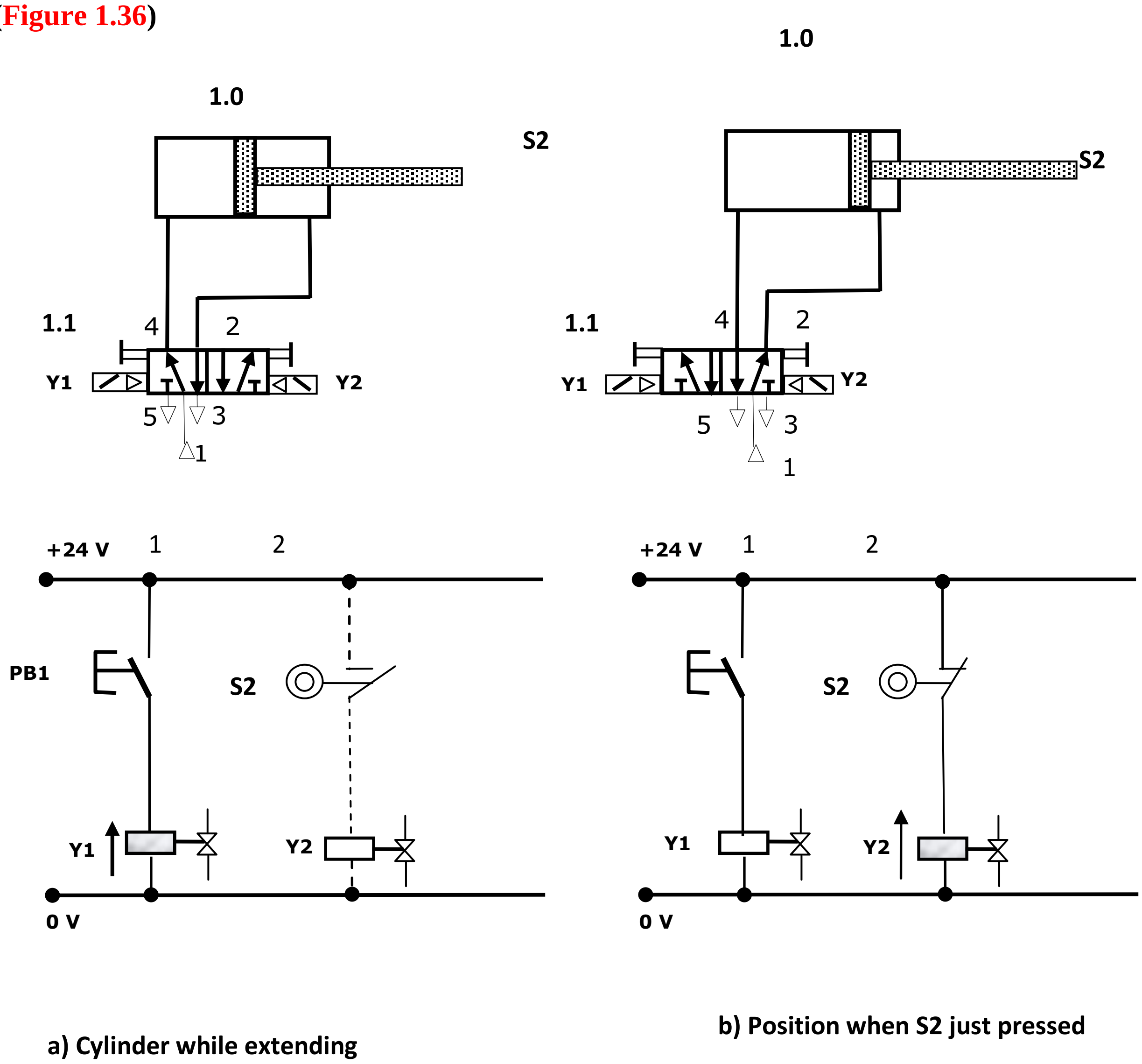


Figure 1.36: Automatic return of double acting cylinder using double solenoid

1.3.11 Indirect control of automatic return of a double acting cylinder (double solenoid) (Figure 1.37)

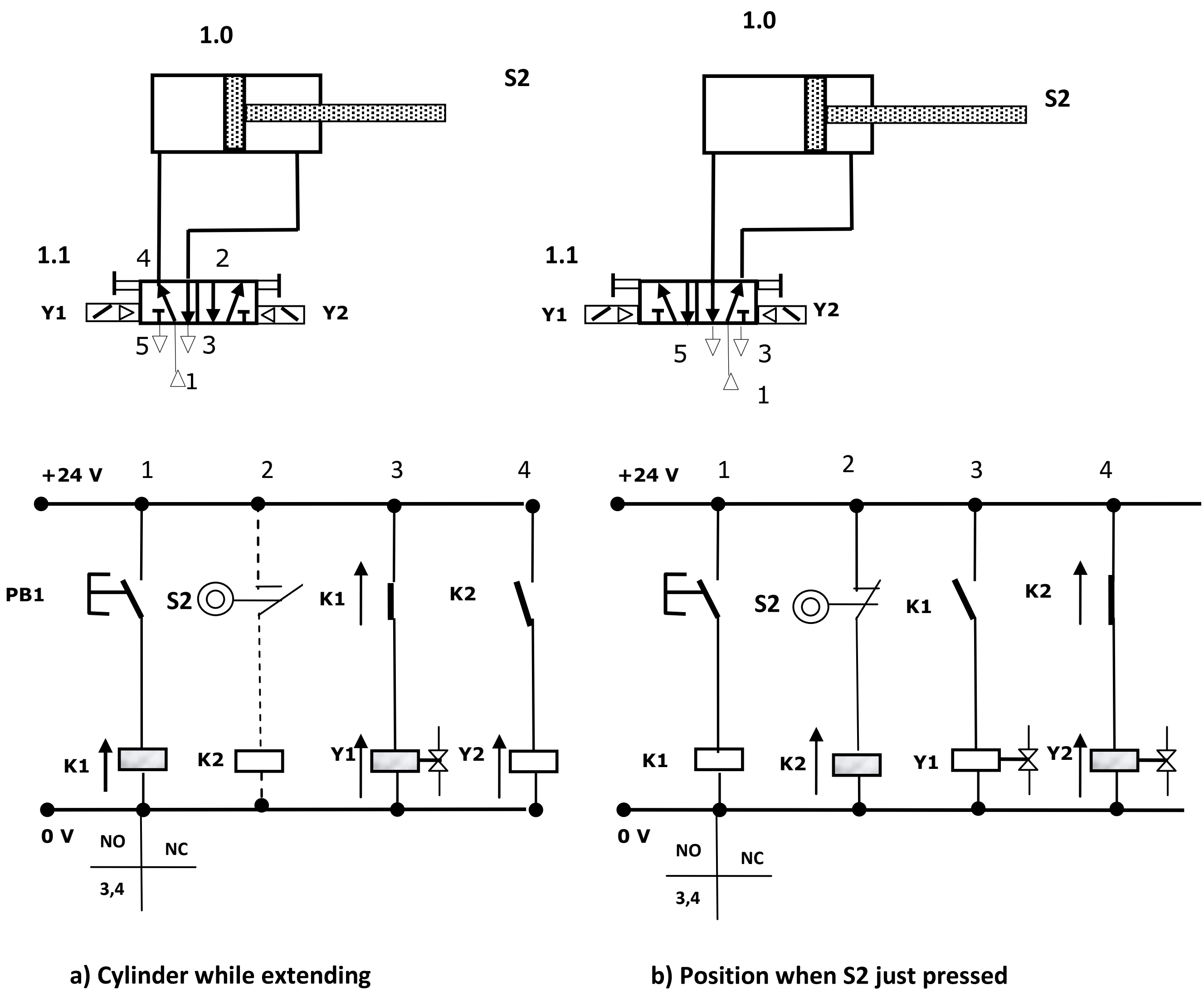


Figure 1.37: Indirect Automatic return of double acting cylinder using double solenoid

1.3.12 Automatic return of a double acting cylinder (using proximity switch) (Figure 1.38)

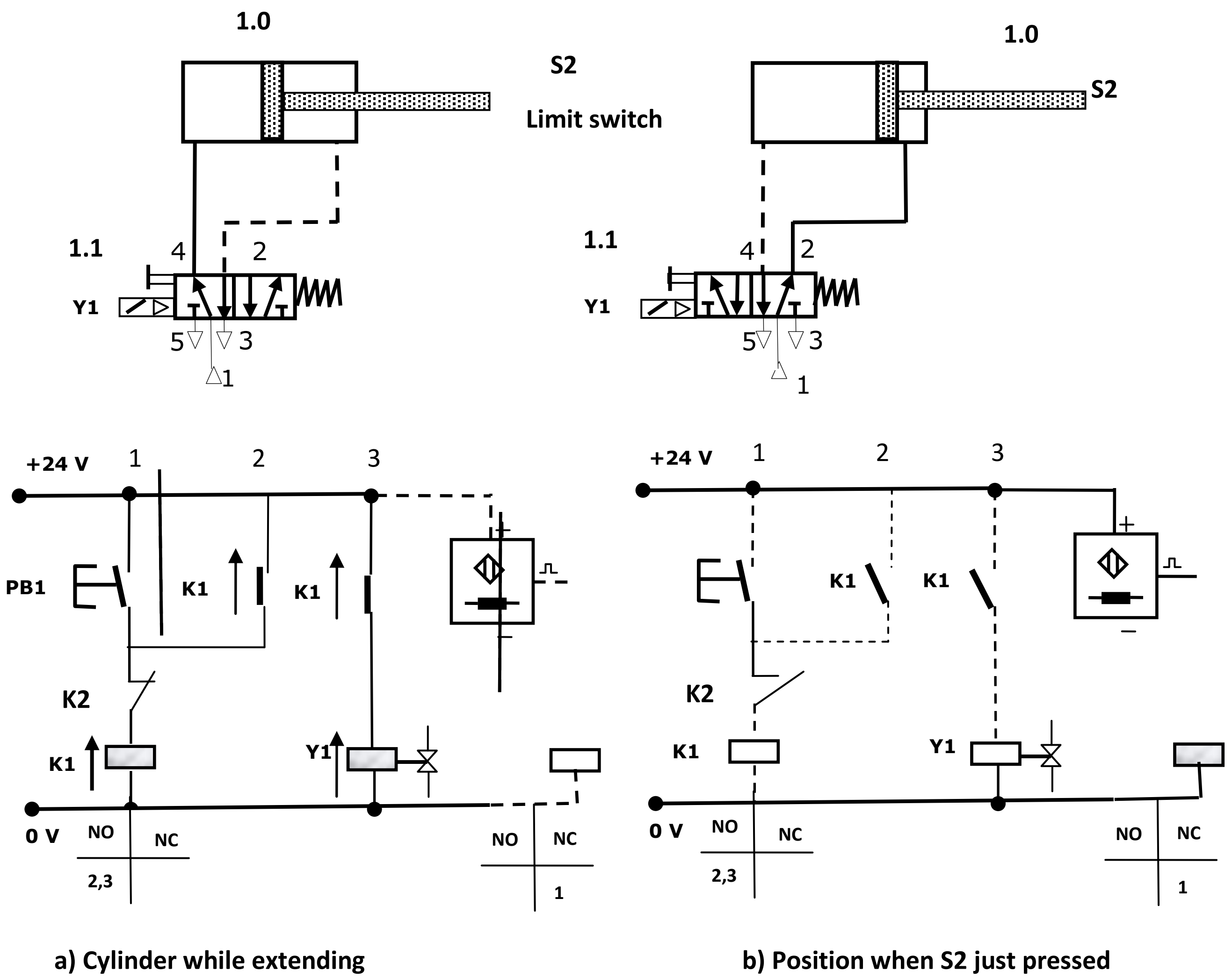
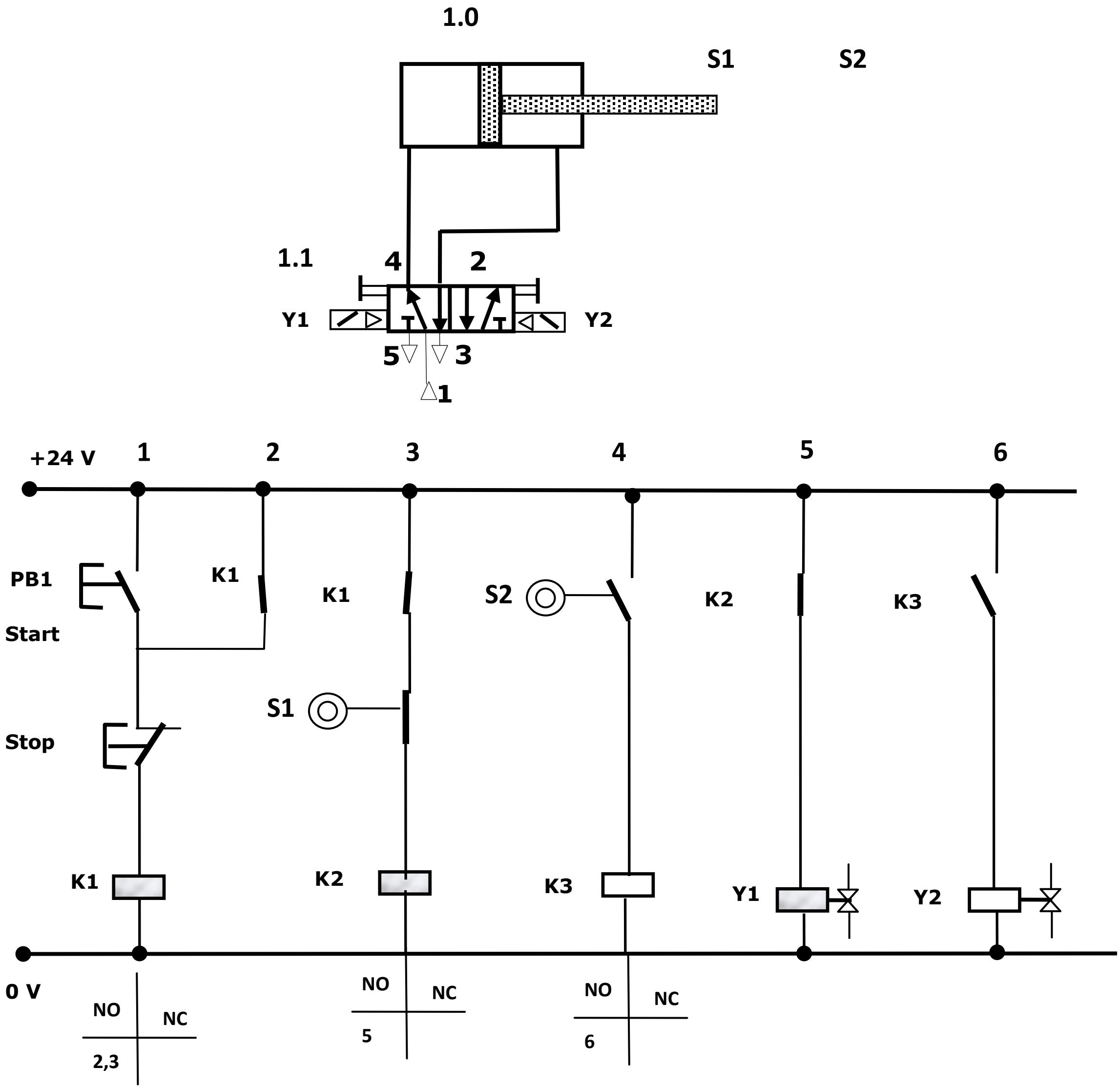


Figure 1.38: Automatic return of double acting cylinder using proximity switch

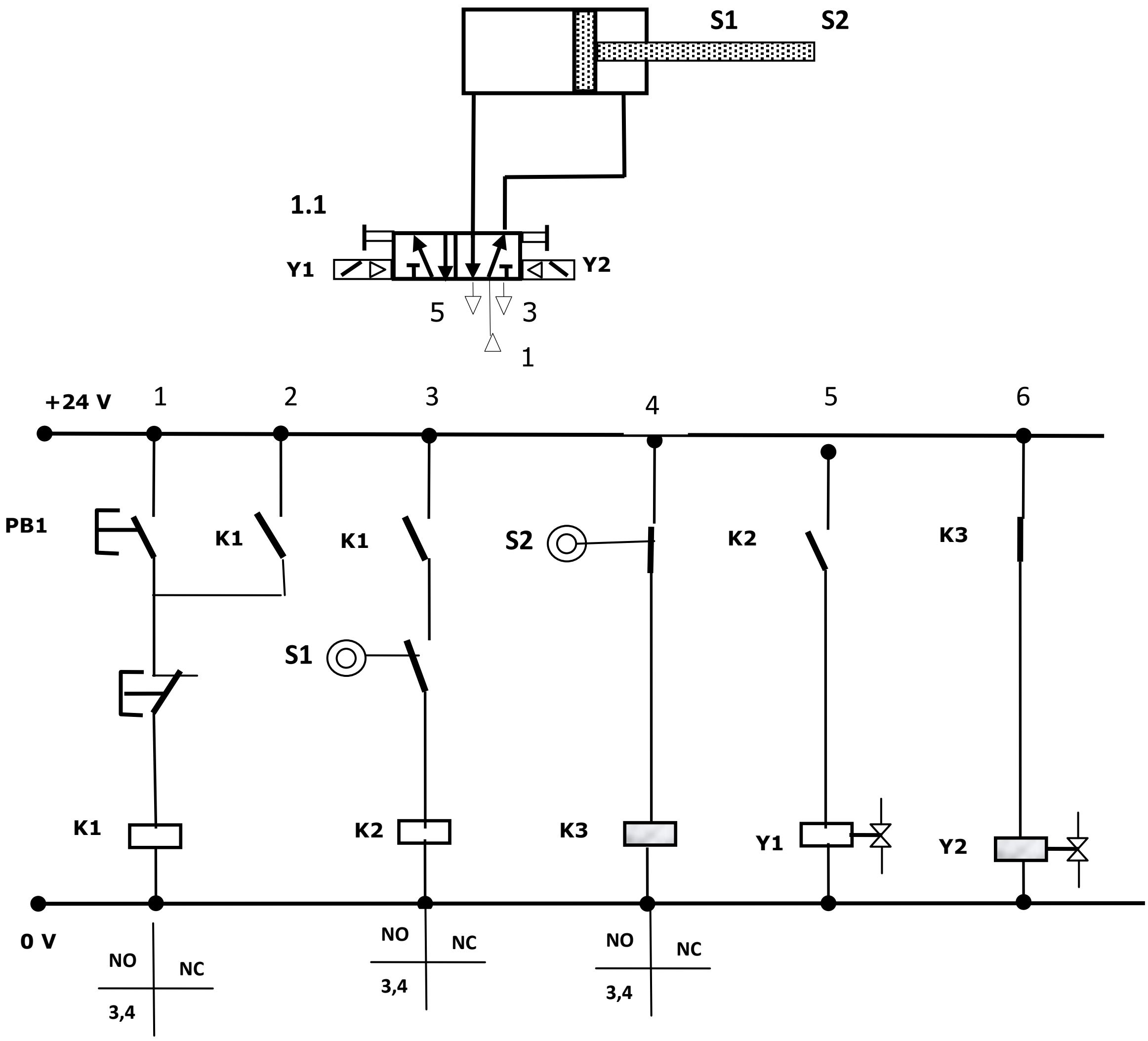
1.3.13 Oscillating motion of a double acting cylinder (Forward) (Figure 1.39)



a) Cylinder while extending

Figure 1.39: Oscillating motion of double acting cylinder (forward motion)

1.3.14 Oscillating motion of a double acting cylinder (Return) (Figure 1.40)



b) Position when S2 just pressed

Figure 1.40: Oscillating motion of double acting cylinder (return motion)

1.3.15 Control of system with timed response

Control systesm which are assinged a particular timing sequence must be equipped with electrical time lag relays. There are control systems which are purely affected by time or combination of path scanning and time.

These time-lag relays , which are usually electronic time lagg relays nowadays, have two basic types of timed response. They are referred to as time-lag relays with energising delay and de-energising dealy. Time lag-relay with energising delay and Time lag-relay with de-energising delays are shown in Figure 1.41 and 1.42.

a) Time lag-relay with energsing delay

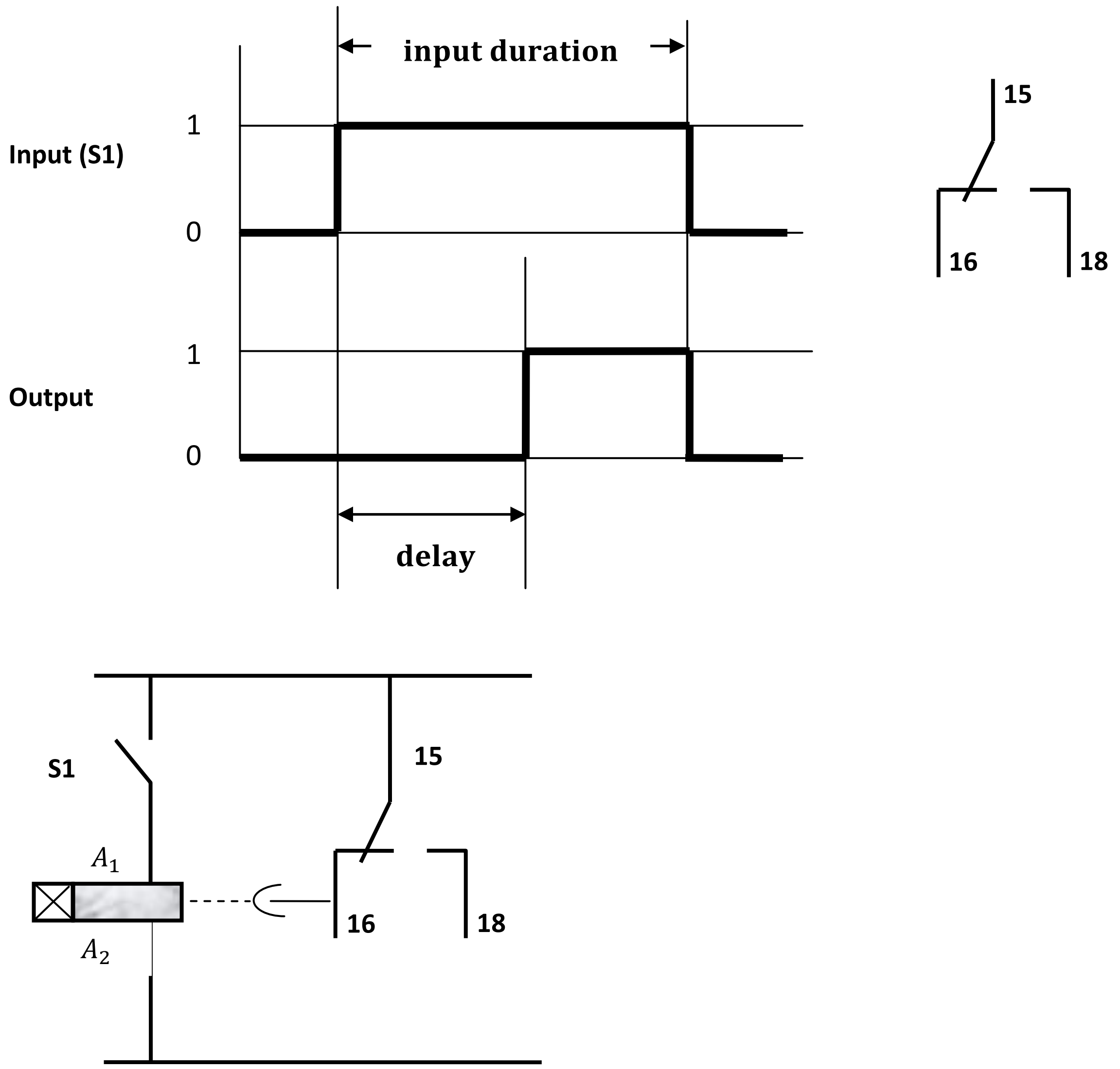
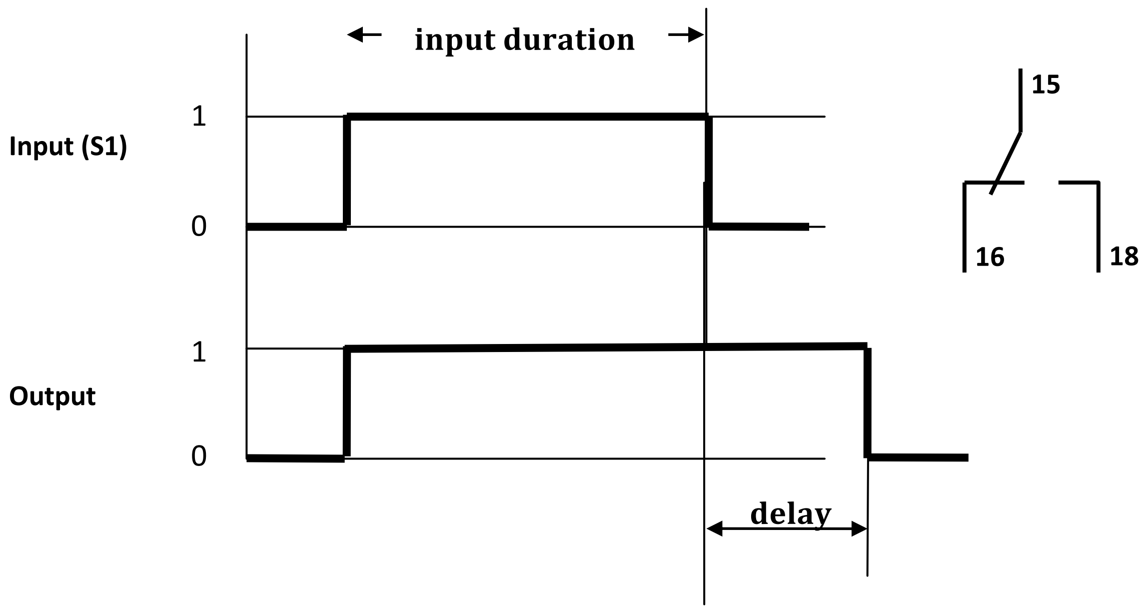


Figure 1.41: **Time lag-relay with energsing delay**

b) **Time lag relay with de-energixing delay**



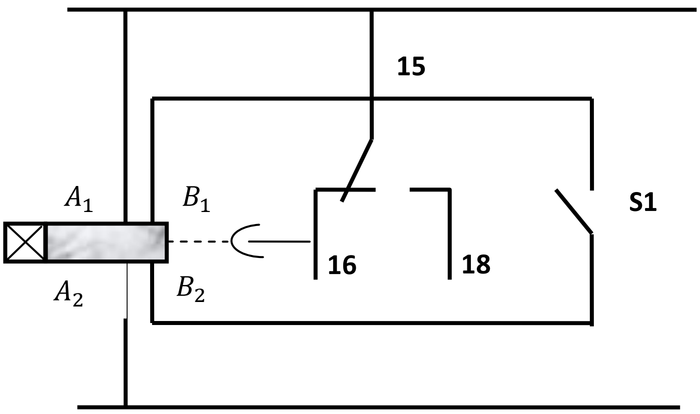


Figure 1.42: **Time lag relay with de-energixing delay**

c) **Control of double acting cylinder with time delay (Double solenoid) (Figure 1.43)**

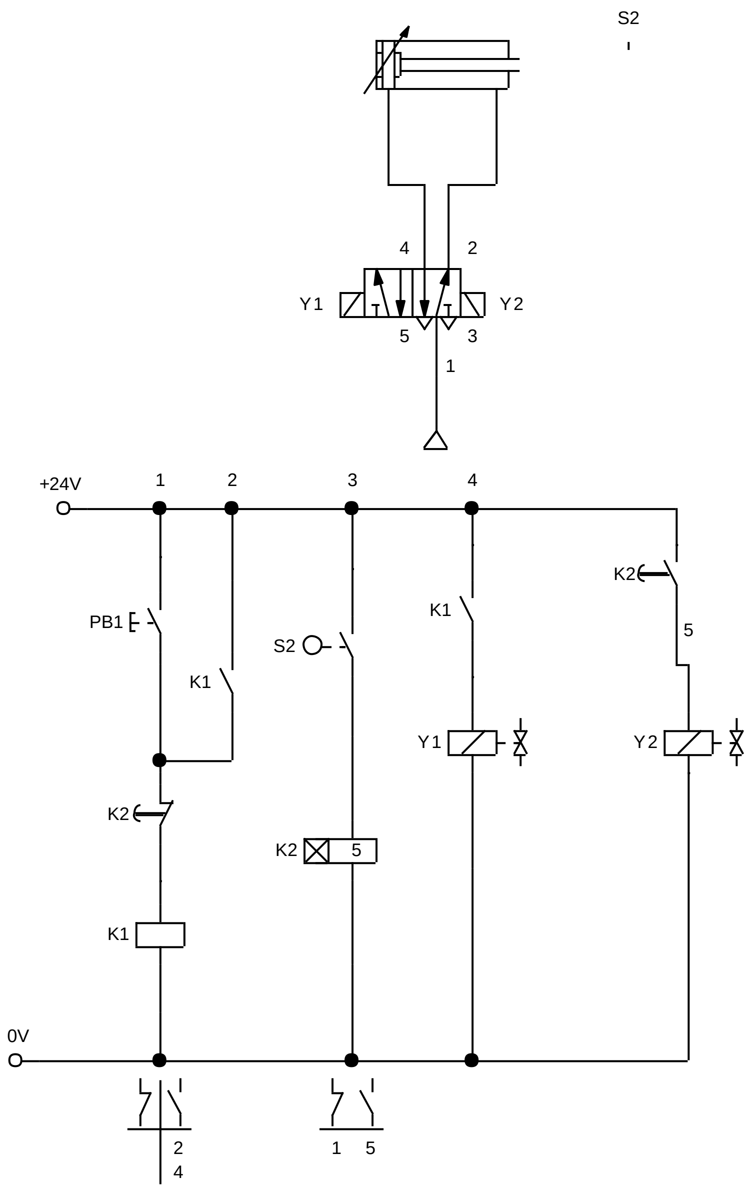


Figure 1.43 Control of double acting cylinder with time delay

When manual pushbutton PB1 is pressed, relay K1 changes state and the normally open contact k1 of relay is connected to solenoid coil Y1. When the normally open contact closes , the solenoid valve changes state,

The cylinder travels to its final forward position where it actuates limit switch S2. This limit switch stats the time lag relay K2 (with energising delay)

After 5 seconds the normally open contact of time lag relay energises the solenoid coil Y2 of the directional control valve. The vavle changes over and causes the piston to travel to its final rear position.

d) Control of double acting cylinder using timer(single solenoid) (Figure 1.44)

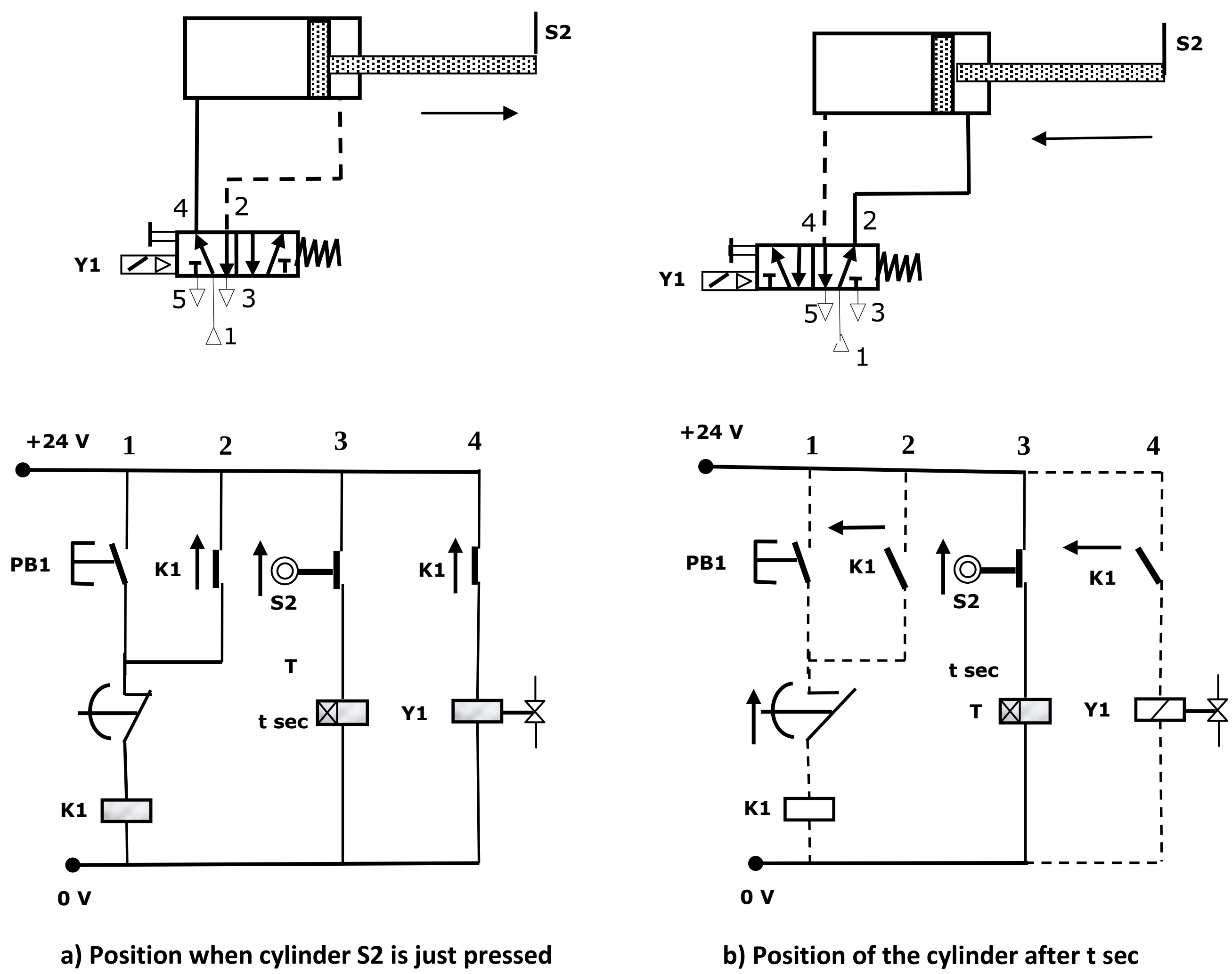


Figure 1.44 Control of double acting cylinder with time delay

A latching circuit is used to obtain the necessary memory function. The position of the circuit when push button PB1 is pressed and then released is given in Figure xx. The cylinder extends to its forward-end position and actuates limit switch S2 automatically. As the return motion is to be delayed, on –delay timer is used to obtain the necessary time delay. The required time delay should be set on the timer. Limit switch S2 controls the timer coil T. After the set delay, the timer contact interrupts the latching circuit, thus causing the return motion of the cylinder as shown in Figure…

1.3.17 Control of double acting cylinder using electric counter with two end sensors (Figure 1.45)

Design a electro pneumatic circuit for a double acting cylinder to perform a continuous to and fro motion. The cylinder has to stop automatically after performing 50 cycles operations

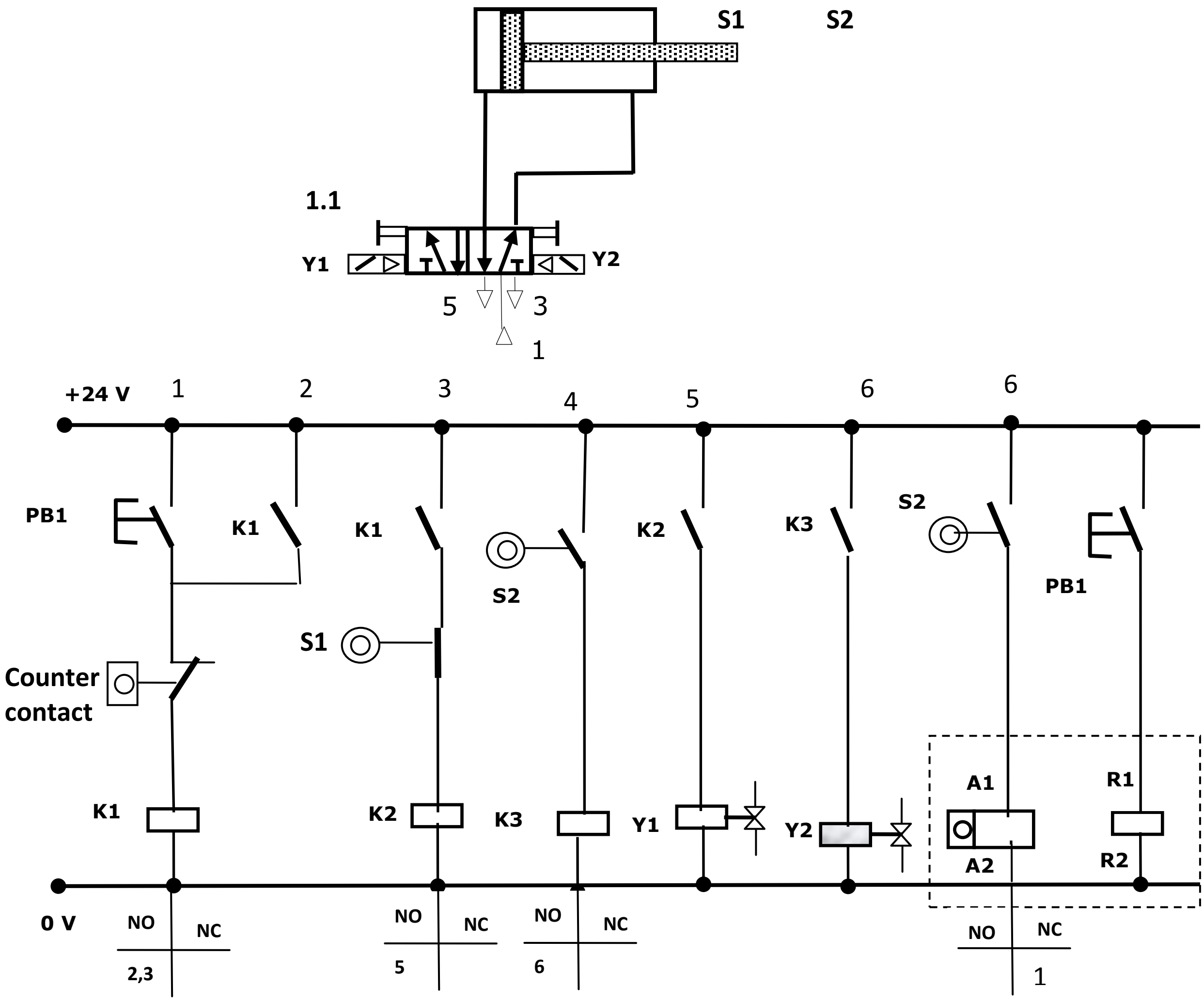


Figure 1.45 Control circuit using timer

When push button PB1 is pressed, it energises the coil K1 in branch 1. K1 in branch 2 is latched with PB1. Contact K1 in branch 3 energises the coil K2 in branch 3, which in turn closes contact K2 in branch 5 causing solenoid coil Y1 in branch 5 to energise and move the direction control valve. Cylinder moves forward.

When cylinder touches the limit switch S2 in branch 7, it sends a signal pulse to counter coil (A1 andA2) in branch 7. After a desired number of cycles is reached (50 cycles), then counter contact C in branch 1 opens and de-energises the K1 and cylinder stops.

1.3.18 Oscillation of double acting cylinder using end positioning with proximity switches

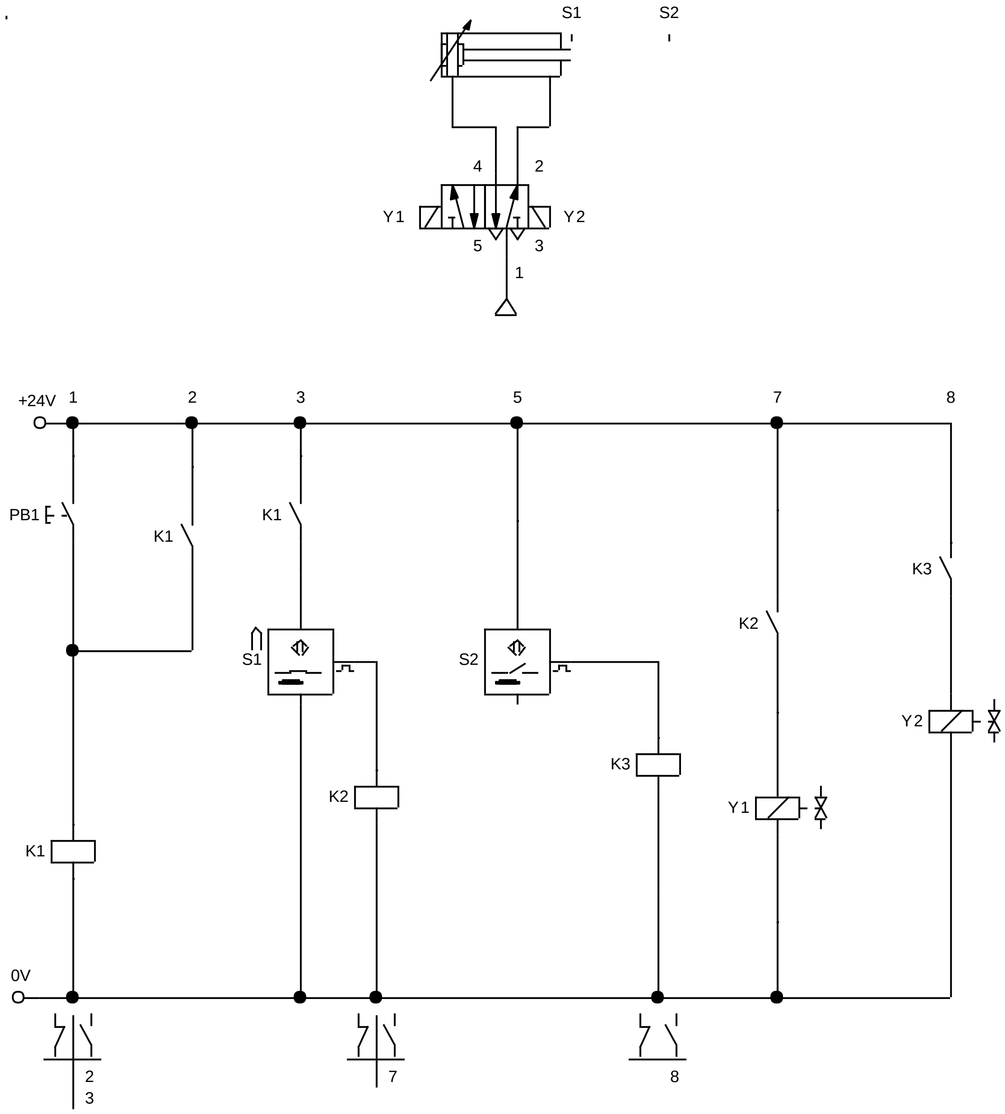


Figure 1.46 Oscillation of double acting using proximity switches

1.3.19 Control of double acting cylinder using pressure switch

Components are to be stamped using stamping machine. A double acting cylinder is used to push the die attached down to a fixture when two push buttons are pressed simultaneously. The die is to return to the initial position upon reaching sufficient stamping pressure as sensed by a pressure switch and one second delay. Develop an electro pneumatic control circuit to implement the control task for the stamping operation.

Solution is shown in the **Figure 1.47**, which is self explanatory

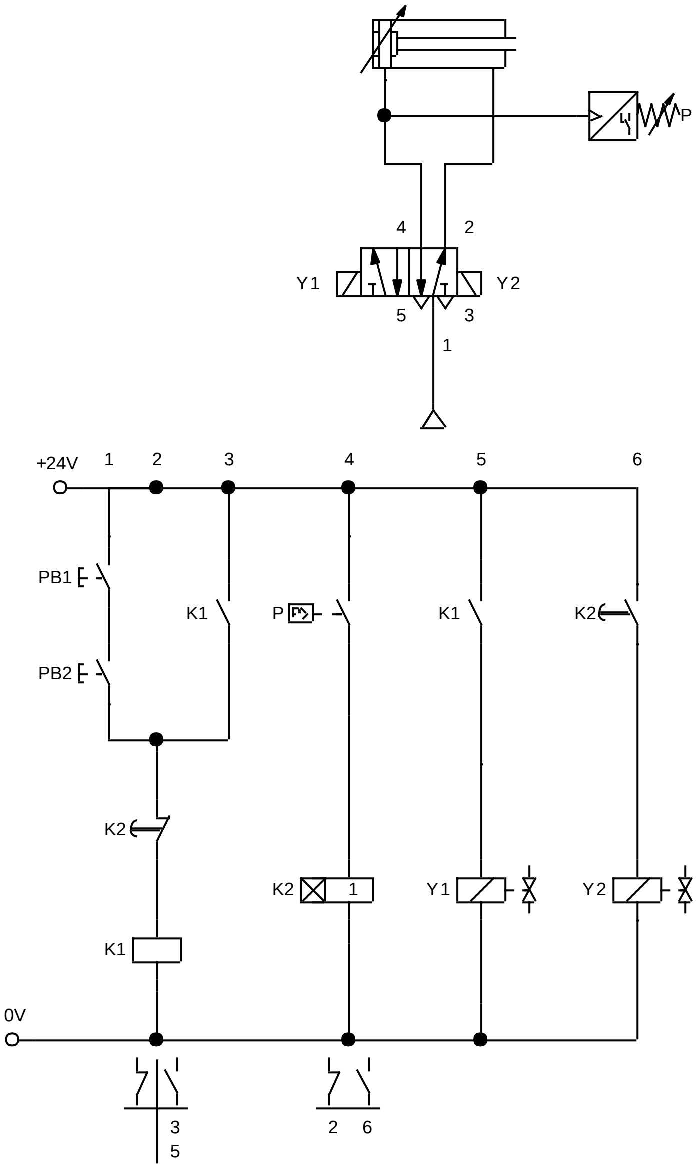


Figure 1.47 Control of double acting cylinder using pressure switch

1.3.20 Control of double acting cylinder using delay on and off timer and counter

Components are to be stamped using stamping machine. A double acting cylinder is used to push the die attached down to a fixture one second after push button is pressed. The die is to return to the initial position upon reaching sufficient stamping time of two seconds is reached. This automatic cycle should stop after 5 cycles. Start button should reset the counter. Develop an electro pneumatic control circuit to implement the control task for the stamping operation.

Solution is shown in the **Figure 1.48**, which is self explanatory

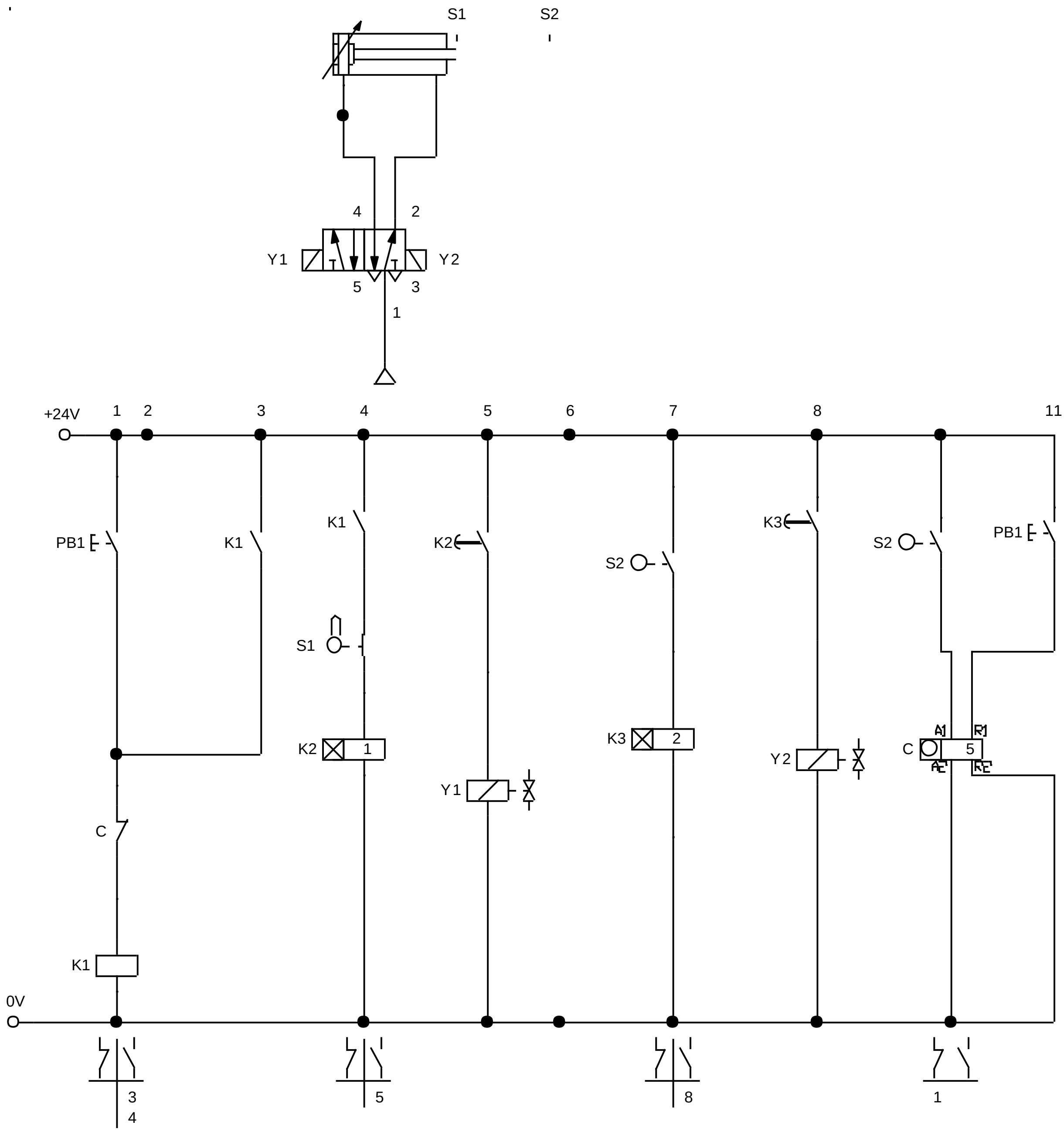


Figure 1.48 **Control of double acting cylinder using counter and delay**

1.3.21 Control of double acting cylinder using delay on and off timer and counter

Components are to be stamped using stamping machine. A double acting cylinder is used to push the die attached down to a fixture one second after push button is pressed. The die is to return to the initial position upon reaching sufficient stamping time of two seconds is reached. This automatic cycle should stop after 5 cycles. Start button should reset the counter. Initial position sensing is through magnetic reed switch. Develop an electro pneumatic control circuit to implement the control task for the stamping operation.

Solution is shown in the **Figure 1.49**, which is self explanatory

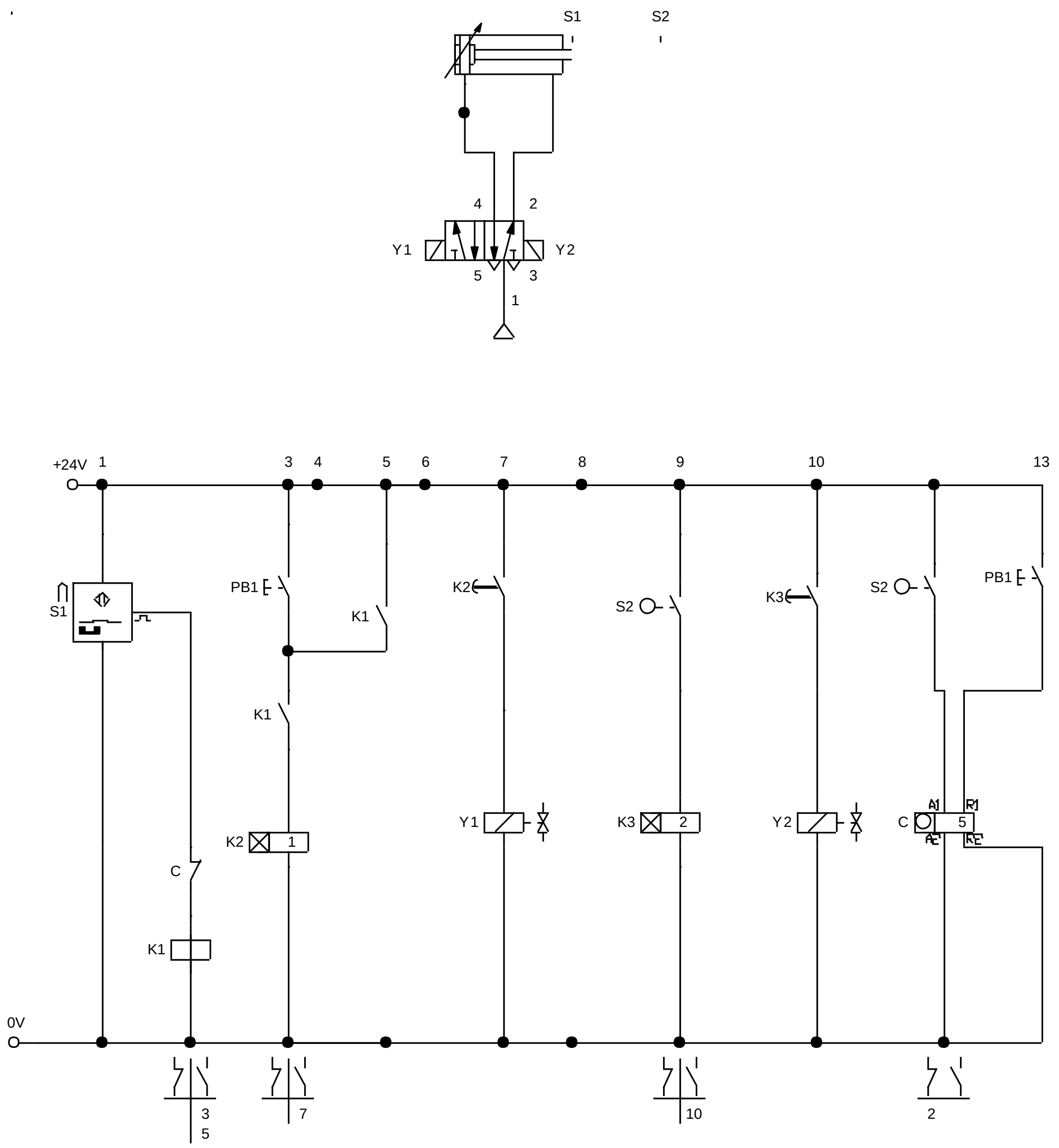


Figure 1.49 **Control of double acting cylinder using reed switch for initial sensing**

Objective Type Questions

1. A relay is considered as an electro -------------- actuated switch

2. When both start and stop buttons are pressed simultaneously, circuits goes to OFF condition in Dominant ------ circuit.

3. Inductive sensors cannot be used to detect ………………….. materials.

4. Push buttons are operated manually, where as limit switches are operated ----------------------

5. --------------- switches are magnetically actuated proximity switch

State True or False

1. Electro pneumatic circuits are less reliable than pure pneumatic circuits.

2. A pressure switch is electric – pneumatic signal convertor

3 A relay is electro pneumatically operated switch that operates under the control of additional electrical circuits.

4. An electrical latching relay circuit is an example of memory function

5. Limit switches are usually operated manually.

Review Questions

1. Explain briefly the working principle of an electro magnetic relay

2. Draw the symbol for an electromagnetic relay with 2 NO and 1 NC

3. What are the different ways to implement memory function in electro pneumatic circuits?

4. Explain the step displacement diagram for A+B+B-A- sequence.

5. List few disadvantages of using electro pneumatic circuits.

6. Explain the principle of cascade method using electro pneumatics with a suitable sequence example

7. What are the functions of sensors

8. Explain the working principle of a limit switch

9. Explain the working principle of a reed switch

10. Draw the symbols for the following

a) limit switch b) Inductive type proximity switch c) capacitive type proximity switch d) optically operated sensor

11. Differentiate between through beam sensors and diffuse sensors

12. Compare inductive , capacitive and optical types of sensors

13. Explain the functions of on delay timers with suitable circuit.

14. Explain the functions of an off delay timers with suitable circuit.

15. Differentiate between the behaviours of on timer delay and off timer delay with the help of a timing diagram.

16. Explain the function of a pressure switch

17. Explain the function of up counter and down counter.

18. Draw a group changing cascade circuit for two groups, three groups, and four groups

19. Explain with the help of neat sketch the construction and working of pressure sequence valve

20. What are the different ways to sense the end position and movement of cylinders

Answers

Fill in the Blanks

1. magnetic
2. off
3. non-metallic
4. automatically
5. reed

State True or False

1. False
2. False
3. False
4. True
5. False

Lecture 42

PNEUMATIC CIRCUIT DESIGN USING PLC

Learning Objectives

Upon completion of this chapter, Student should be able to

- Define PLC and its function
- Explain the difference between hard wired control and PLC control
- Explain the different section of PLC
- List the advantages of PLC over electromagnetic relays.
- Explain the functions of major components of PLC
- Explain various programming approaches used in PLC
- Describe the functions of memory functions, timers and counters
- Convert the logic functions into ladder diagram
- Design PLC circuits for single and multi actuators

1.1 Introduction

A programmable logic controller (PLC) is essentially a user friendly micro-processor based microcomputer, consisting of hardware and software, designed to control the operation of Industrial equipment and processes. An important advantage of the PLC is that it can be easily programmed and reprogrammed. PLC has tremendous impact on Industrial control and instrumentation due to its high reliability and flexibility at the design and implementation stages. The decreasing cost of microprocessor with increasing facilities in them is acting as catalyst in their widening scope of applications. In recent years, PLC are being used in place of electromechanical relays or cam operated logic controllers to control fluid power systems. Modern day PLCs are developed into a sophisticated and highly versatile control system component capable of performing complex mathematics functions and operate at fast microprocessor speeds. Some leading PLC manufacturers are ABB, Allen Bradley, Honeywell, Siemens, GE Fanuc, Mitsubishi, Modicon, Omron etc.

1.2 PLC – Defined

PLC can be defined as digital electronic device that uses a programmable memory to store instructions and to implement functions such as logic, sequencing, counting, timing and arithmetic in order to control machine, processes and instrumentation

PLC is user –friendly digital computer used for making logic decisions and providing output. It consists of solid state digital elements and is a replacement for hard-wired electro-mechanical relays to control pneumatic systems.

The term ‘programmable logic controller’ is defined as follows by IEC 1131 (PLC standard) part 1

“A digitally operating electronic system, designed for use in an industrial environment, which uses a programmable memory for the internal storage of user- oriented instructions for implementing specific functions such as logic , sequencing , timing , counting and arithmetic, to control through digital or analog inputs and outputs, various types of machines or processes. Both the PC and its associated peripherals are designed so that they can easily integrated into an industrial control system and easily used in all their intended function”

PLC is quite similar to digital computers. They also have certain features which are specific to logic controllers. They are

1. PLC are rugged and designed to withstand vibrations, temperature , humidity and noise
2. The interfacing for input and output is part of the controller
3. They are easily programmable and primarily use logic and switching functions

1.3 Hard-wired control systems

In hard wired control systems, relays are used. For example: In Electricalcontrol, the wiring of control elements such as sensors, solenoids, counters etc. are through relays control. Such relay controlled systems are also called as hard-wired control system because any modification in control program involves rewiring of the circuit. Therefore, hardwired controls are cumbersome and difficult to modify when production requirement changes regularly. Hard-wired control systems are difficult to maintain because any small problem in design could be a major problem in terms of tracing and rewiring.

Hard wired control systems consists of three division

1. Input section – Consists of push –buttons, switches and sensors. They transfer signals to the processing section

2. Processing section –Consists of relay coils and contacts. They determined the relationship between the inputs received and outputs required

3. Output section –Consists of solenoids, lamps, and contactor coils etc. The processed signals are transferred to this section.

1.4 PLC Systems.

PLC systems offer number of advantages over hard wired electromechanical relay control systems. Unlike the electromechanical relays, PLCs are not hard-wired to perform specific functions. Thus, when system operation requirement change, a software program is readily changed instead of having to physically rewire relays. In addition, PLCs are more reliable, faster in operation, smaller in size, and can be readily expanded.

PLC systems consists of three division

1. Input section – Consists of push –buttons, switches and sensors which are connected to specific input addresses in the program. They transfer address information to the processing **section**

2. Processing section –The microprocessor receives the input signals from input sections and executes the information (called instructions) in the software program and sends the processed signals to output section

3. Output section –Takes the signal from processing section and modify the signal from the processor to operate output devices connected to specific output addresses.

Advantages of PLCs over Electromechanical relays

The PLC replaces electromechanical relays due to their following advantages

a. PLCs are more reliable and faster in operation
b. They are compact and can be expanded easily
c. They require less electrical power
d. They are less expensive when compared to Hardwired systems of same number of control functions
e. Hard-wired electromechanical relays lack flexibility. For example, when system operation requirement change, then the relays have to be rewired.
f. PLCs have very few hardware failures when compared to electro-mechanical relays
g. Special functions such as time-delay actions and counters , can be easily performed using PLCs

Comparison between Relay and PLC is presented in **Table 1.1**

Table 1.1 Comparison between Relay and PLC

Features	**Electromechanical relay**	**PLC**
Size of the controller	Bulky	Compact
Programming	Time consuming	Easy
Flexibility	Rewiring required	Reprogramming required
Cost	Expensive	Less expensive
Maintenance	Poor	Minimum
Fault finding and troubleshooting	Difficult	Easy to identify fault and repair

1.5 MAJOR COMPONENTS OF PLC

As discussed earlier, A PLC is essentially a microcomputer consisting of hardware and software. The major components are

1. Power Supply module
2. Input module
3. Central processing unit
4. Output modules
5. Software

a)Power supply module:

Usually input output modules require 24V DC and processor require 5V DC. Usually power supply is integral part of PLC. Power supply units convert 120/230 V AC line voltage to standard supply of 24 VDC or 5V DC using standard rectifier circuits

b)Input module

Input devices include push buttons, sensors, potentiometers, pressure switches. The function of the input module is to covert high voltages from input devices to low level logic voltages that the CPU uses internally for processing.

Input module can process both analog input and digital input. Digital inputs are more preferred in Industry.

c)Analog input module is used to convert analog signal form analog devices like temperature sensors, pressure sensors etc. to digital signals using ADC (Analog –to digital convertor). Analog signal is varying voltage in the range of 0-12 V or current in the range of 5-20 mA. These values of current or voltage is converted into integer value (say16 bit word)

Digital `is used to convert signal digital input to 5V digital signals that CPU uses internally to execute a user program.

d)Central processing unit

The central processing unit controls and processes all operations within the PLC and hence called brain of the PLC. The CPU can perform various arithmetic and data manipulation function with the local and remotely located Input/output sections. Further, the processor can perform many communication functions it needs to interface with a personal computer, remote Input/Output, other PLCs and peripheral devices

Functions of CPU are :

1. It receives input from various sensing devices and switches
2. It executes the user program
3. It makes various decisions to control the operation of the equipment or process
4. It can perform various arithmetic and data manipulation functions
5. It delivers corresponding output signals to various load control devices such as relay coils and solenoids

e)Output module

Output devices include contactor coils, solenoid coils, lamps, etc. Output module amplifies the low–level logic signals generated by the CPU and pass these modified signals to the final control elements to operate the output devices.

f) Software

PLC consists of two parts: Operating systems and user program. The PLC operating system provides effective support ranging from the creation of project structure to the creation of user programs. The OS system is accessed through a graphical user interface window (also known as Main window). The main window contains all the functions needed to set up a project , configure the hardware , write and test programs. User program can be written in any standard PLC programming language like ladder diagram or statement list.

While processing a PLC program, the CPU scans and executes the main program cyclically;A program scan cycle consists of sequential operations that include input scan, program scan, and output scan. In the input scan, the CPU updates the process image input table, in the output scan;

the CPU updates the process image output table. After the completion of each scan cycle, the CPU returns to the beginning of the next cycle and again repeats the cycle. The time taken to scan one program is called scan –cycle time.

1.6 PROGRAMMING OF PLC

There are various approaches for entering the program into PLC they are

1. Ladder diagram based
2. Low level based on Boolean expressions
3. Functional blocks
4. High level language

Most of the programming methods used today for PLC are based on the ladder logic diagram. Therefore the concept of ladder diagram is explained in the following sections

The PLC programming based on the use of ladder diagram involves writing a program in a similar manner to drawing a switching circuit. The ladder logic diagram is converted into PLC ladder diagram by using the conventions of PLC ladder diagram constructions. This method requires the use of simple keyboard and CRT with minimum graphic capability to display the symbols, representing components and their inter relationship in the ladder logic diagram. The components are of two types, contact and coils. Contacts are used to represent input switches, relay contacts and similar elements. Coils are used to represent load such as solenoids, relays, timers, counters etc. The programmer inputs the ladder diagram rung by rung into the PLC memory with the CRT displaying the results for verification.

The ladder diagram has two vertical sides (also called rungs) (**Figure 1.1**). The left side line represent line with a positive voltage and right side represent a line with zero voltage. Between these two sides are the horizontal rungs for the assumed power flow. The symbols representing the various program elements are placed on the rungs in order to realize the required control task.

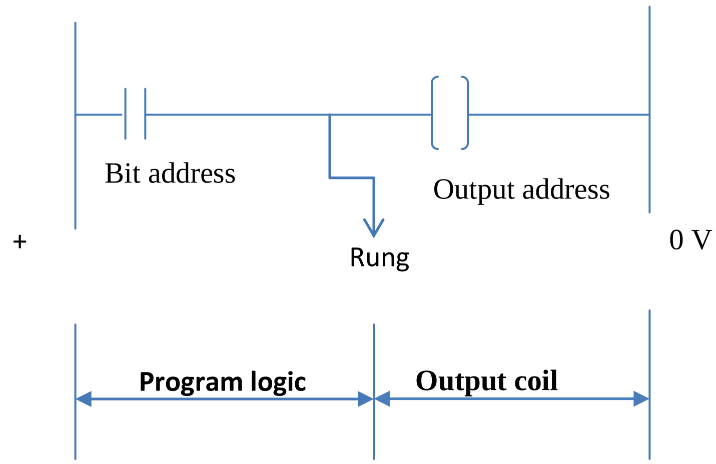


Figure 1.1 Ladder diagram

There are five program elements/operations commonly used in PLC ladder diagram they are

1. PLC Bit logic operations
2. Timer Operations
3. Counter operations
4. Comparison operations
5. Arithmetic operations.

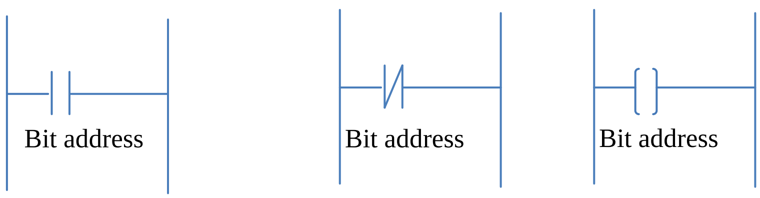
1 PLC bit logic operations : Some important programming elements for bit logic operations are

a) NO contact
b) NC contact
c) Coil

Each of these elements can be selected from the program window. NO and NC elements should not be confused with the hardware NO and NC contacts of switching devices.

NO Contact of PLC:

The PLC representation of NO contact is given in Figure 1.2. This contact scans for the signal state ON (1) at the specified bit address. Power flows through NO contact if the scanned bit address has a signal state ON (1). This contact is used for scanning the signal state of input devices or output devices or other internal program elements.



NO program contact **NC program contact** **PLC coil**

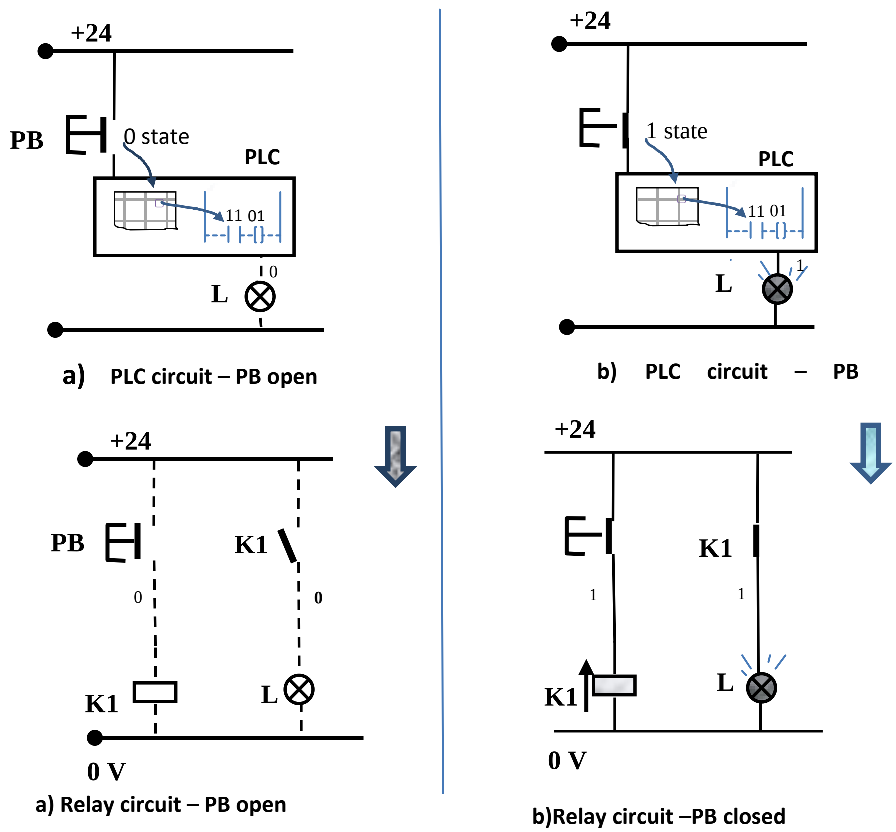


Figure 1.2 **PLC circuit with NO contact position using NO push button**

NC Contact of PLC:

The PLC representation of NC contact is given in **Figure 1.3** This contact scans for the signal state OFF (0) at the specified bit address. Power flows through NC contact if the scanned bit address has a signal state OFF (0). This contact is used for scanning the signal state of input devices or output devices or other internal program elements. **Figure 1.4** shows PLC circuit with NC contact position using NO push button. **Figure 1.5** shows PLC circuit with NC contact position using NC push button.

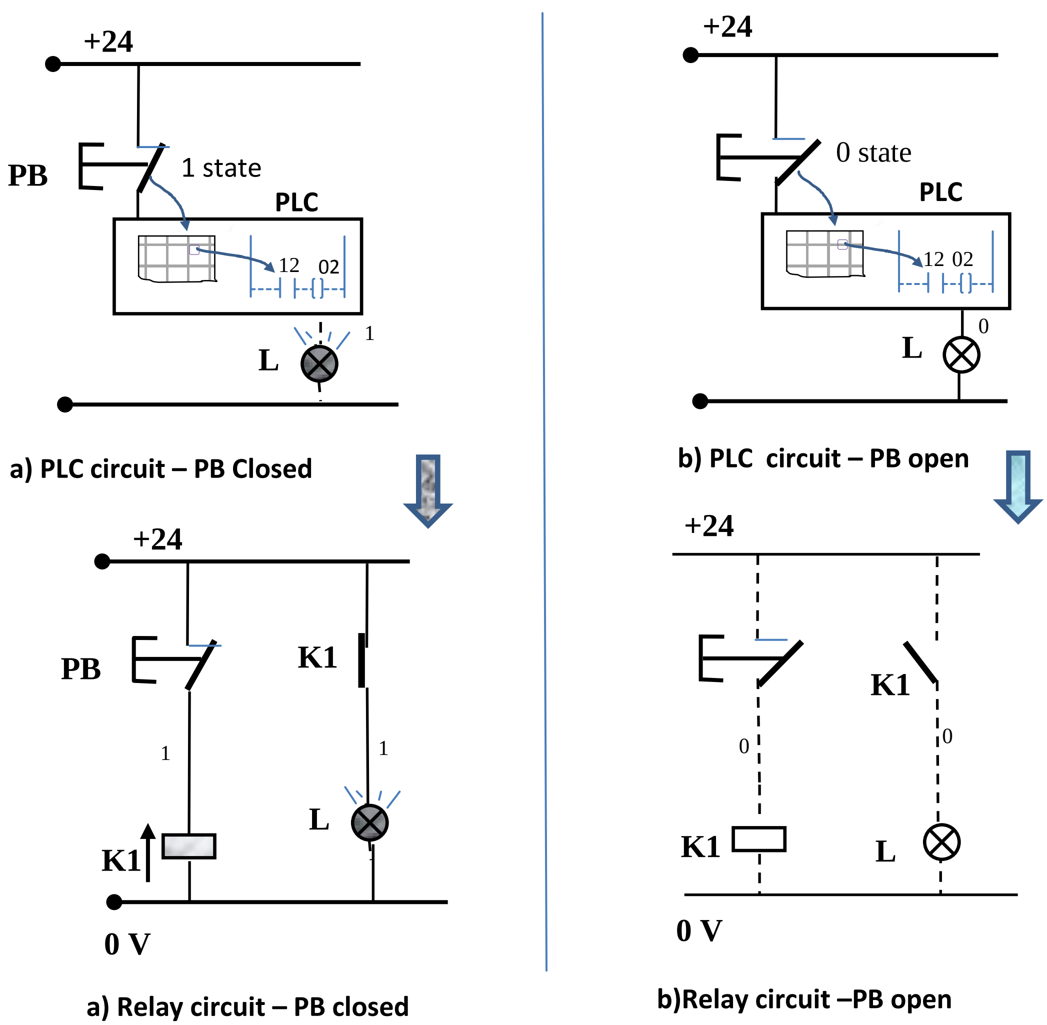
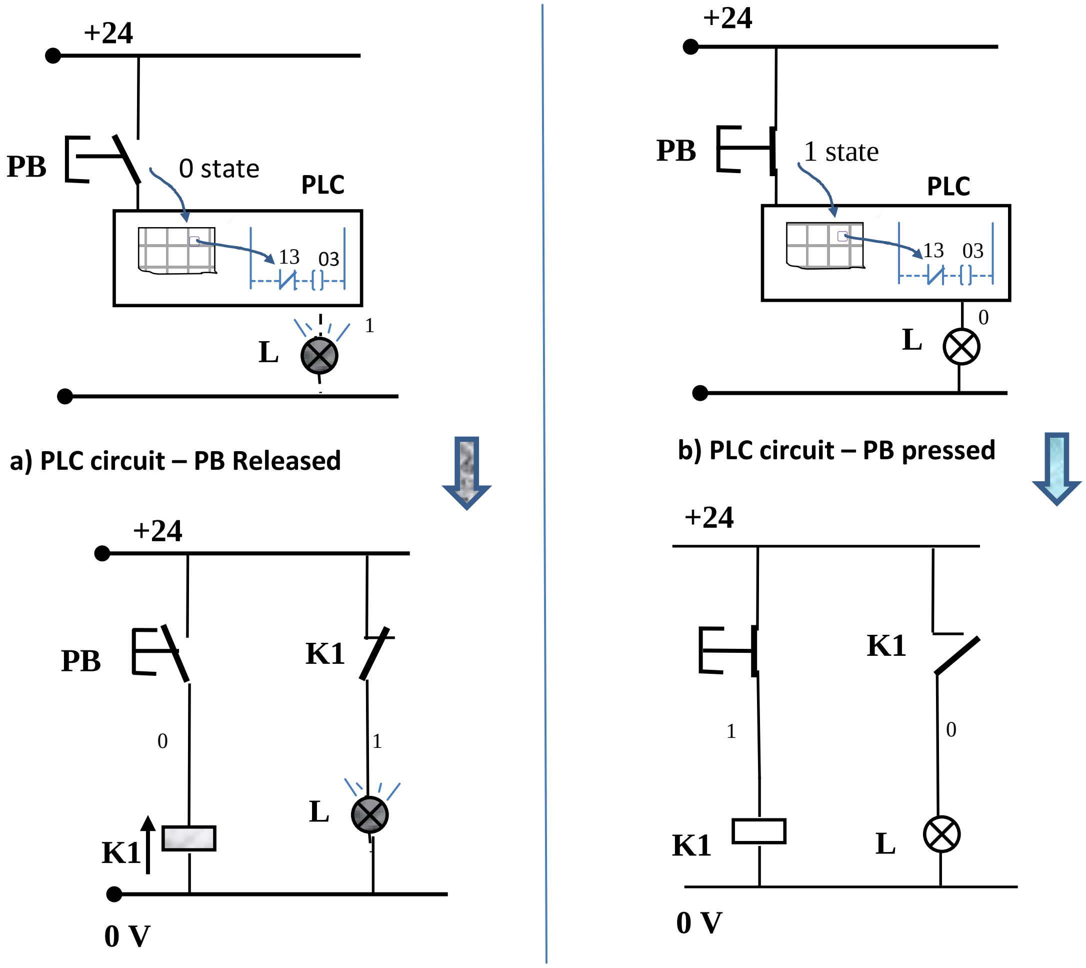


Figure 1.3 **PLC circuit with NO contact position using NC push button**



a) PLC circuit – PB Released

b) PLC circuit – PB pressed

a) Relay circuit – PB released

b) Relay circuit –PB pressed

Figure 1.4 **PLC circuit with NC contact position using NO push button**

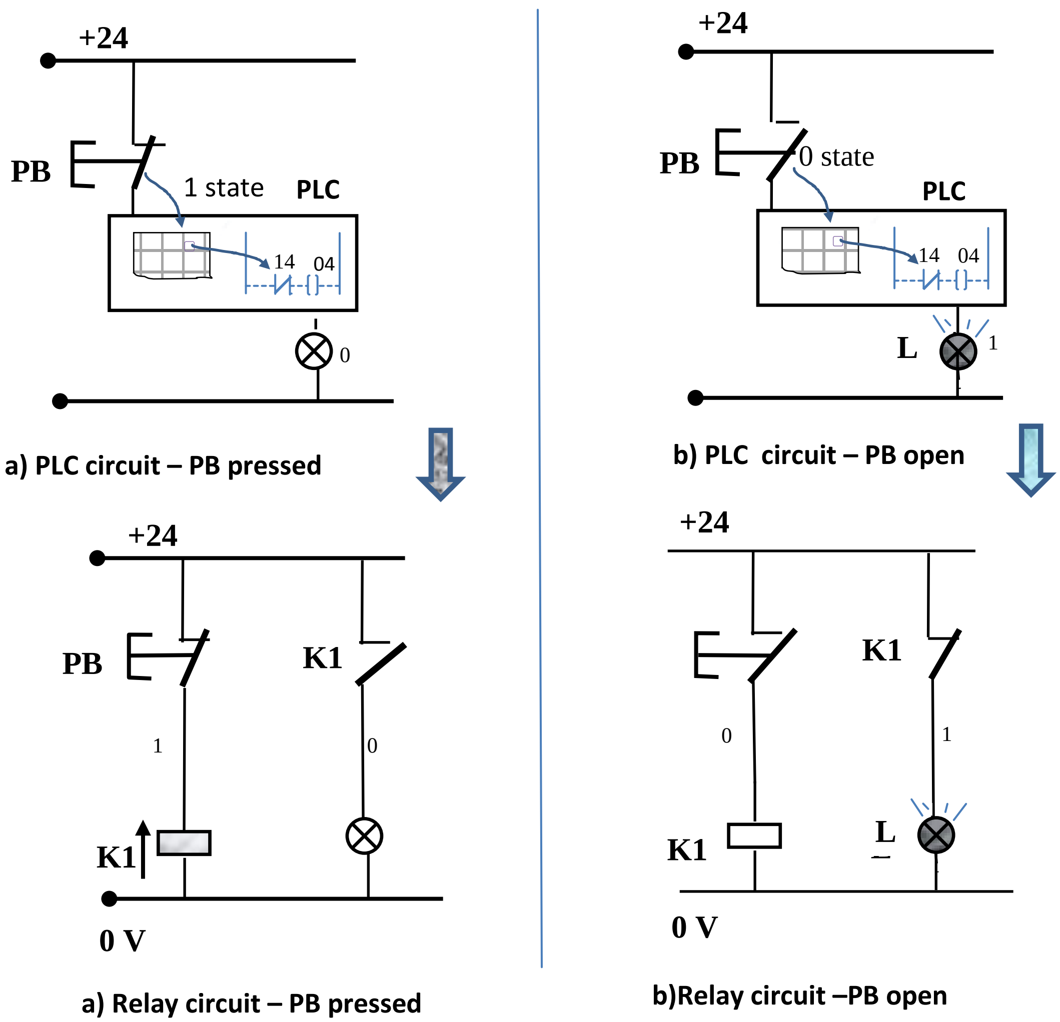


Figure 1.5 PLC circuit with NC contact position using NC push button

1.7 PLC TIMERS

Many control tasks require the programming of time. For example, cylinder 2 is to extend, if the cylinder 1 is retracted- but only after a delay of few seconds. The timers of a PLC are realised in the form of software modules and are based on the generation of digital timing. Memory space is allocated in system memory to store the values of the delay time. The represnetion of the timer address varies from manufacturer to manufacturer. For sake of understanding we shall as T1, T2 for timer addresses. The typical number of timers available in commerical PLC are 64, 128, 256 , 512 or even more. To explictiy reset timer, an RLO of 1 has to be appled at the reset port.

There are two types of PLC timer

PLC on delay timer : The timer will be ON state when it recieves a start input siganl and The signal state of output changes to 0 from 1 , when preset timing is reached. The signal state of the output changes from 0 to 1 when preset time has been reached with reference to change of RLO (Result of logic operation) from 0 to 1(ON) at the start input .Functional diagram is shown in **Figure 1.6**

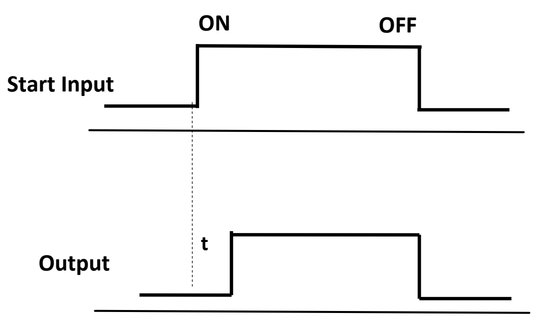
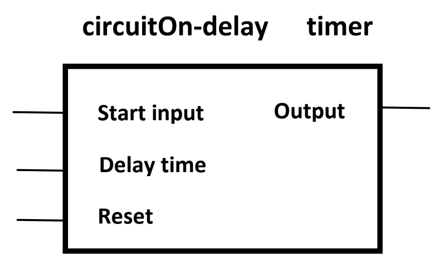
Timing diagram	**Symbol**
	

Figure 1.6 : PLC On delay timer

PLC off delay timer : The timer will be ON state when it recieves a start input siganl and The signal state of output changes to 1from 0 , when preset timing is reached. The signal state of the output changes from 1 to 0 when preset time has been reached with reference to change of RLO from 1 to 0(OFF) at the start input. Functional diagram is shown in **Figure 1.7**

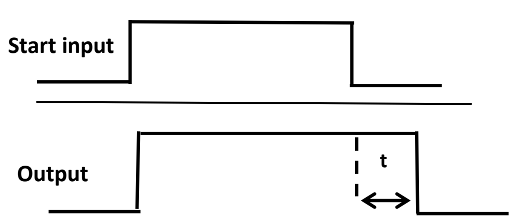
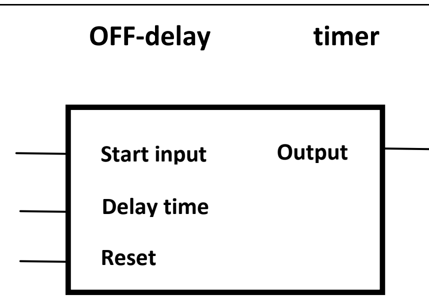
Timing diagram	**Symbol**
	

Figure 1.7 : PLC OFF delay timer

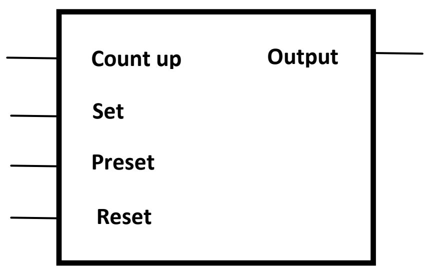
1.8 PLC COUNTERS

Counters are used to detect pieces numbers and events. Controllers frequently need to operate with counters in practice. For example: a counter in circuit is required if exactly 20 identical components are to be conveyed to a converyor belt via a sorting device.

There are two basic counter types a) Count Up b) Count down

When the input to count up counter goes true the accumulator value will increase by 1 (no matter how long the input is true). If the accumulator value reaches the preset value the counter bit will be set. A count down counter will decrease the accumulator value until the preset vaule is reached. Symbols are shown in **Figure 1.8**

Up counter



Down counter

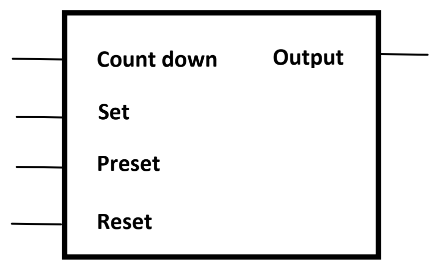


Figure 1.8 : Up and down counter

1.9 PLC Memory elements

Memory elements are used to store intermediate values. Memory function are achieved using flags (bit memory locations) and system memory. Specified bit memory can be set or reset using a set coil. A latch in ladder logic uses one instruction to latch and a second instruction to unlatch, as shown in **Figure 1.9**. The output with an S inside will turn the output D on when the input A becomes true. D will stay on even if A turns off. Output D will turn off if input B becomes true and output with a R inside becomes True.

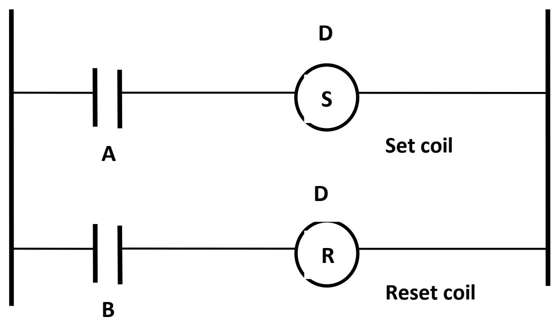


Figure 1.9 a ladder logic latch

Both set coil and reset coil can be combined in one box as shown in **Figure 1.10**. Following instructions are to be followed while writing a program for memory function

- Setting up a memory location
- Resetting up a memory location

The memory address locations vary from manufacturer to manufacturer. For sake of simplicity we shall use M1, M2 … to represent memory. Number of bit memories available in PLC for memory are 1024, 2048 8192 and more.

If the power flows either momentarily or continuously to the set coil, the specified memory address is set to signal state 1. If power flows momentarily or continuously to the reset coil , the corresponding memory address is reset to signal state 0. If there is no power in the set input or reset input, the memory address remains unaffected. The output of the memory function can be accessed through either NO or NC program element

Set reset memory	**Reset set memory bloc**

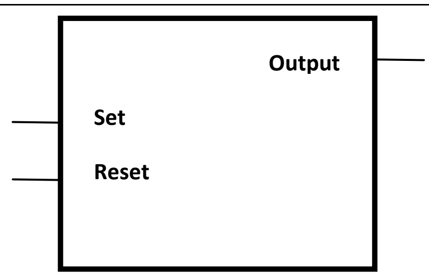
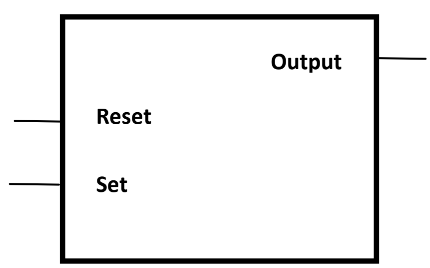

	

Figure 1.10 : Memort box representation	

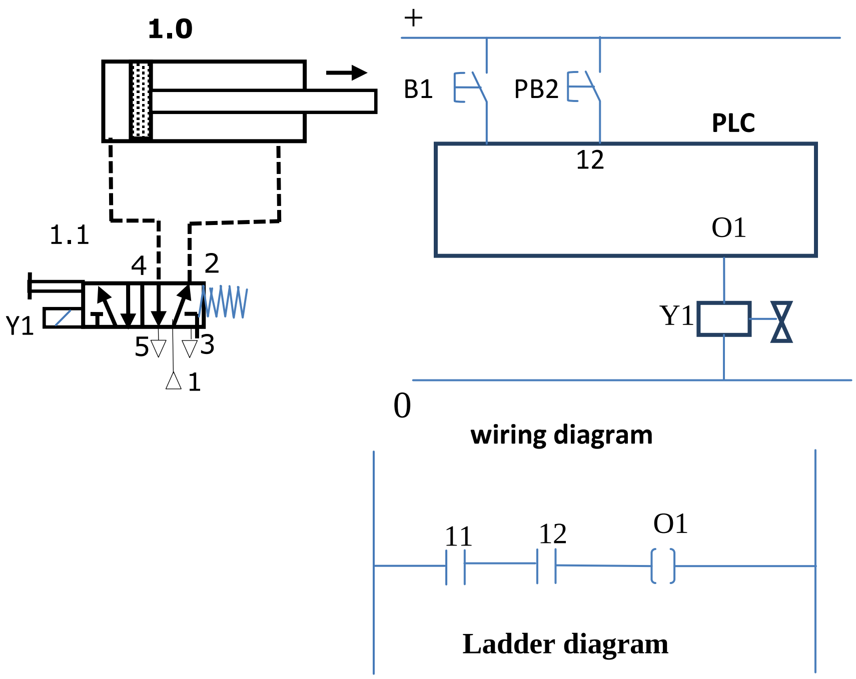
Set and reset functions are combined in one memory box as shown in Figure 1.10. They can be further classified into two categories

a) Memory box with set priority
b) Memory box with reset priority

The functions of a memory box are similar to the memory coils. In the memory box with set priority, the associated memory address is set when signal state 1 appears simultaneously at both the set and reset inputs. In the memory box with reset priority, the associated memory address is reset when signal state 1 appears simultaneously at both reset and set inputs. This concept is similar to Dominant ON and Dominant OFF functions of electrical latching circuits discussed in chapter lecture 41

Example 1 : Double acting cylinder is used to perform machinng operation. Pneumatic cylinder is advanced by pressing two push buttons simultaneously. If any one of the push button is released, cylinder comes back to start position. Draw the pneumatic circuit, PLC wiring diagram and ladder diagram to implement this task.

Solution is shown in Figure 1.11



Assignment Table	
PB1	11
PB2	12
Y1	01

Figure 1.11 a) Pneumatic diagram b) wiring diagram c) ladder diagram

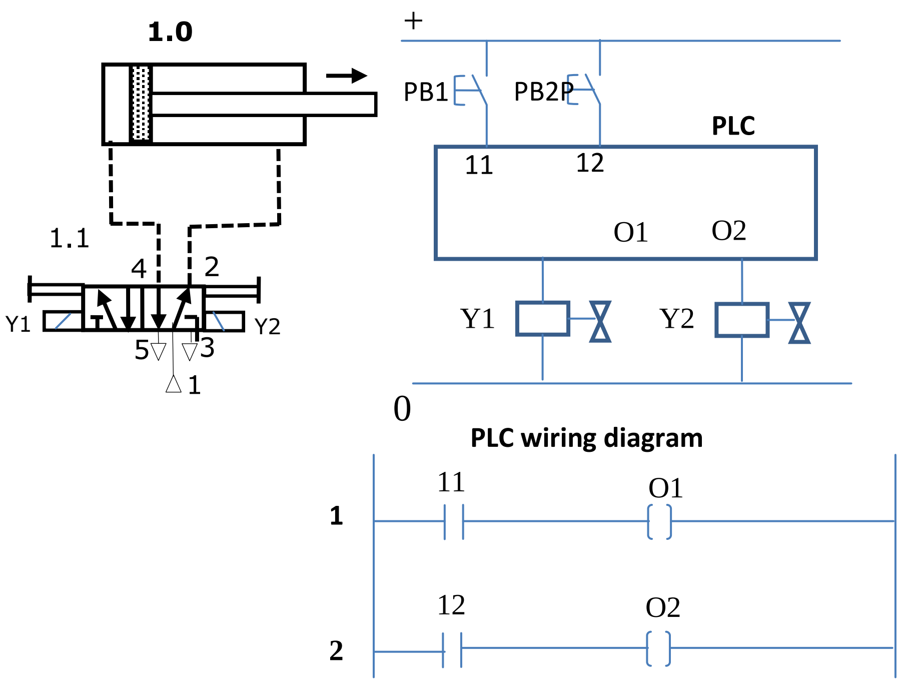
As shown in the PLC wiring diaram , The pushbuttons PB1 and PB2 are connected at memory address I1 and I2. I1 and I2 are connected in series in ladder diagram to relase this AND logic funtion.

When the push buttons PB1 and PB2 are pressed simultaneously, the addresses I1 and I 2 turn to state 1 from state 0 , as a result power flows thorugh the coil and there will be output at coil 01. Output at the coil 01 operated the solenoid coil and cylinder moves foraward to do the required operation.

If any one of PB1 and PB2 is pressed, then corresponding bit addresses turns to 0, since I1 and I2 are in series , if any of them turns to 0 state , there will not be any outout at 01 and thus solenoid is deenergised and returns back.

Example 2 : Double acting cylinder is used to perform forward and return motion. Pneumatic cylinder is advanced by pressing push buttons PB1. Cylinder is returned by pressing push button PB2. Draw the pneumatic circuit, PLC wiring diagram and ladder diagram to implement this task.

Solution is shown in Figure 1.12



Assignment Table	
PB1	11
PB2	12
Y1	01
Y2	02

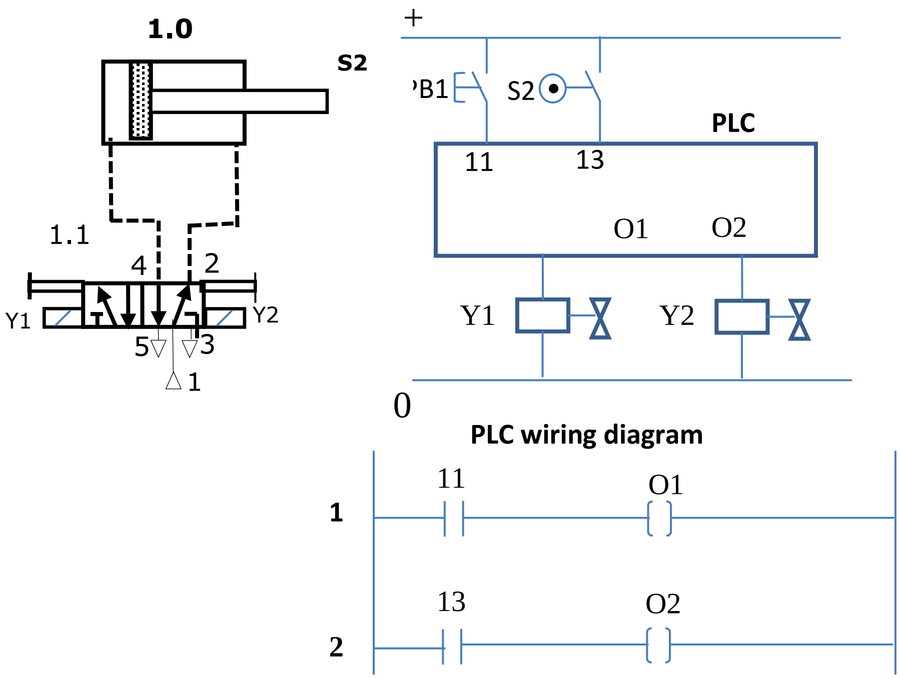
Figure 1.12 **a) Pneumatic diagram b) wiring diagram c) ladder diagram PLC**

PLC Wiring diagram and Ladder diagrams are shown in Figure 1.7. when the pushbutton PB1 is pressed state of the address I1 turns to 1 and thus there will be output 01. The output of 01 operates the solenoid Y1 and cylinder moves forward,

When the cylinder reaches the extreme forward position, and Push button PB2 is operated , the state of address I2 turns to 1 and thus there will be output 02. The output of 02 operates the solenoid Y2 and cylinder return back to initial position.

Example 3 : Double acting cylinder is used to perform forward and return automatically after reaching the extreme forward position. Pneumatic cylinder is advanced by pressing push buttons PB1. Draw the pneumatic circuit, PLC wiring diagram and ladder diagram to implement this task.

Solution is shown in Figure 1.13



Assignment Table	
PB1	11
S2	13
Y1	01
Y2	02

Figure 1.13 a) Pneumatic diagram b) wiring diagram c) ladder diagram

PLC Wiring diagram and Ladder diagrams are shown in Figure 1.7. when the pushbutton PB1 is pressed state of the address I1 turns to 1 and thus there will be output 01. The output of 01 operates the solenoid Y1 and cylinder moves forward,

When the cylinder reaches the extreme forward position, and Limit switch S2 is operated , the state of address I3 turns to 1 and thus there will be output 02. The output of 02 operates the solenoid Y2 and cylinder return back to initial position.

Example 4 : Double acting cylinder is used to perform pressing operation. Cylinder has to move forward when PB1 button is pressed and return for set time of 20 seconds before it automatically returns to intial position. Limit switch S2 is used for end sensing of the forward motion of the cylinder. Draw the pneumatic circuit, PLC wiring diagram and ladder diagram to implement this task.

Solution is shown in Figure 1.14

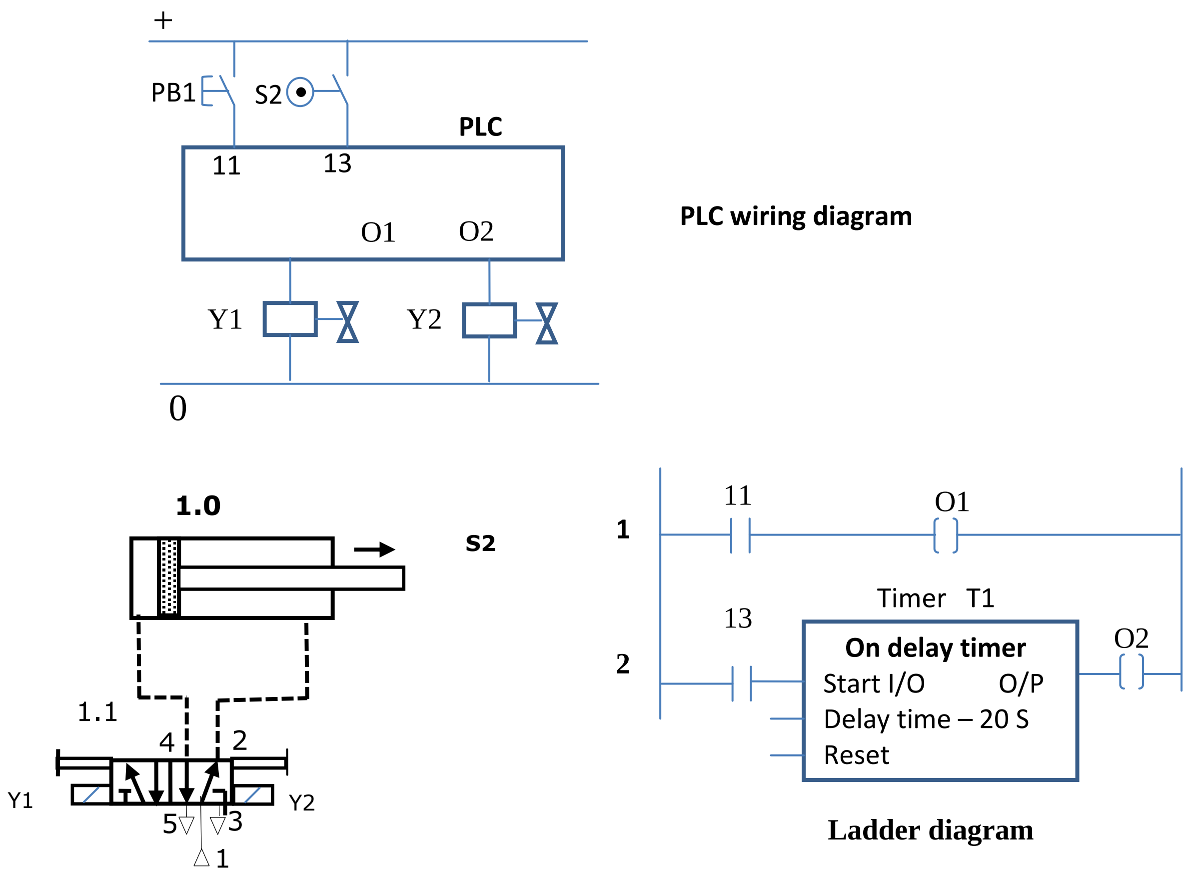
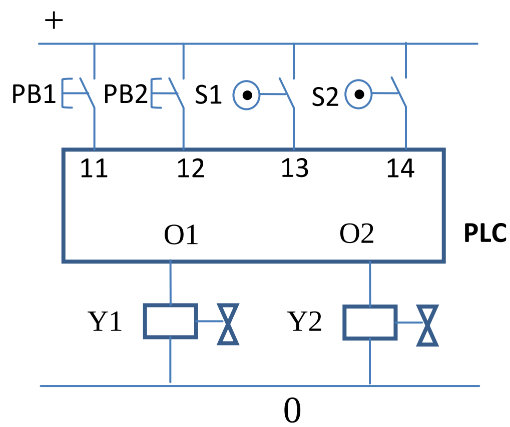


Figure 1.14 a) Pneumatic diagram b) wiring diagram c) ladder diagram

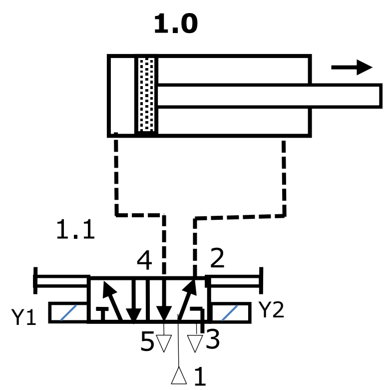
When PB1 is pressed, address I1 input state goes to 1 and there is an output at O1. Due to output at O1, the solenoid coil Y1 is operated and cylinder moves forward. When cylinder reaches end position , limit switch S2 is operated and as a result address I3 changes to 1 and consequently starts the timer T1. The signal state of timer T1 changes to 1 after 20 seconds is reached. At the end of 20 seconds there will be output from Timer T1 set output O2. Coil Y2 is energised thus causing the return motion of the cylinder.

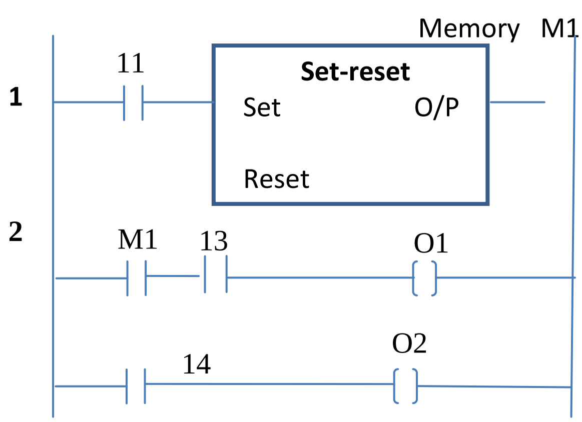
Example 5 : Double acting cylinder is used to perform continuous to and fro motion. Cylinder has to move forward when PB1 button is pressed and once to and fro reciprocation starts it should continue till stop button PB2 is pressed. Limit swithces are used for end position sensing. Draw the pneumatic circuit, PLC wiring diagram and ladder diagram to implement this task.

Solution is shown in Figure 1.15



PLC wiring diagram





Memory signal from I1 sets and Memory signal from I2 resets.

M1=1, I3=1, Output O1 =1, Y1 is ON

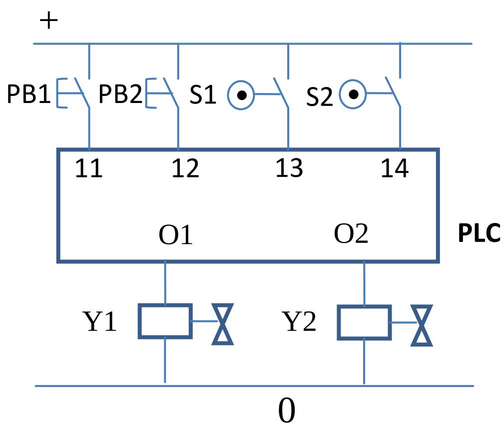
I4=1 , output O2 =1, Y2 is ON

Figure 1.15 a) Pneumatic diagram b) wiring diagram c) ladder diagram

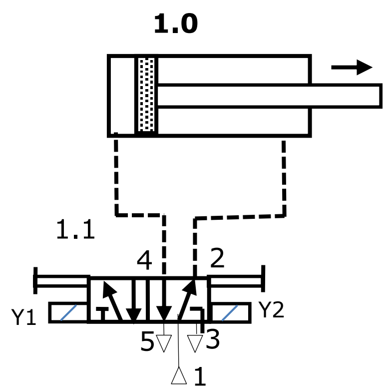
The start and stop operations can be implemented using memory flag with address M1 that is set by PB1 and reset by PB2. The state of the memory element M1 is scanned through an NO contact, is combined in series with the state of sensor S1 to get start and stop controls.

Example 6 : Double acting cylinder is used to perform to and fro operation. Cylinder has to move forward when PB1 button is pressed and continue to and fro motion till 10 cycles of operations is performed. Draw the pneumatic circuit, PLC wiring diagram and ladder diagram to implement this task.

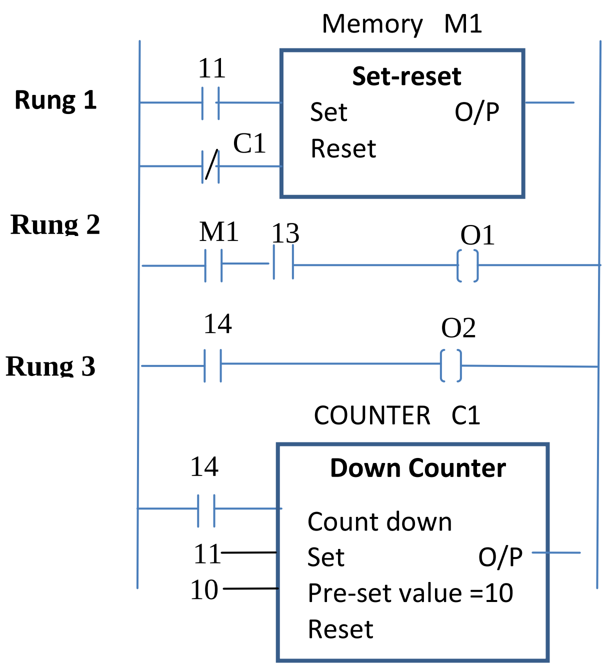
Solution is shown in Figure 1.16



PLC wiring diagram



Pneumatic diagram



I1 sets flag M1, C2 resets M1. NC of C1 scans for Zero state of counter C1

M1=1, I3=, O1=1 Y1 is ON

I4=, O2=1 Y2 is ON

Ladder diagram

The fully automatic operation of cylinder can be obtained as earlier using limit switch S1 and S2. Start and stop operation can be implemented using memory flag with address M1 that is set by PB1 at I1 and reset by NC contact of a down counter. The state of memory flag M1 scanned through an NO contact (rung 2) is combined in series with the state sensor S1 to get start and stop controls.

Example 7 : Draw the pneumatic circuit, PLC wiring diagram and ladder diagram to implement A+B+B-A- sequence.

Solution is shown in Figure 1.17 and Figure 1.18

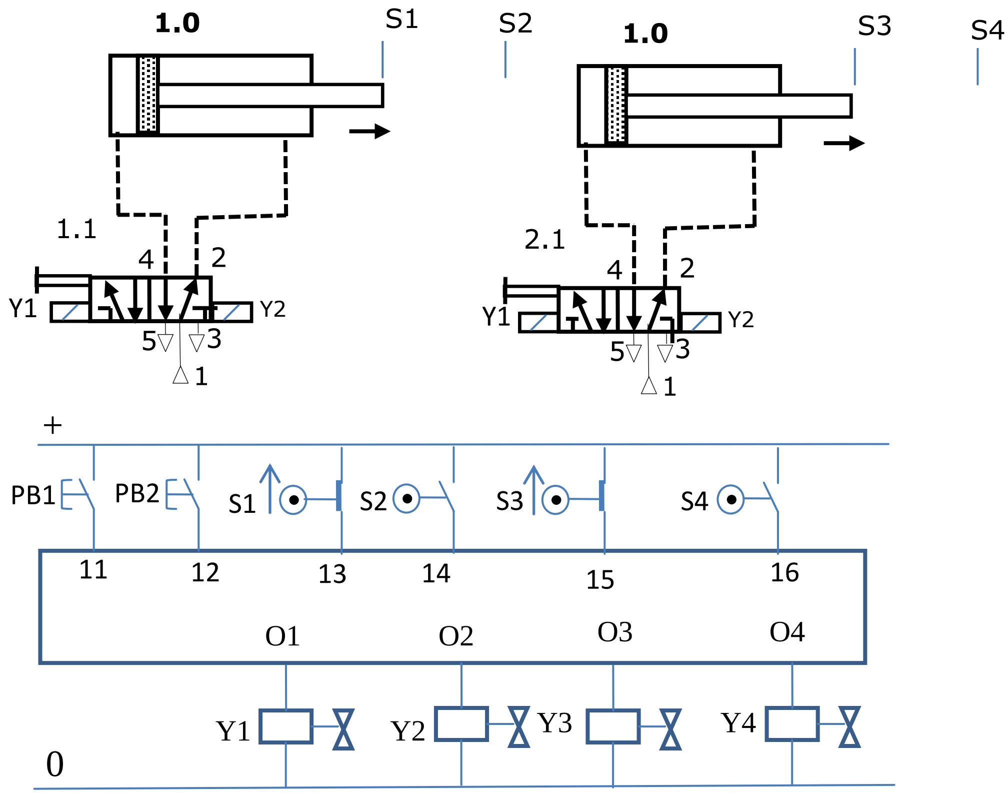


Figure 1.17 PLC wiring diagram

In this sequence circuit, PB2 is used to initiate the program. Pressing PB2 causes the last memory state M4 to set and all other memory flags M1, M2 and M3 to reset. Initially S1 and S3 are actuated and generate outputs.

Condition 1: Pressing PB1 sets Memory flag M1 and resets Memory flag M4. Solenoid Y1 is energised. Cylinder A extends (A+). Sensor S1 is deactivated once A travels and S2 is activated when end position is reached.

Condition 2: When S2 is actuated, memory M2 is set and Memory flag M1 is reset. Solenoid Y3 is energised. Cylinder B extends (B+). Sensor S3 is deactivated once B travels and S4 is activated when end position is reached.

Condition 3: When S4 is actuated, memory M3 is set and Memory flag M2 is reset. Solenoid Y4 is energised. Cylinder B retracts (B-). Sensor S4 is deactivated once B travels and S3 is activated when initial position is reached

Condition 4: When S3 is actuated, memory M4 is set and Memory flag M3 is reset. Solenoid Y2 is energised. Cylinder A retracts (A-). Sensor S2 is deactivated once B travels and S1 is activated when initial position is reached

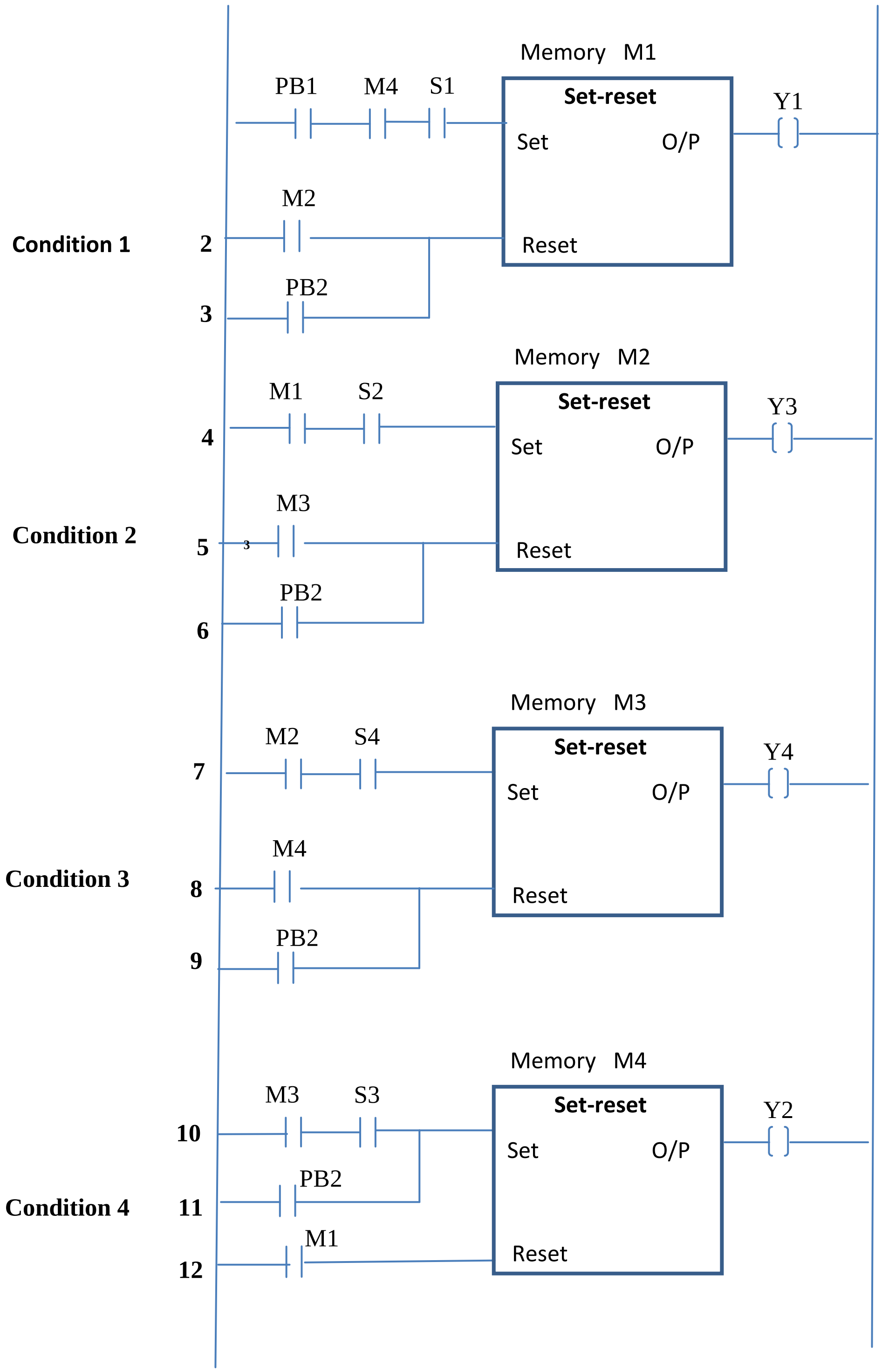


Figure 1.18 Ladder diagram

1.11 Areas of Application of a PLC

Every system or machine has a controller, depending on the type of technology used, controller can be divided into pneumatic, hydraulic, electrical and electronic controllers. Frequently we use combination of different technologies. Furthermore, differentiation is made between hard wired programmable and programmable logic controller. The first is used primarily in cases, where any reprogramming by the user is out of the question and job size warrants the development of a special controller. Typical application for such controllers can be found in automatic washing machine, video cameras, and cars.

However, if the job size does not warrant the development of special controllers or if the user is to have the facility of making simple or independent program changes, or of setting timers and counters, then the use of universal controllers, where the program is written to an electronic memory, is the preferred option. The PLC represents such a universal controller. It can be used for different applications and via the program installed in its memory , provides the user with a simple means of changing , extending and optimising control processes.

PLC are widely used in Industries due to following reasons.

- Cost of PLC automation is less and PLC is very versatile
- PLC can be commissioned and installed easily
- Programming of PLC is quite simple. Ladder programming is flexible
- They are not hard wired control. They can be programmed and reprogrammed to accommodate frequent changes in program
- Monitoring of on line work process is easy, therefore trouble shooting and maintenance of PLC is not a difficult task
- They can be classified as low cost automation devices
- They can be used in harsh environment where humidity and temperature are high. Their working is not effected by vibration and shock
- They can be used to execute complex mathematical algorithms, servomotor control, Stepping control, axis control, self – diagnosis, on line monitoring, condition monitoring, system trouble shooting, communicating to other PLCs, data acquisition, Networking, storage and report generation

- PLC are most suitable for low cost automation, where frequent changes to the control requirement would be expected during their operational life like in Batch type of production systems.

1.12 PLC standards

Previously valid PLC standards focussing mainly on PLC programming were generally used in Europe at end of the seventies. This included non-networked PLC system, which primarily execute logic operations on binary signals. DIN 19 239, for example, specifies programming language which possess the corresponding language commands for these applications.

Since 1992, an international standard now exists for programmable logic controllers and associated peripheral devices like programming and diagnostic tools, testing equipment, man to machine interfaces.

In 1992, IEC 1131 standards were developed as an open framework for PLC architecture. The second edition of IEC 1131 (Known as IEC 61131) was published in 2003. The new IEC standard consists of five parts

- Part 1 : General information
- Part 2 : Equipment requirement and tests
- Part 3 : programming languages
- Part 4 : User guidelines in preparation with IEC
- Part 5: Messaging service specification

Parts 1 to 2 of this standard were adopted unamended as European standard EN 61 131, parts 1 to 2. As such, they also hold the status of a German standard. Part 3 of new IEC deals with programming languages and defines two graphical and two textural PLC programming language standards. The standard also defines both graphical and textual sequential function chart elements to organise programs for sequential and parallel control processing. It is now possible to program PLC using following languages

- IL- Instruction list
- ST – structural text
- LD – Ladder diagram
- FBD - Functional block diagram
- SFC- Sequential function chart

The purpose of the new standard was to define and standardize the design and functionality of a PLC and the languages required for programming to the extent where users were able to operate using different PLC systems without any particular difficulties.

Large number o f major PLC suppliers are members of association called PLCopen which supports IEC 1131. Allen Bradley, Klockner–Moeller, Phillips Siemens or Mitsubishi to mention a few.

Objective Type Questions

1. The most important sections of PLC are input section, output section and ------
2. The Structure of PLC consists of power supply module, CPU, I/O modules and --------
3. User program can be written in any standard PLC programming language like statement list and ------------------
4. The AND function combines the bit addresses of inputs and produces an RLO (Results of logic operation) of ..... when all the inputs are scanned for 1
5. The function of timer is to provide ------------------ between work operations

State True or False

1. Hard wired control systems are used widely when production requirements change regularly

2. Relay controls are less expensive compared to PLC controls

3 While processing a PLC program, CPU scans and executes the main program cyclically.

4. The NO and NC PLC program contact is same as the hardware NO and NC contacts.

5. The OR function combines the bit addresses of inputs and produces an RLO (Results of logic operation) of 1 when any one or more of inputs are scanned for 1.

Review Questions

1. What is a hard wired control? what are its disadvantages

2. List five differences between PLC control and Relay control

3. List three input devices commonly used in PLC control

4. Explain the working of functions of CPU , Input and output module and memory in PLC

5. Mention few applications of PLC

6. What are the functions of timer in PLC

7. Briefly explain the structure of PLC

8. Differentiate between ON time delay and OFF time delay with help of symbols

9. What is meant by bit logic operations in relation to a PLC

10. Give functions of following PLC program elements a) Program coil b) NC contact c) NO contact d) Set coil e) reset coil f) Set-reset box g) reset-set box h) on delay timer i) off delay timer j) up counter k) down counter

Answers

Fill in the Blanks

1. Program section
2. Software
3. Ladder diagram
4. 1(one)
5. delay

State True or False

1. False
2. False
3. True
4. False
5. True