I.C. Engine Cycles

Thermodynamic Analysis

AIR STANDARD CYCLES

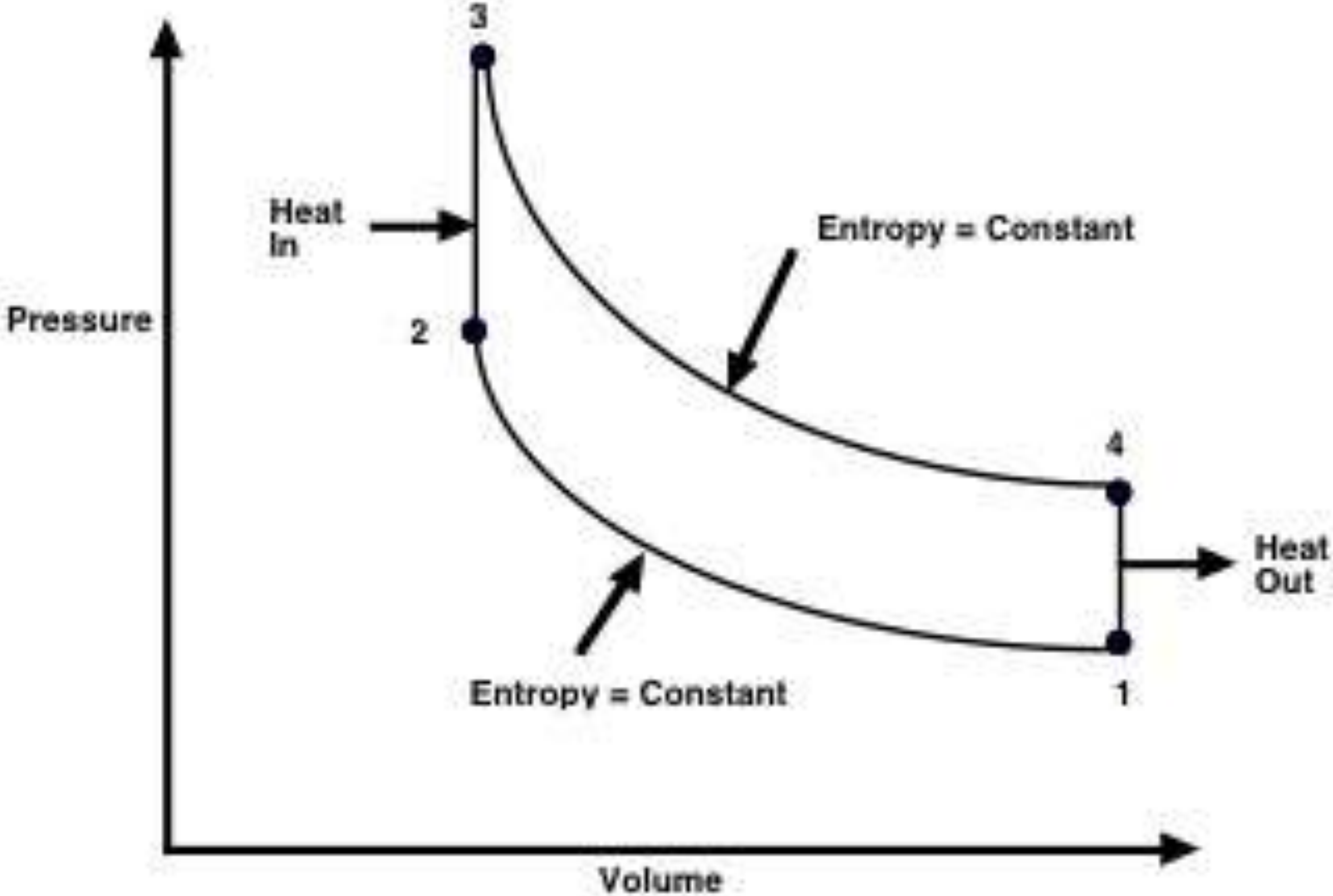
Air as a perfect gas

All processes ideal and reversible

Mass same throughout

Constant Specific Heat

1. OTTO CYCLE



Basic Otto Cycle

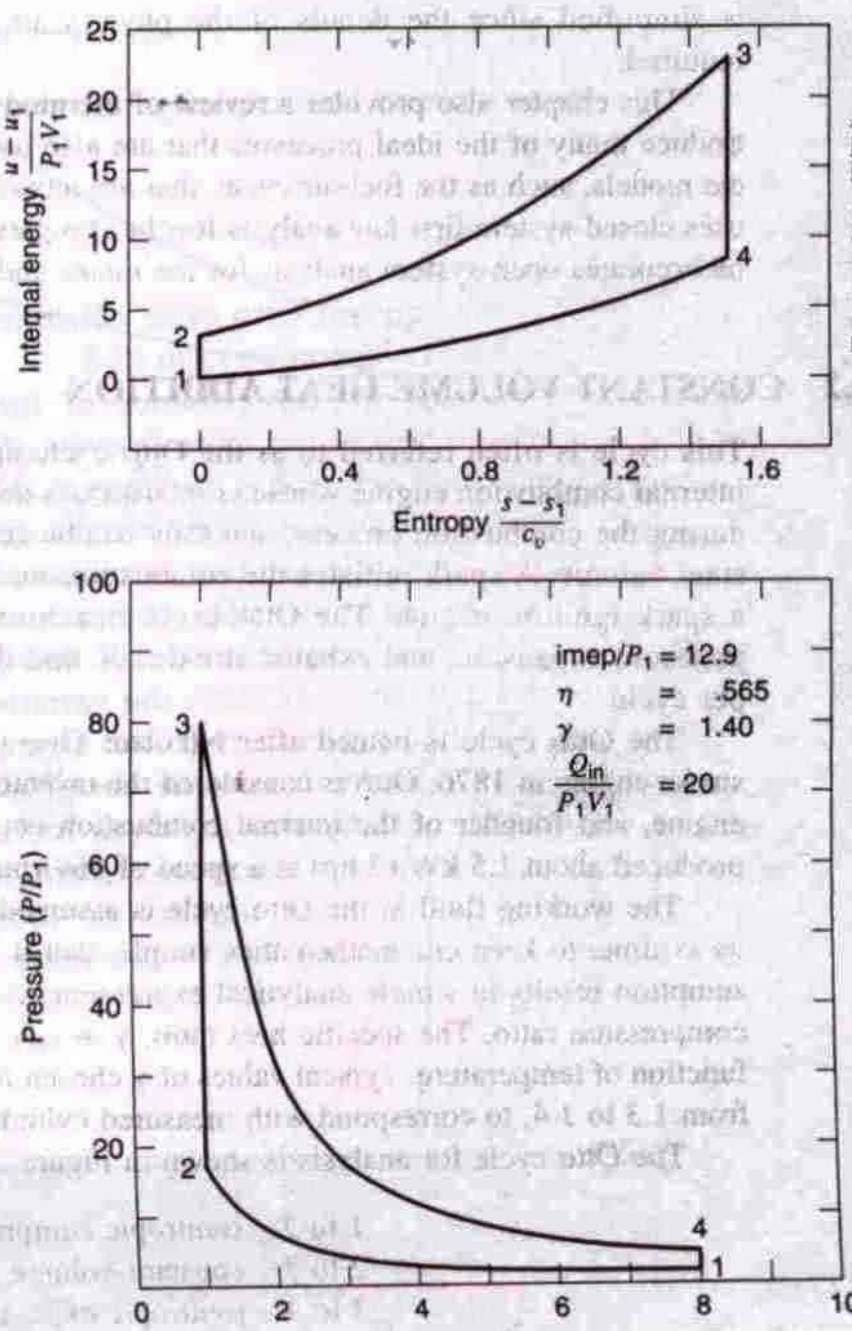


Figure 2-2 The Otto cycle.

OTTO CYCLE

Efficiency is given by

$$\eta = 1 - \frac{1}{r^{\gamma - 1}}$$

Efficiency increases with increase in compression ratio and specific heat ratio (γ) and is independent of load, amount of heat added and initial conditions.

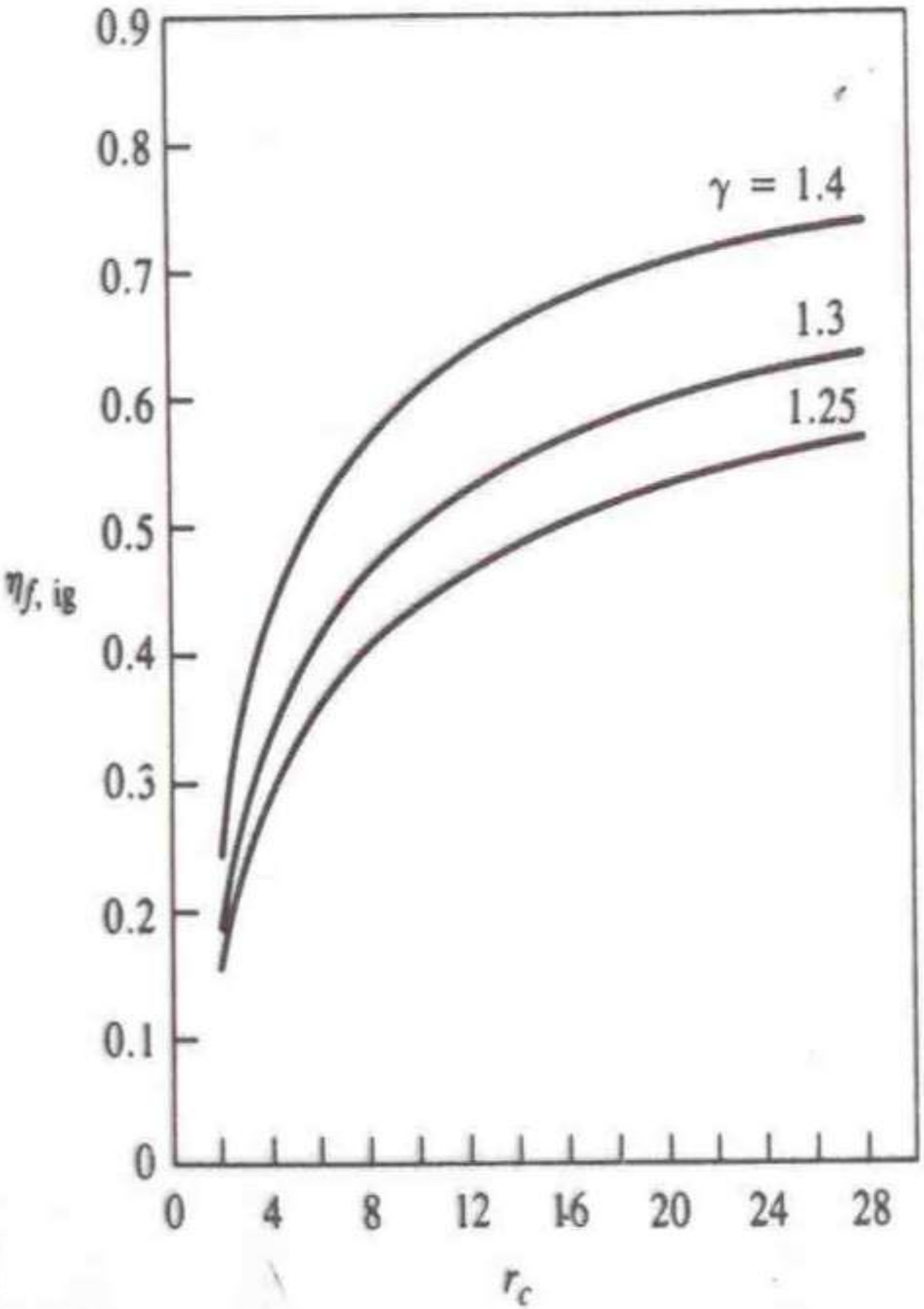


FIGURE 5-5
Ideal gas constant-volume cycle fuel conversion efficiency as a function of compression ratio; $\gamma = c_p/c_v$.

Efficiency of Otto cycle is given as

$\eta = 1 - (1/r^{(\gamma-1)})$

$\Gamma = 1.4$

r	η
1	0
2	0.242
3	0.356
4	0.426
5	0.475
6	0.512
7	0.541
8	0.565
9	0.585
10	0.602
16	0.67
20	0.698
50	0.791

CR ↑from 2 to 4, efficiency ↑ is 76%

CR ↑from 4 to 8, efficiency ↑ is 32.6%

CR ↑from 8 to 16, efficiency ↑ is 18.6%

OTTO CYCLE
Mean Effective Pressure

It is that constant pressure which, if exerted on the piston for the whole outward stroke, would yield work equal to the work of the cycle. It is given by

$$mep = \frac{W}{V_1 - V_2}$$

$$= \frac{\eta Q_{2-3}}{V_1 - V_2}$$

OTTO CYCLE
Mean Effective Pressure

We have:

$$V_1 - V_2 = V_1\left(1 - \frac{V_2}{V_1}\right)$$

$$= V_1\left(1 - \frac{1}{r}\right)$$

Eq. of state:

$$V_1 = M\,\frac{R_0}{m}\,\frac{T_1}{p_1}$$

To give:

$$mep = \eta\,\frac{Q_{2-3}\,\dfrac{p_1 m}{M R_0 T_1}}{1 - \dfrac{1}{r}}$$

OTTO CYCLE
Mean Effective Pressure

The quantity $Q_{2\text{-}3}/M$ is heat added/unit mass equal to Q', so

$$mep = \eta \frac{Q' \dfrac{p_1 m}{R_0 T_1}}{1 - \dfrac{1}{r}}$$

OTTO CYCLE
Mean Effective Pressure

Non-dimensionalizing mep with p_1 we get

$$\frac{mep}{p_1} = \eta \left[\frac{1}{1 - \frac{1}{r}} \right] \left[\frac{Q' m}{R_0 T_1} \right]$$

Since:

$$\frac{R_0}{m} = c_v (\gamma - 1)$$

OTTO CYCLE
Mean Effective Pressure

We get

$$\frac{mep}{p_1} = \eta \frac{Q'}{c_v T_1} \frac{1}{\left[1 - \frac{1}{r}\right][\gamma - 1]}$$

Mep/p_1 is a function of heat added, initial temperature, compression ratio and properties of air, namely, c_v and γ

Choice of Q'

We have

$$Q' = \frac{Q_{2-3}}{M}$$

For an actual engine: $Q_{2-3} = M_f Q_c$

$$= F M_a Q_c \ in\ kJ/cycle$$

F=fuel-air ratio, M_f/M_a

M_a=Mass of air,

Q_c=fuel calorific value

Choice of Q'

We now get: $$Q' = \frac{FM_a Q_c}{M}$$

$$Now \quad \frac{M_a}{M} \approx \frac{V_1 - V_2}{V_1}$$

$$And \quad \frac{V_1 - V_2}{V_1} = 1 - \frac{1}{r}$$

Thus:

$$Q' = FQ_c\left(1 - \frac{1}{r}\right)$$

Choice of Q'

For isooctane, FQ_c at stoichiometric conditions is equal to 2975 kJ/kg, thus

$Q' = 2975(r - 1)/r$

At an ambient temperature, T_1 of 300K and c_v for air is assumed to be 0.718 kJ/kgK, we get a value of $Q'/c_vT_1 = 13.8(r - 1)/r$.

Under fuel rich conditions, $\varphi = 1.2$, $Q'/c_vT_1 = 16.6(r - 1)/r$.

Under fuel lean conditions, $\varphi = 0.8$, $Q'/c_vT_1 = 11.1(r - 1)/r$

OTTO CYCLE
Mean Effective Pressure

We can get mep/p_1 in terms of $r_p = p_3/p_2$ thus:

$$\frac{mep}{p_1} = \frac{r\left(r_p - 1\right)\left(r^{\gamma-1} - 1\right)}{\left(r-1\right)\left(\gamma-1\right)}$$

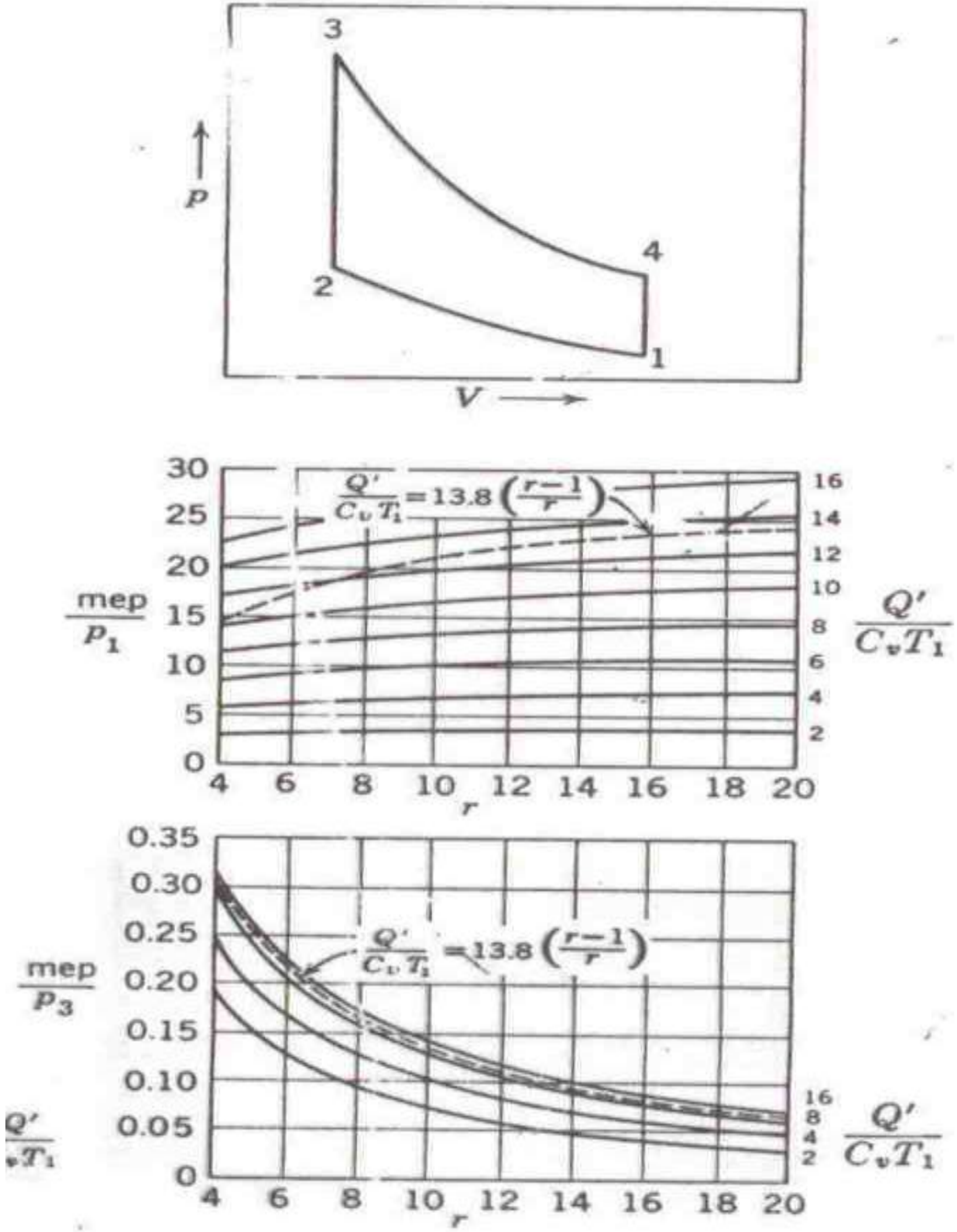
We can obtain a value of r_p in terms of Q' as follows:

$$r_p = \frac{Q'}{c_v T_1 r^{\gamma-1}} + 1$$

OTTO CYCLE
Mean Effective Pressure

Another parameter, which is of importance, is the quantity mep/p_3. This can be obtained from the following expression:

$$\frac{mep}{p_3} = \frac{mep}{p_1}\frac{1}{r^{\gamma}}\frac{1}{\dfrac{Q'}{c_v T_1 r^{\gamma-1}}+1}$$



t-volume air cycles: $r = V_1/V_2$; p = absolute

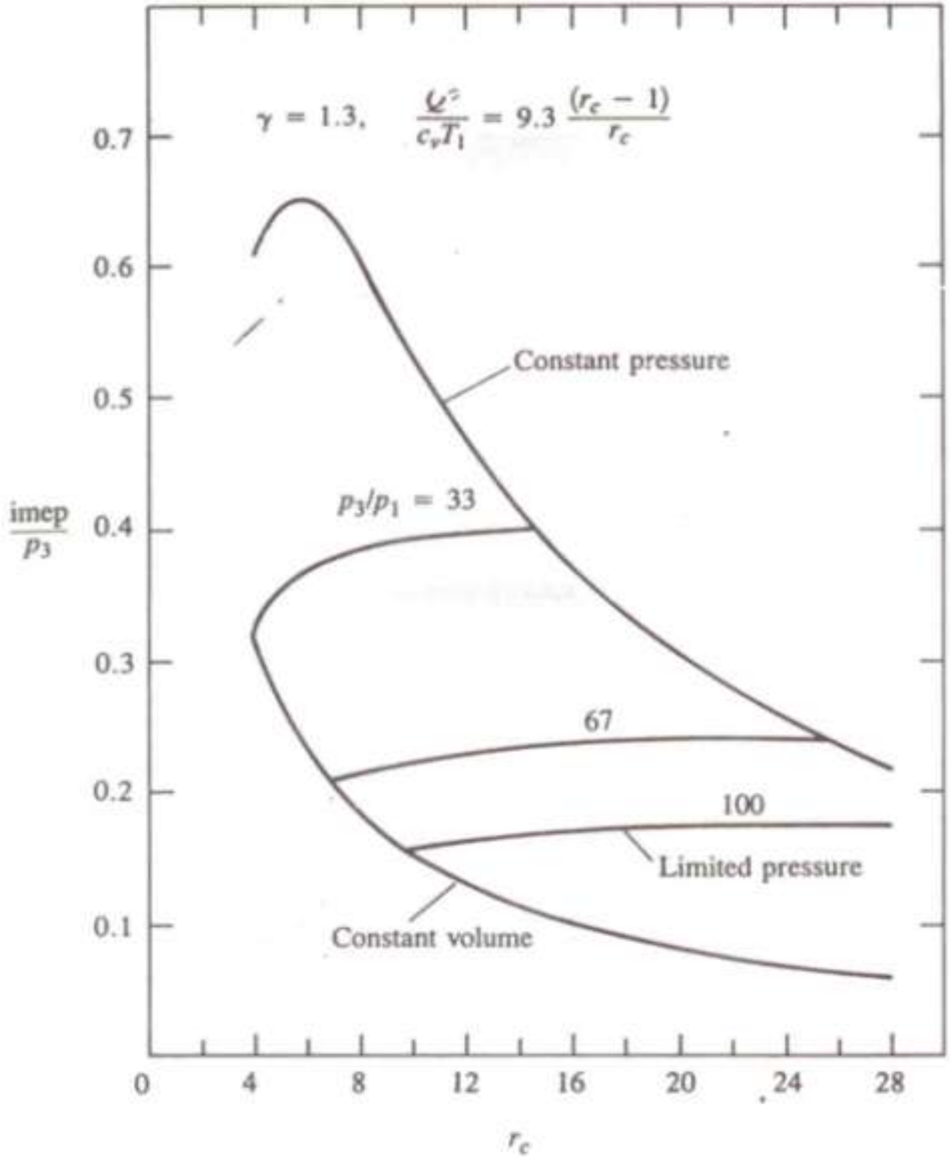
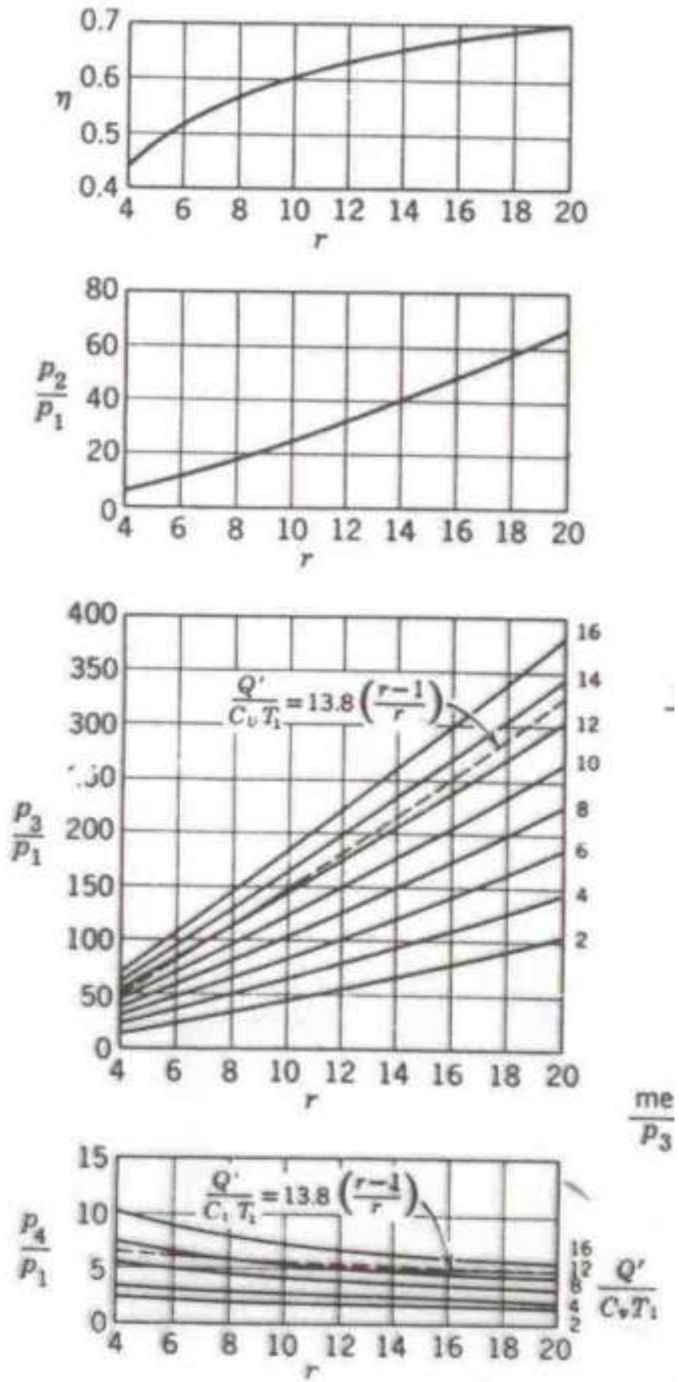
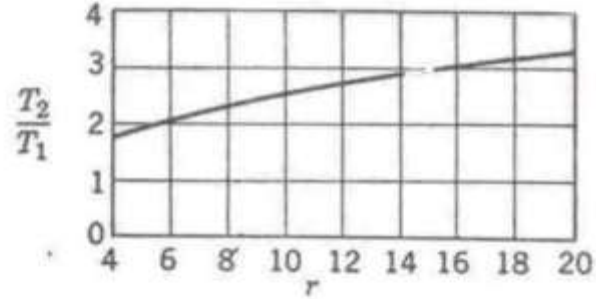


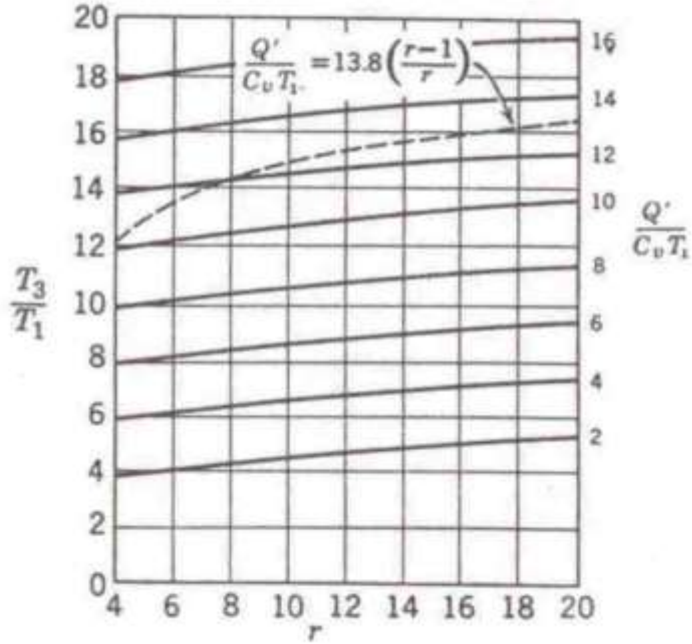
FIGURE 5-8
Indicated mean effective pressure (imep) divided by maximum cycle pressure (p_3) as a function of compression ratio for constant-volume, constant-pressure, and limited-pressure cycles. Details same as Fig. 5-7.



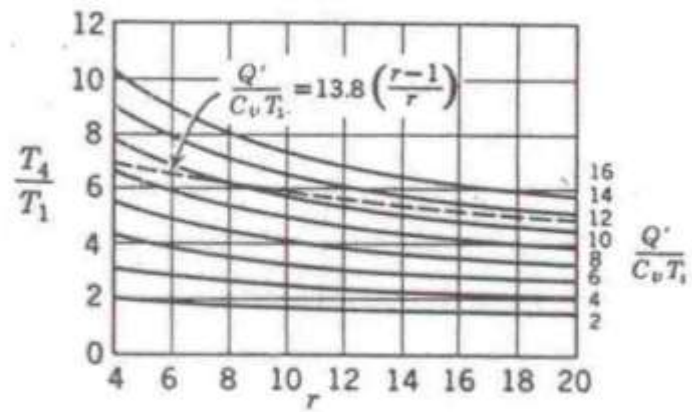
η
0.7
0.6
0.5
0.4
4 6 8 10 12 14 16 18 20
r
$\frac{p_2}{p_1}$
80
60
40
20
0
4 6 8 10 12 14 16 18 20
r
$\frac{Q'}{C_v T_1} = 13.8\left(\frac{r-1}{r}\right)$
$\frac{p_3}{p_1}$
400
350
300
200
150
100
50
0
16 14 12 10 8 6 4 2
4 6 8 10 12 14 16 18 20
r
$\frac{me}{p_3}$
$\frac{Q'}{C_v T_1} = 13.8\left(\frac{r-1}{r}\right)$
$\frac{p_4}{p_1}$
15
10
5
0
16 12 8 4 2
$\frac{Q'}{C_v T_1}$
4 6 8 10 12 14 16 18 20
r



$\frac{T_2}{T_1}$
r



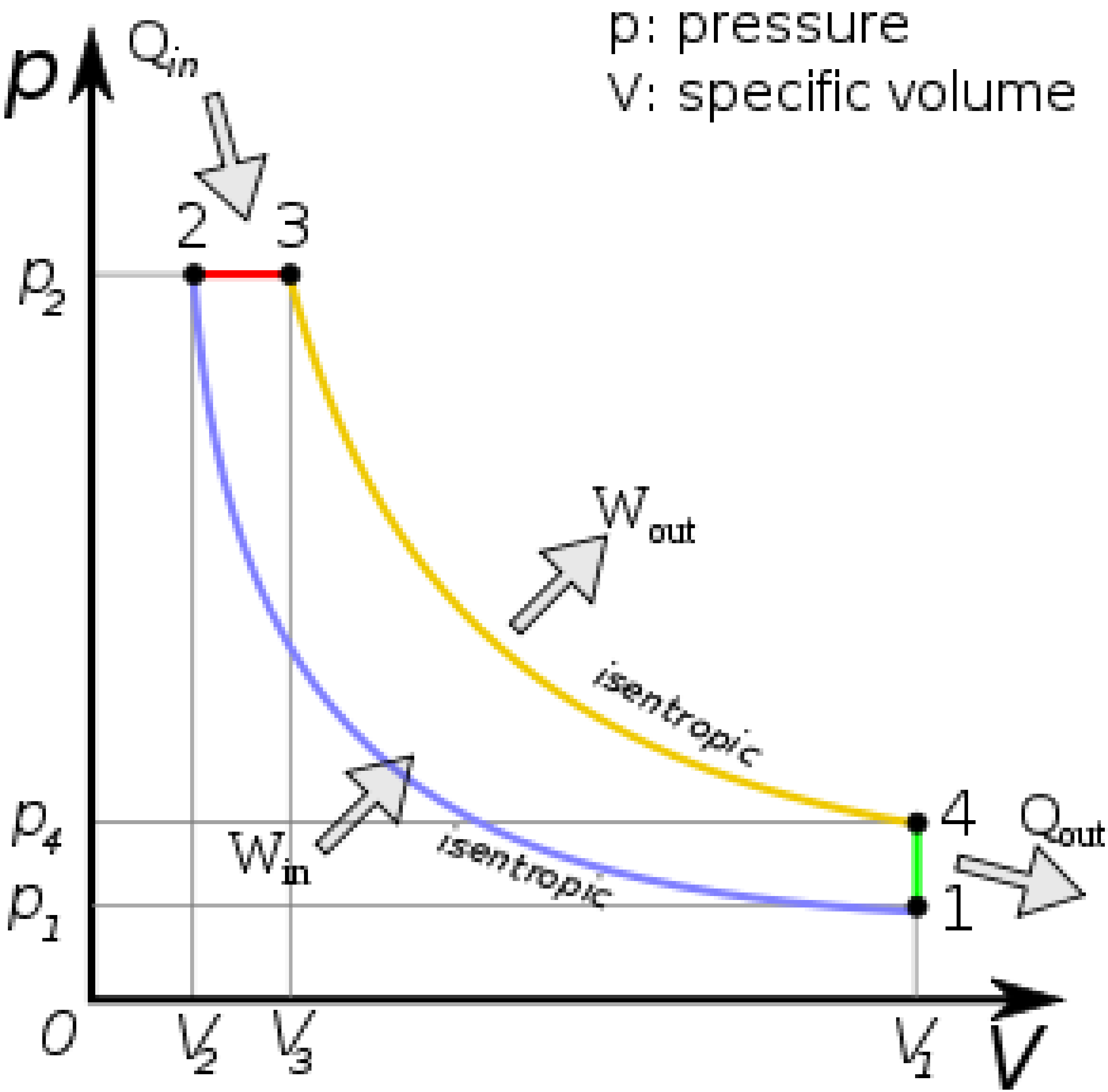
$\frac{Q'}{C_v T_1} = 13.8\left(\frac{r-1}{r}\right)$
$\frac{T_3}{T_1}$
$\frac{Q'}{C_v T_1}$
r



$\frac{Q'}{C_v T_1} = 13.8\left(\frac{r-1}{r}\right)$
$\frac{T_4}{T_1}$
$\frac{Q'}{C_v T_1}$
r

Air Standard Cycles

2. DIESEL CYCLE



p: pressure
V: specific volume
p
Q_in
2
3
p_2
W_out
isentropic
p_4
W_in
isentropic
4
Q_out
p_1
1
O
V_2
V_3
V_1
V

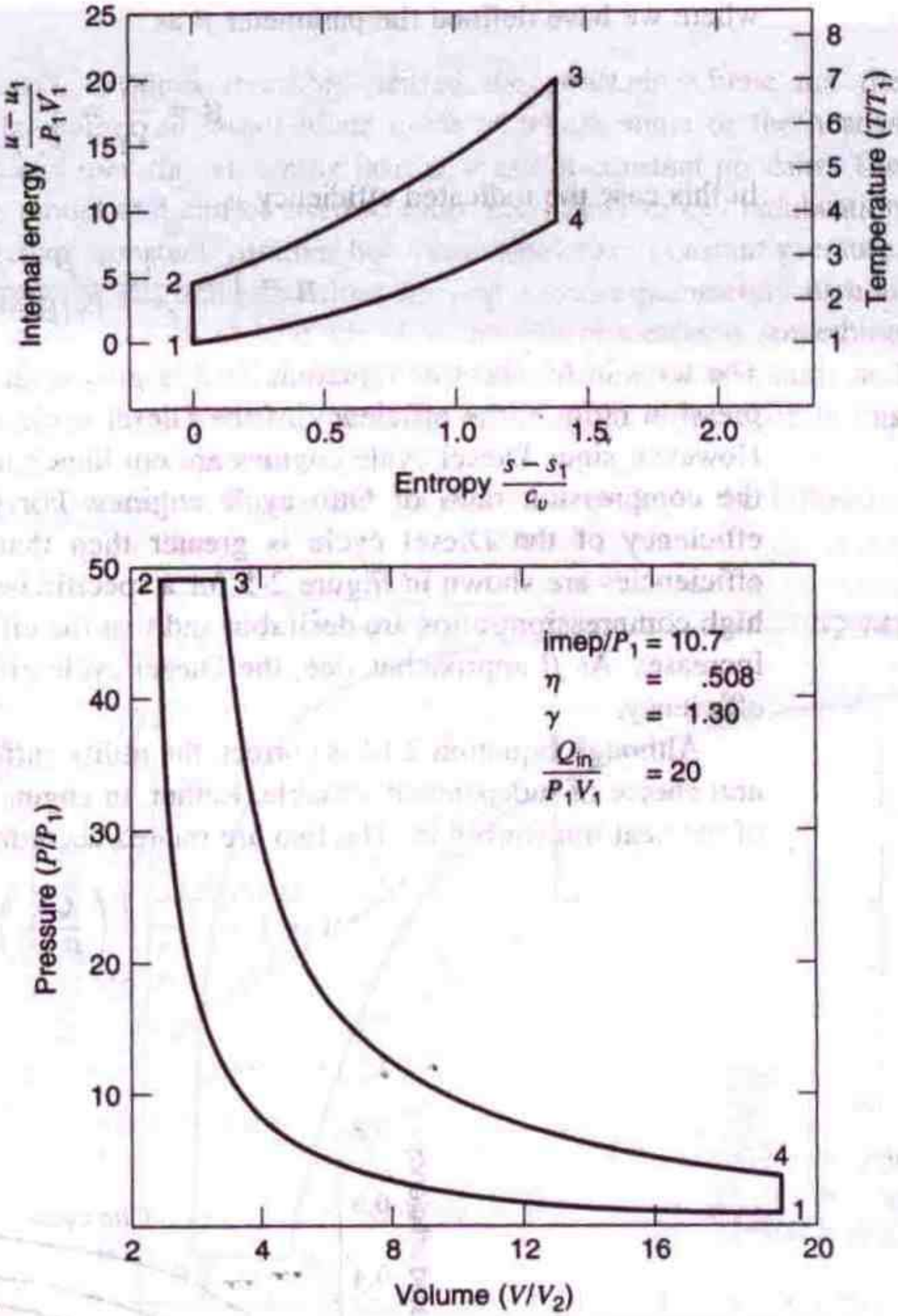


Figure 2-4 The Diesel cycle.

Diesel Cycle

Thermal Efficiency of cycle is given by

$$\eta = 1 - \frac{1}{r^{\gamma-1}}\left[\frac{r_c^{\gamma}-1}{\gamma(r_c-1)}\right]$$

r_c is the cut-ff ratio, V_3/V_2

We can write r_c in terms of Q':

$$r_c = \frac{Q'}{c_p T_1 r^{\gamma-1}} + 1$$

We can write the mep formula for the diesel cycle like that for the Otto cycle in terms of the η, Q', γ, c_v and T_1:

$$\frac{mep}{p_1} = \eta \frac{Q'}{c_v T_1} \frac{1}{\left[1 - \frac{1}{r}\right][\gamma - 1]}$$

Diesel Cycle

We can write the mep in terms of γ, r and r_c:

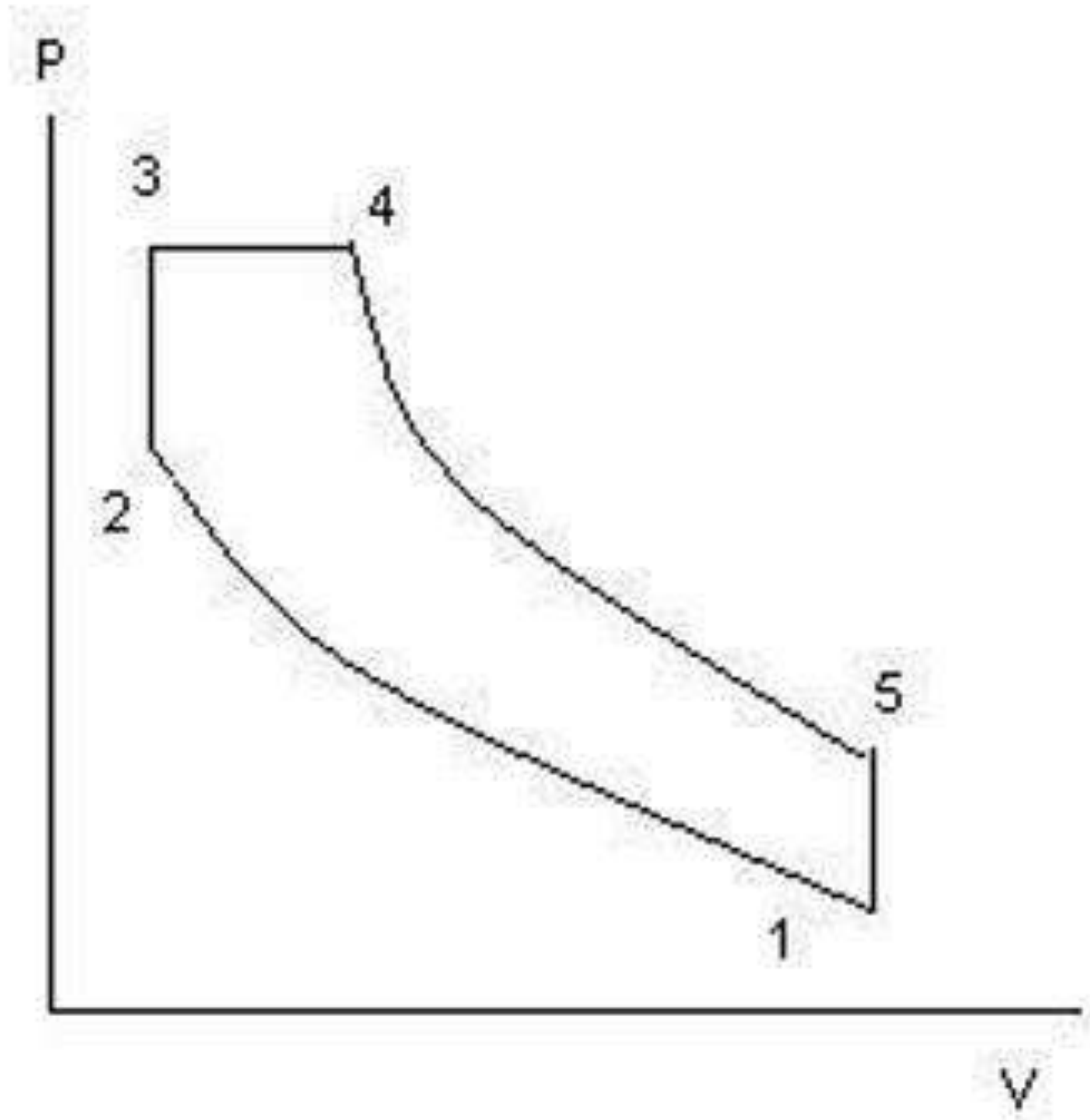
$$\frac{mep}{p_1} = \frac{\gamma\, r^{\gamma}\left(r_c - 1\right) - r\left(r_c^{\gamma} - 1\right)}{\left(r - 1\right)\left(\gamma - 1\right)}$$

The expression for mep/p_3 is:

$$\frac{mep}{p_3} = \frac{mep}{p_1}\left(\frac{1}{r^{\gamma}}\right)$$

Air Standard Cycle

3. DUAL CYCLE


P
3
4
2
5
1
V

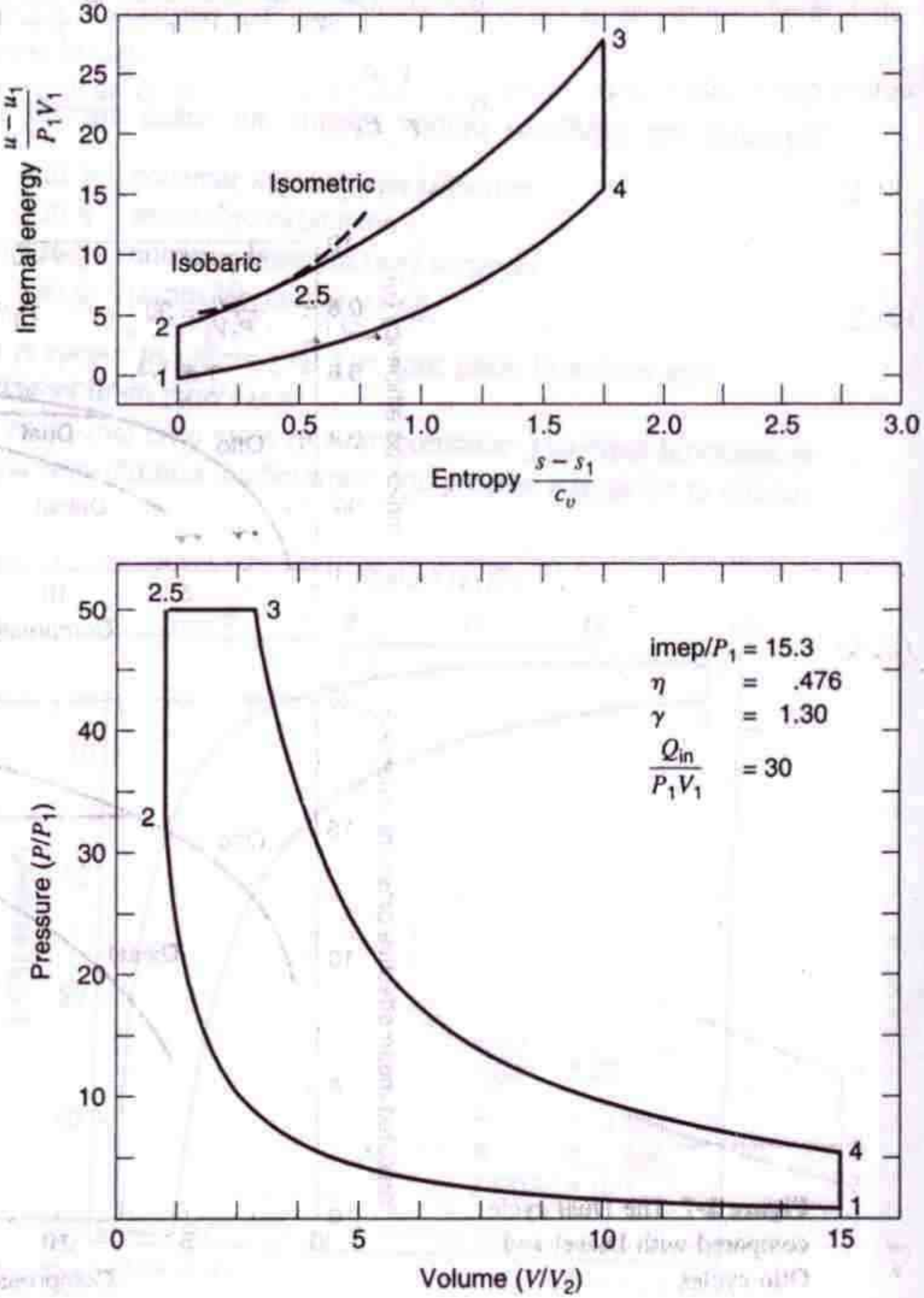


Figure 2-6 The Dual cycle.

Dual Cycle

The Efficiency is given by

$$\eta = 1 - \frac{1}{r^{\gamma-1}}\left[\frac{r_p\, r_c^{\gamma} - 1}{\left(r_p - 1\right) + \gamma\, r_p\left(r_c - 1\right)}\right]$$

We can use the same expression as before to obtain the mep.

We can obtain the mep in terms of the cut-off and pressure ratios as before. This is given in the next slide.

Dual Cycle

$$\frac{mep}{p_1} = \frac{\gamma\, r_p\, r^{\gamma}\left(r_c - 1\right) + r^{\gamma}\left(r_p - 1\right) - r\left(r_p\, r_c^{\gamma} - 1\right)}{\left(r - 1\right)\left(\gamma - 1\right)}$$

For the dual cycle, the expression for mep/p_3 is given in the next slide.

Dual Cycle

For the dual cycle, the expression for mep/p_3 is as follows:

$$\frac{mep}{p_3} = \frac{mep}{p_1}\left(\frac{p_1}{p_3}\right)$$

Dual Cycle

We can write an expression for r_p the pressure ratio in terms of the peak pressure which is a known quantity:

$$r_p = \frac{p_3}{p_1}\left(\frac{1}{r^{\gamma}}\right)$$

We can obtain an expression for r_c in terms of Q’ and r_p and other known quantities as follows:

Dual Cycle

$$r_c = \frac{1}{\gamma}\left(\left[\left\{\frac{Q'}{c_v T_1 r^{\gamma-1}}\right\}\frac{1}{r_p}\right] + (\gamma - 1)\right)$$

We can also obtain an expression for r_p in terms of Q' and r_c and other known quantities as follows:

$$r_p = \frac{\left[\frac{Q'}{c_v T_1 r^{\gamma-1}} + 1\right]}{1 + \gamma\, r_c - \gamma}$$

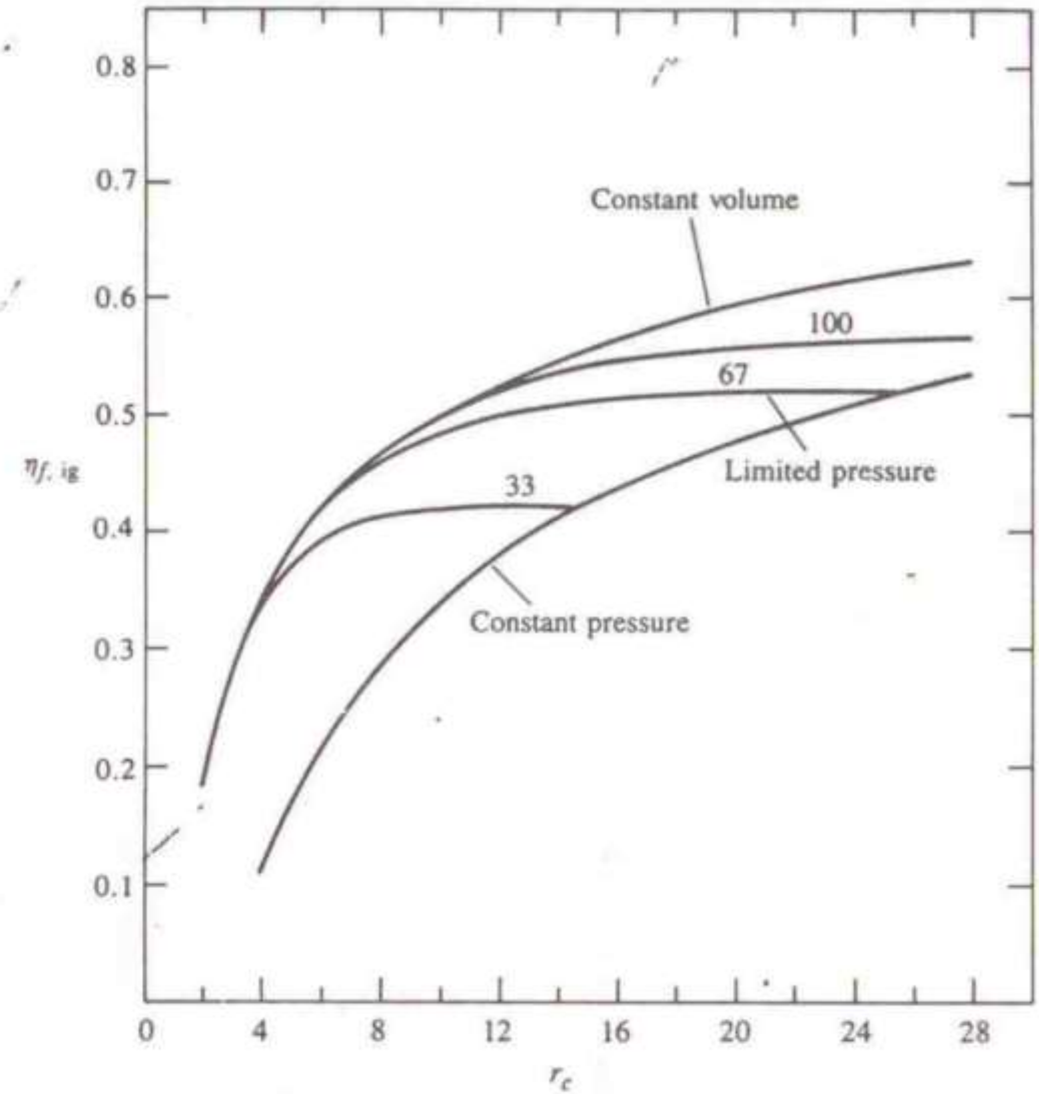


FIGURE 5-7
Fuel conversion efficiency as a function of compression ratio, for constant-volume, constant-pressure, and limited-pressure ideal gas cycles. $\gamma = 1.3$, $Q^*/(c_v T_1) = 9.3(r_c - 1)/r_c$. For limited-pressure cycle, $p_3/p_1 = 33, 67, 100$.

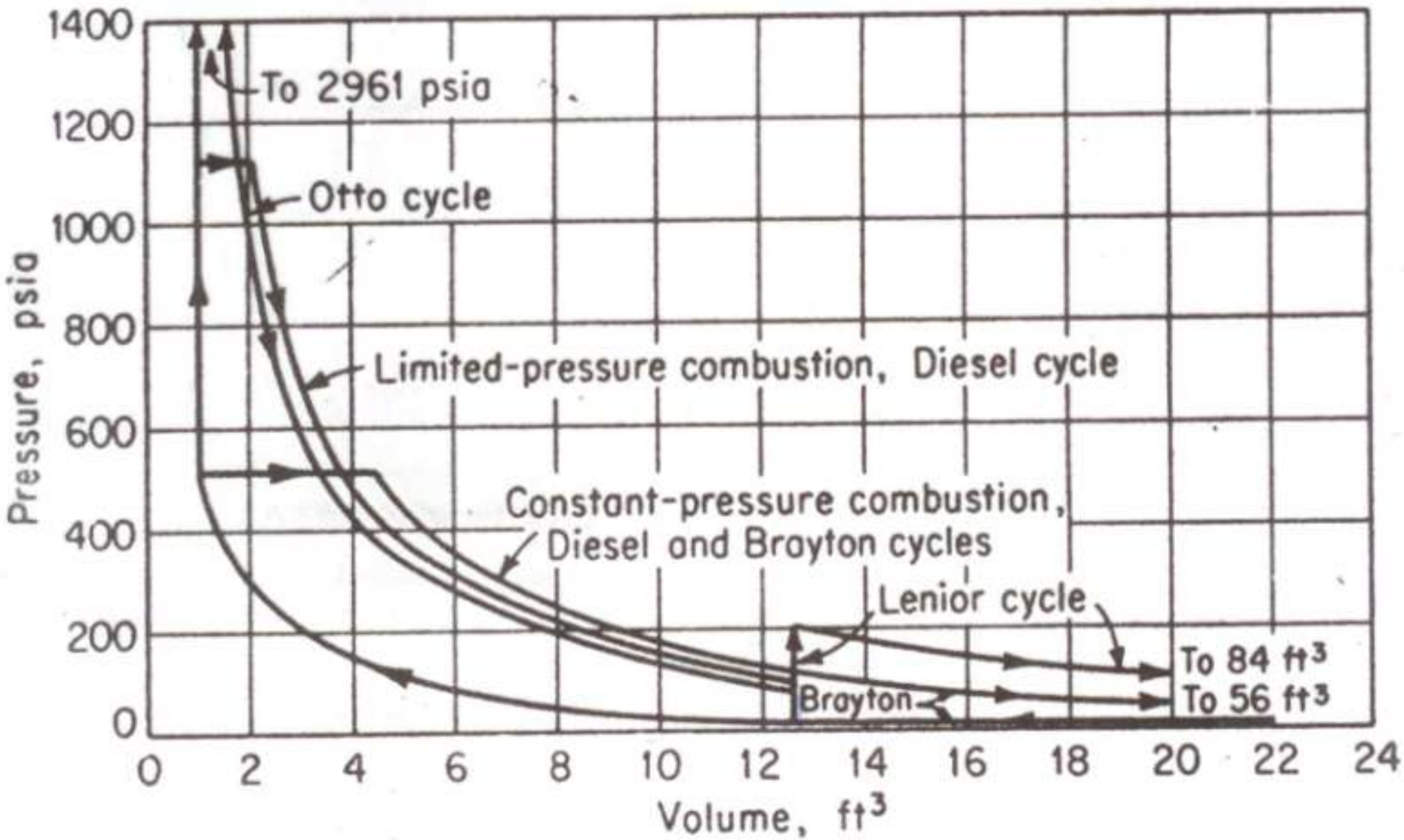


Fig. 3-6 Pressure-volume diagrams of air-standard cycles with a compression ratio of 12.85:1 (except Lenoir) and the same heat input. The Atkinson cycle would be indicated by extending the Otto expansion process to 41 ft³ at 14.7 psia.

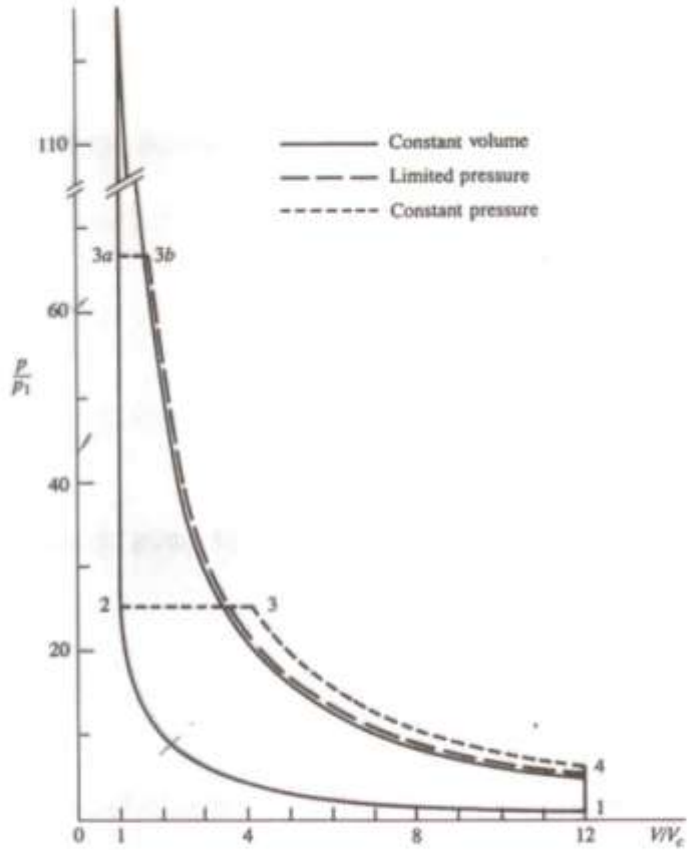


FIGURE 5-6
Pressure-volume diagrams for constant-volume, limited-pressure, and constant-pressure ideal gas standard cycles. $r_c = 12$, $\gamma = 1.3$, $Q^*/(c_v T_1) = 9.3(r_c - 1)/r_c = 8.525$, $p_{3a}/p_1 = 67$.

TABLE 5.2
Comparison of ideal cycle results

	$\eta_{f,i}$	$\dfrac{\text{imep}}{p_1}$	$\dfrac{\text{imep}}{p_3}$	$\dfrac{p_{max}}{p_1}$
Constant volume	0.525	16.3	0.128	128
Limited pressure	0.500	15.5	0.231	67
Constant pressure	0.380	11.8	0.466	25.3

$\gamma = 1.3$; $r_c = 12$; $Q^*/(c_v T_1) = 8.525$.

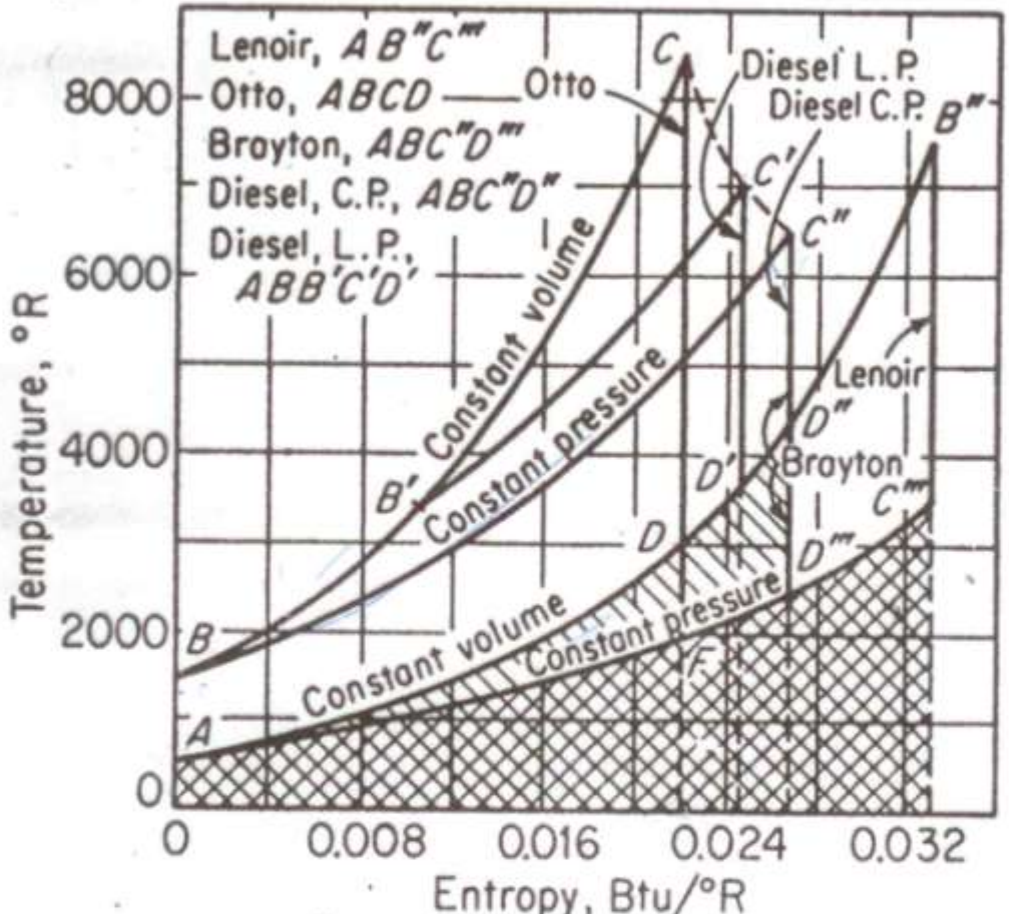


Fig. 3-7 Temperature-entropy diagram of air-standard cycles with the same compression ratio and the same heat input. (Atkinson, *ABCDF*.)

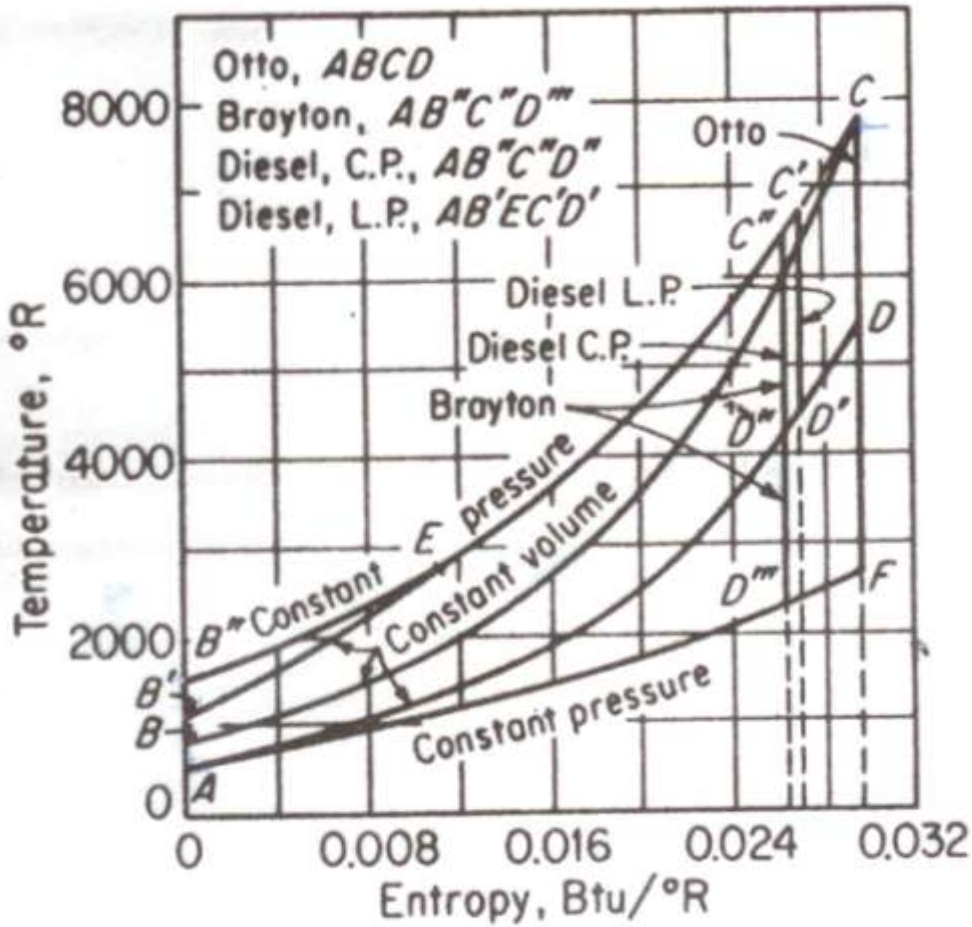
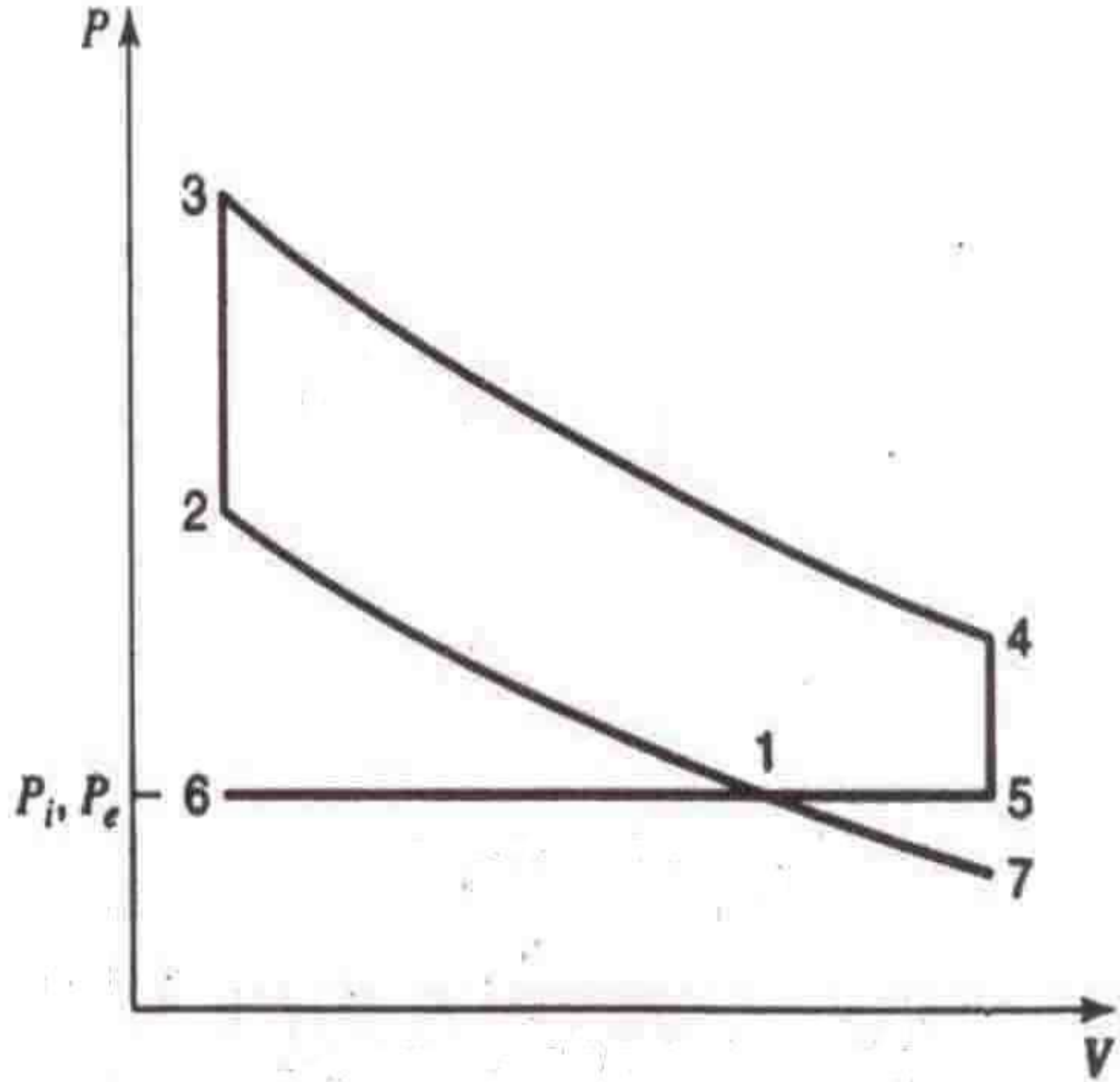


Fig. 3-8 Temperature-entropy diagram of air-standard cycles having the same maximum pressures and heat input. (Atkinson, *ABCDF*.)

Air Standard Cycle

4. MILLER CYCLE



The Miller cycle.

$$\lambda = r_e/r_c$$

rejection has two components

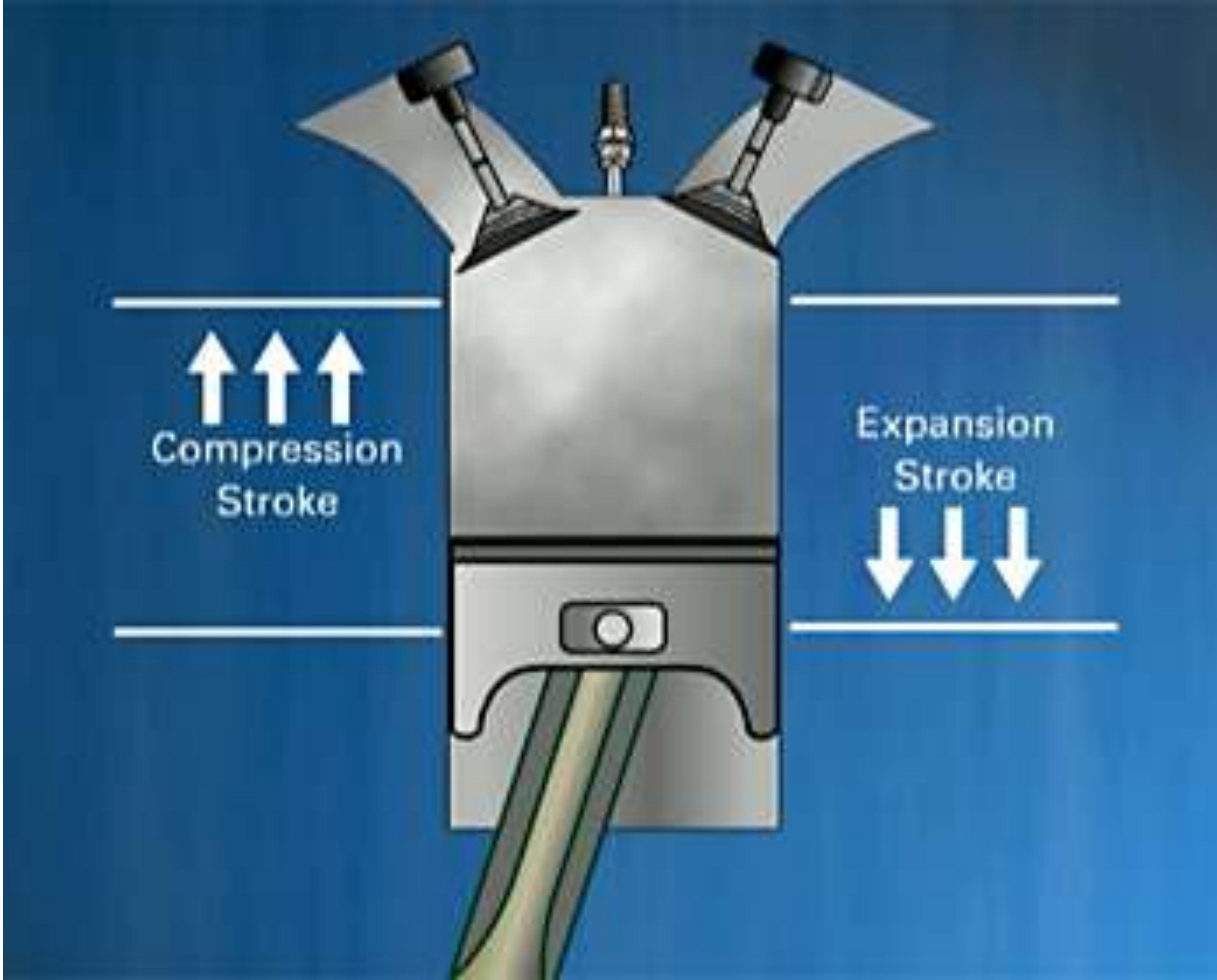
$$Q_{\text{out}} = mc_v(T_4 - T_5) + mc_p(T_5 - T_1)$$

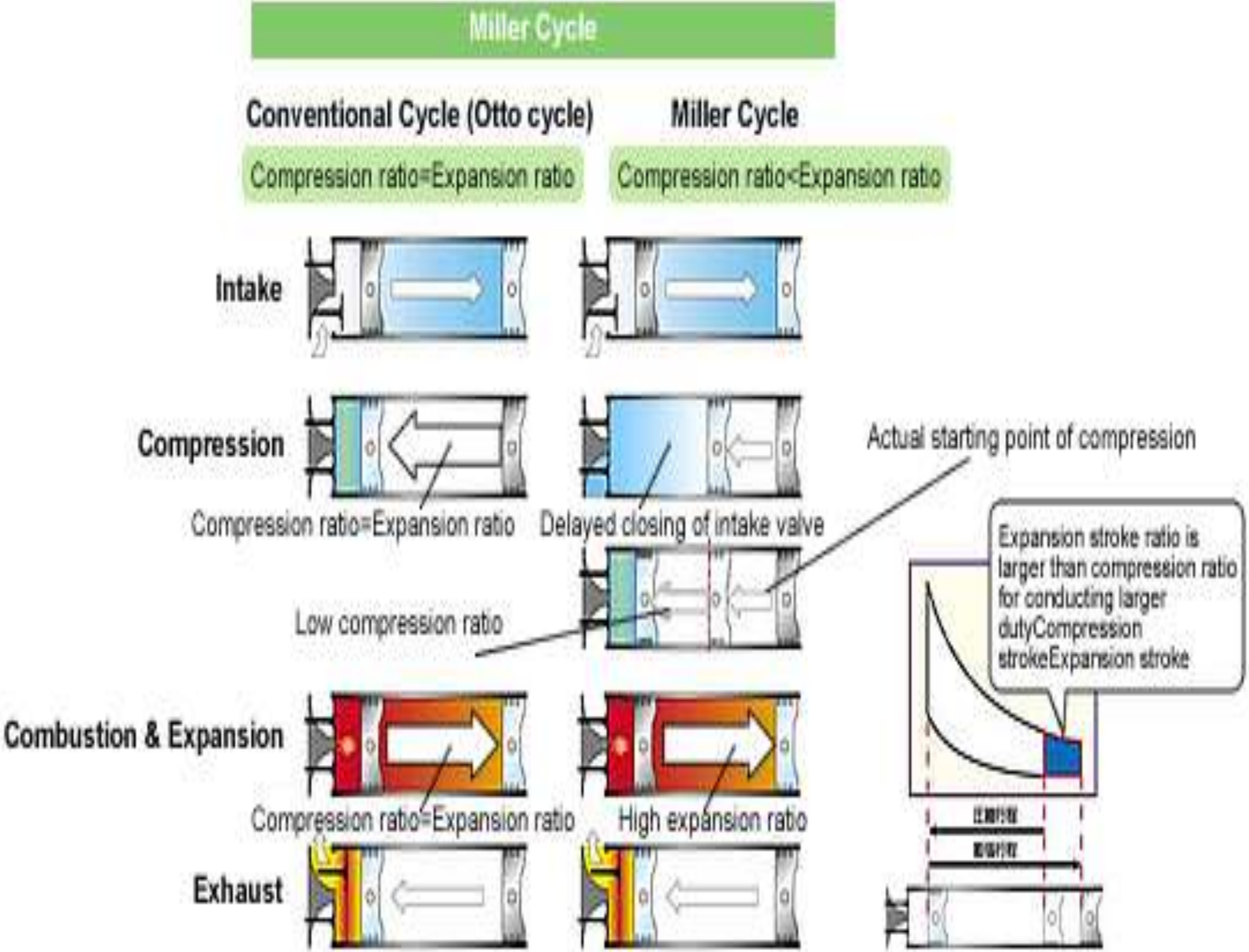
se the thermal efficiency is

$$\eta = 1 - (\lambda r_c)^{1-\gamma} - \frac{\lambda^{1-\gamma} - \lambda(1-\gamma) - \gamma}{(\gamma - 1)} \frac{P_1 V_1}{Q_{\text{in}}}$$

reduces to the Otto cycle thermal efficiency as $\lambda \to 1$. Th

$$\frac{\text{imep}}{P_1} = \frac{Q_{\text{in}}}{P_1 V_1}\left(\frac{r_c}{\lambda r_c - 1}\right)\eta$$


Compression
Stroke
Expansion
Stroke


Miller Cycle
Conventional Cycle (Otto cycle)
Miller Cycle
Compression ratio=Expansion ratio
Compression ratio<Expansion ratio
Intake
Compression
Compression ratio=Expansion ratio
Delayed closing of intake valve
Actual starting point of compression
Low compression ratio
Expansion stroke ratio is larger than compression ratio for conducting larger dutyCompression strokeExpansion stroke
Combustion & Expansion
Compression ratio=Expansion ratio
High expansion ratio
Exhaust

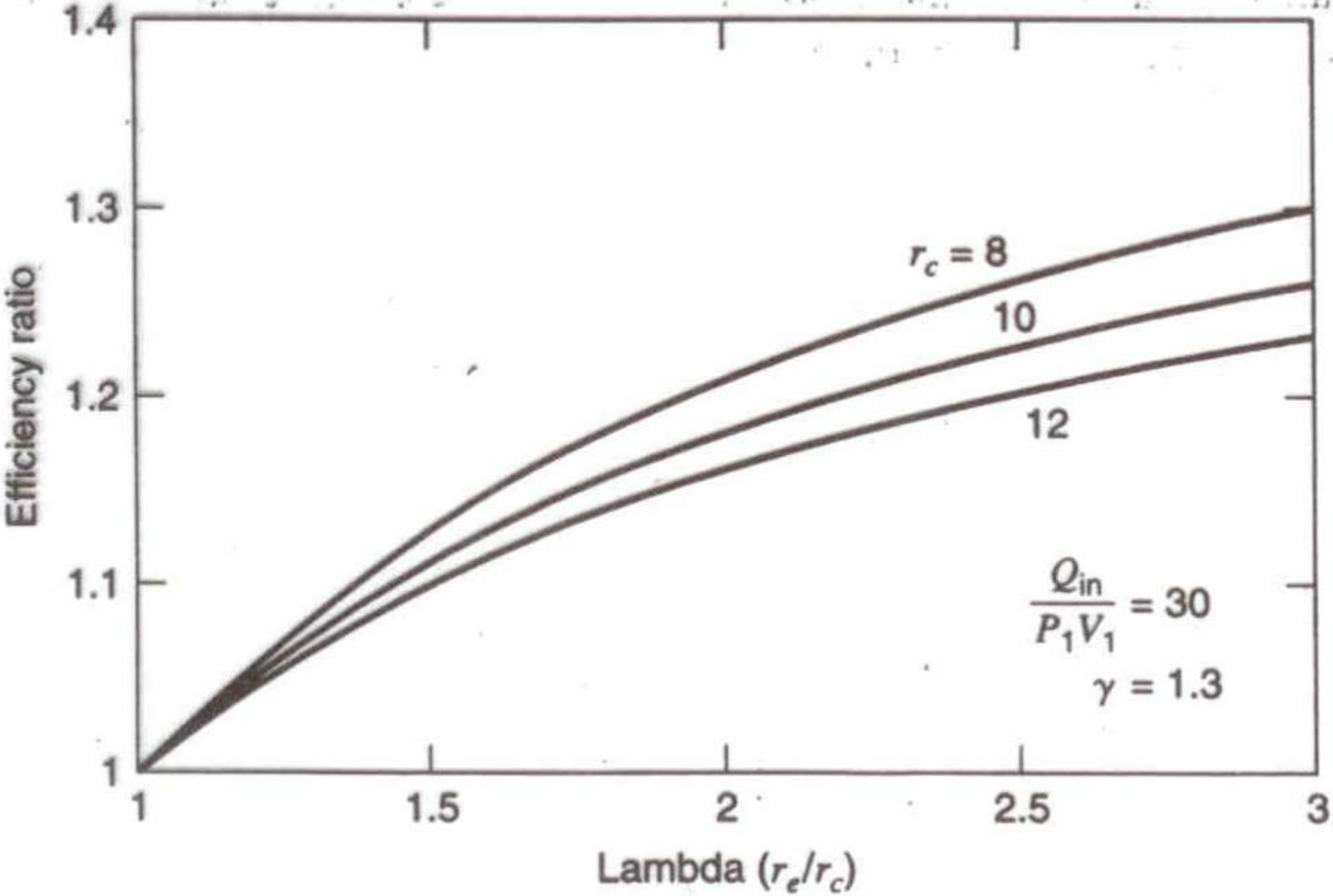


Figure 2-9 Miller/Otto cycle thermal efficiency ratio.

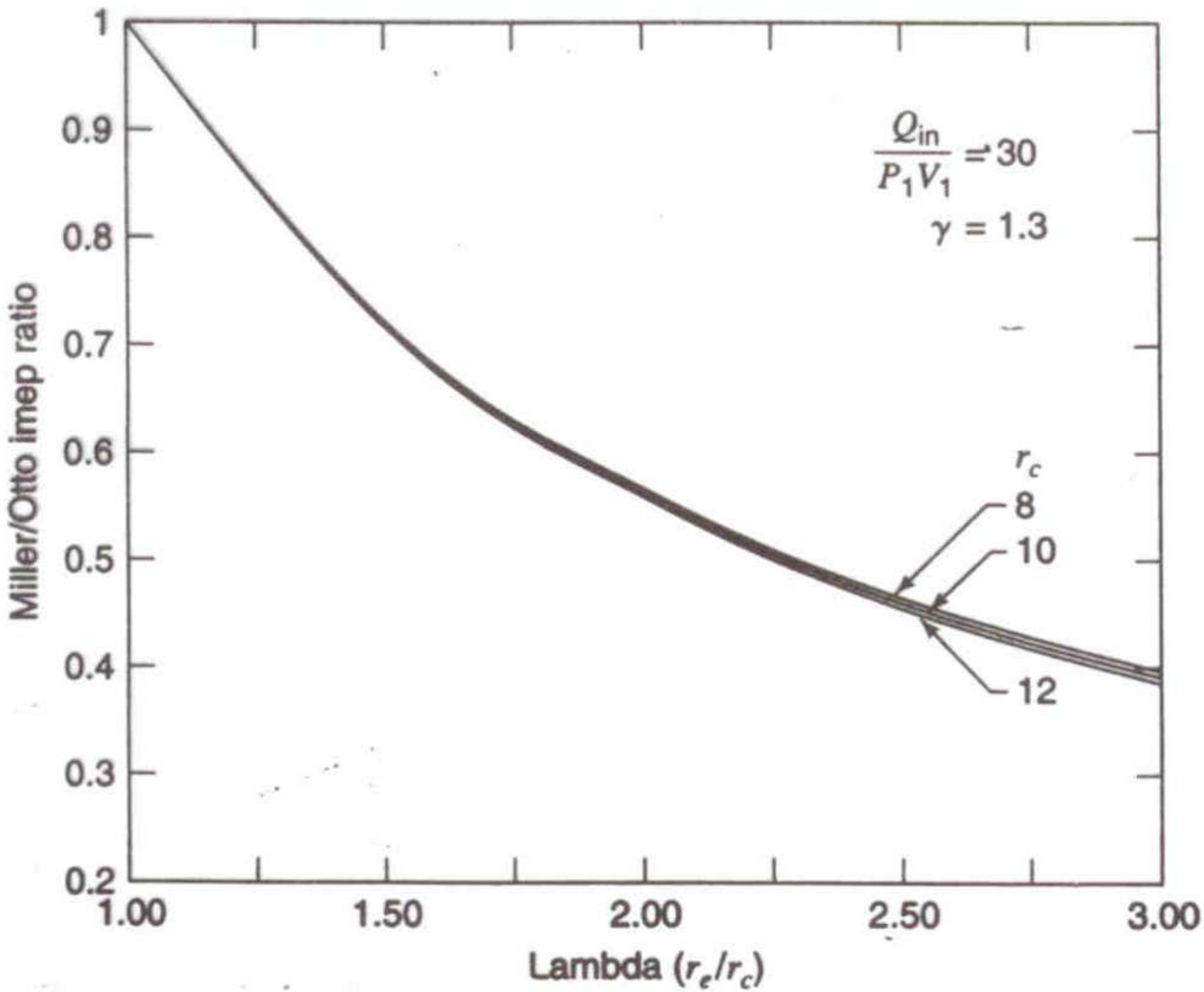
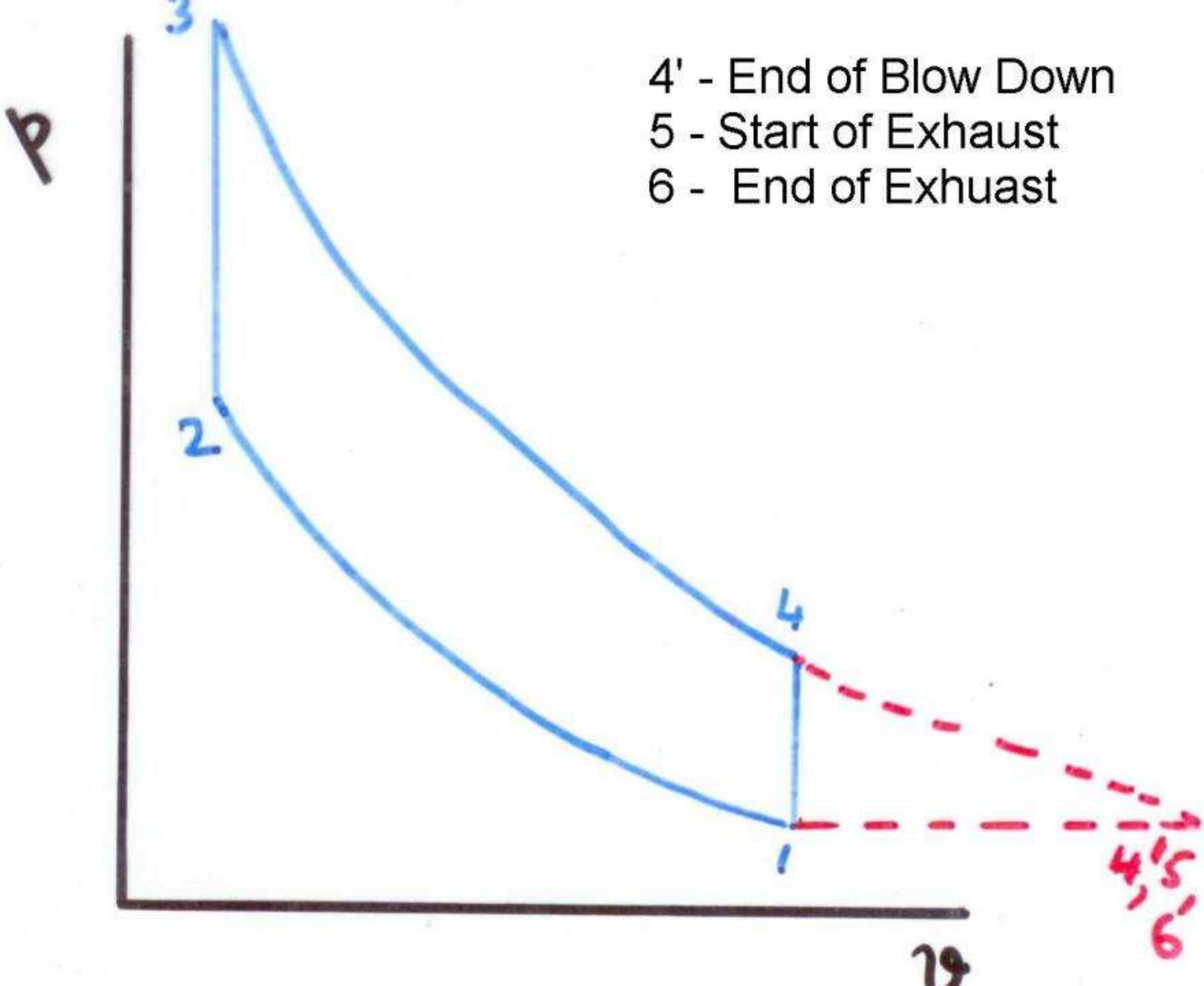


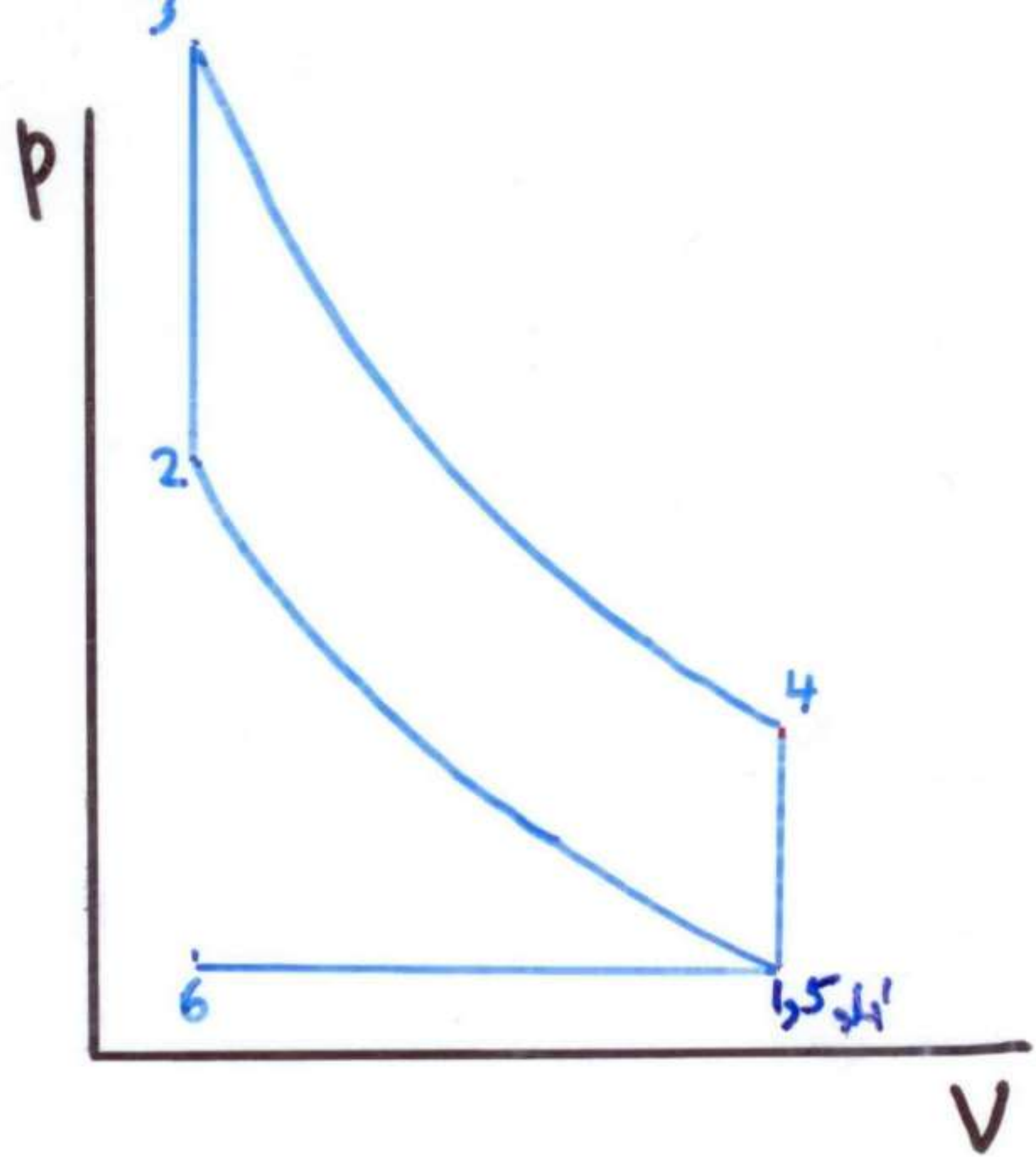
Figure 2-10 Miller cycle imep.

AIR STANDARD ENGINE

EXHAUST PROCESS

4' - End of Blow Down
5 - Start of Exhaust
6 - End of Exhuast




3
p
2
4
6
1,5,4'
V

Exhaust Process

Begins at Point 4

Pressure drops **Instantaneously** to atmospheric.

Process is called **Blow Down**

Ideal Process consists of 2 processes:

1. Release Process
2. Exhaust Process

Release Process

Piston is assumed to be stationary at end of Expansion stroke at bottom center

Charge is assumed to be divided into 2 parts

One part escapes from cylinder, undergoes free (irreversible) expansion when leaving

Other part remains in cylinder, undergoes reversible expansion

Both expand to atmospheric pressure

Release Process

State of the charge that remains in the cylinder is marked by path 4-4’, which in ideal case will be isentropic and extension of path 3-4.

Expansion of this charge will force the second portion from cylinder which will escape into the exhaust system.

Release Process

Consider the portion that escapes from cylinder:

Will expand into the exhaust pipe and acquire high velocity

Kinetic energy acquired by first element will be dissipated by fluid friction and turbulence into internal energy and flow work. Assuming no heat transfer, it will reheat the charge to final state 4”

Release Process

Succeeding elements will start to leave at states between 4 and 4', expand to atmospheric pressure and acquire velocity which will be progressively less. This will again be dissipated in friction.

End state will be along line 4'-4", with first element at 4" and last at 4'

Process 4-4" is an irreversible throttling process and temperature at point 4" will be higher than at 4' thus

$$v_{4''} > v_{4'}$$

Expansion of Cylinder Charge

The portion that remains is assumed to expand, in the ideal case, isentropically to atmospheric.

Such an ideal cycle drawn on the pressure versus specific volume diagram will resemble an Atkinson cycle or the **Complete Expansion Cycle**

COMPLETE EXPANSION

If V is the total volume and v the specific volume, then mass m is given by

$$m = \frac{V}{v}$$

And if m_1 is the TOTAL MASS OF CHARGE:

COMPLETE EXPANSION

$$m_1 = \frac{V_1}{v_1}$$

Let m_e be the RESIDUAL CHARGE MASS, then

$$m_e = \frac{V_6}{v_6}$$

COMPLETE EXPANSION

Let f be the residual gas fraction, given by

$$f = \frac{m_e}{m_1}$$

$$= \frac{V_6}{v_6} \Big/ \frac{V_1}{v_1}$$

$$= \frac{V_6}{V_1} x \frac{v_1}{v_6}$$

$$= \frac{V_2}{V_1} x \frac{v_1}{v_{4'}}$$

$$= \frac{1}{r}\left(\frac{v_1}{v_{4'}} \right)$$

Mass of charge remaining in cylinder after blow down but before start of exhaust stroke is:

$$m_5 = \frac{V_5}{v_5}$$

$$m_6 = \frac{V_6}{v_6}$$

$m_6 = m_e$ or mass of charge remaining in cylinder at end of exhaust stroke or residual gas

$$v_5 = v_6$$

$$V_5 \rangle V_6$$

$$\therefore m_5 \rangle m_6$$

Residual Gas Fraction

$f = (1/r)(v_1/v_{4'})$

$$f = \frac{1}{r}\left\{\frac{\left(\dfrac{p_e}{p_i}\right)^{\frac{1}{\gamma}}}{\left[1+\dfrac{Q'}{c_v T_1 r^{\gamma-1}}\right]^{\frac{1}{\gamma}}}\right\}$$

Temperature of residual gas T_6 can be obtained from the following relation:

$$\frac{T_6}{T_1} = \left(\frac{p_e}{p_i} \right)^{\frac{\gamma - 1}{\gamma}} \left(1 + \frac{Q'}{c_v T_1 r^{\gamma - 1}} \right)^{\frac{1}{\gamma}}$$

INTAKE PROCESS

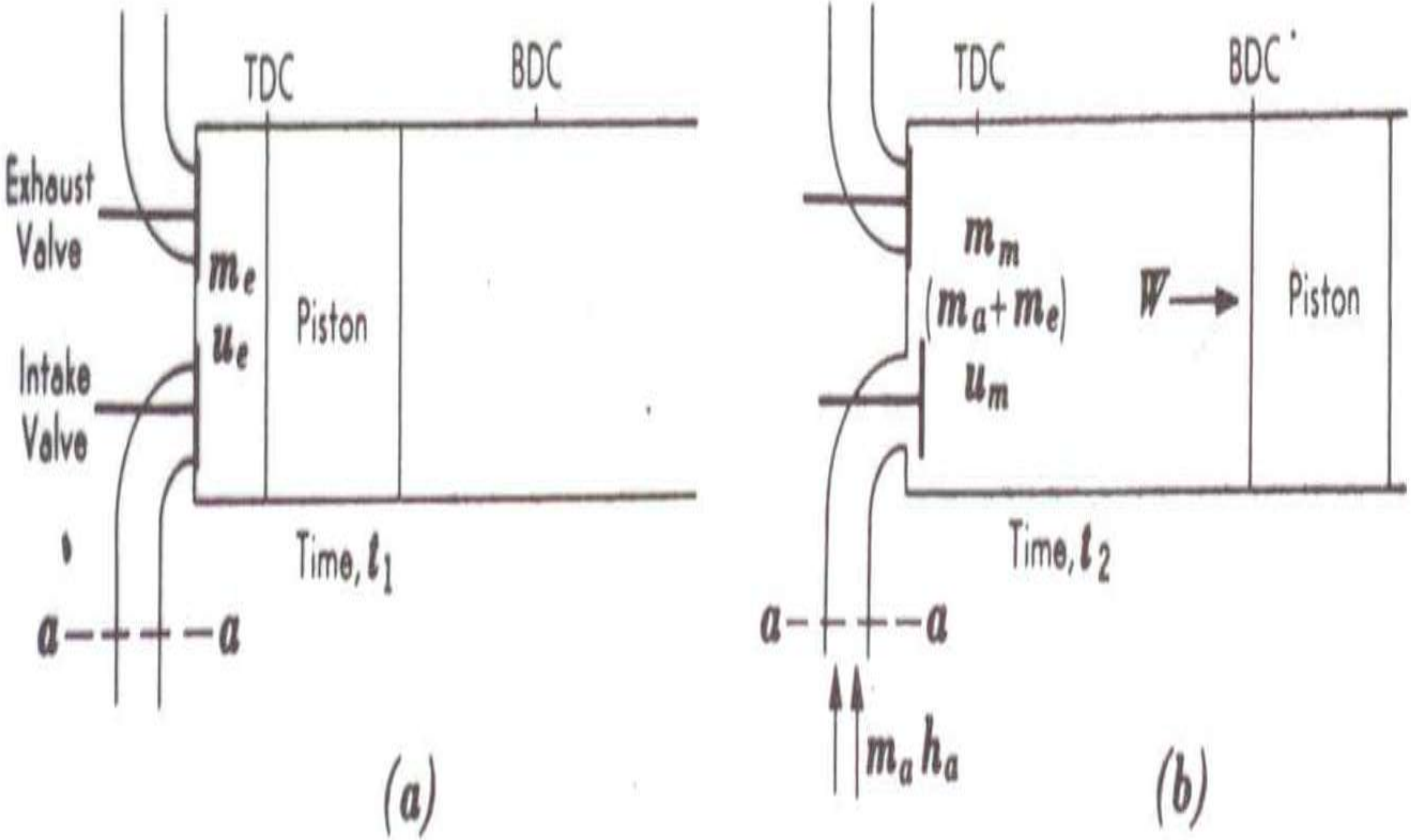


FIG. 6-13. Intake process; (a) system at time t, and (b) at time t_2.

Intake Process

- Intake process is assumed to commence when the inlet valve opens and piston is at TDC.
- Clearance volume is filled with hot burnt charge with mass m_e and internal energy u_e at time t_1.
- Fresh charge of mass m_a and enthalpy h_a enters and mixes with residual charge. Piston moved downwards to the BDC at time t_2.
- This is a **non-steady flow process**. It can be analyzed by applying the energy equation to the ***expanding system*** defined in the figure. Since

Intake Process

- $Q - W = [E_{\text{flow out}} - E_{\text{flow in}} + \Delta E_{\text{system}}]_{t1 \text{ to } t2}$ ….. (1)

- and, since the flow is inward, $E_{\text{flow out}}$ is zero. Process is assumed to be adiabatic therefore Q is zero. Thus

- $- W = - E_{\text{flow in}} + \Delta E_{\text{system}}$ …. (2)

Intake Process

- Assume flow is quasi-steady. Neglect kinetic energy. Energy crossing a-a and entering into the cylinder consists of internal energy u_a and the flow energy $p_a v_a$ so that
- $E_{flow\ in,\ t1\ to\ t2} = m_a (u_a + p_a v_a)$ …. (3)

Intake Process

- Change in energy of the system, ΔE_{system}, between times t_1 and t_2 is entirely a change in internal energy and since

$$m_1 = m_a + m_e \quad \ldots \quad (4)$$

$$\therefore \Delta E_{system} = m_1 u_1 - m_e u_e \quad \ldots \quad (5)$$

- The mass of the charge in the intake manifold can be ignored or made zero by proper choice of the boundary a-a. The work done by the air on the piston is given by

Intake Process

$$W = \int p dV$$

This is Eq. 6

Integrated from tdc to bdc

Intake Process

- This integration is carried out from TDC to BDC. Substituting from Eq. 3, 5 and 6 in Eq. 2 to give

$$-\int_{TDC}^{BDC} p\,dV = -m_a h_a + mu - m_r u_r$$

This is the basic equation of the Intake Process.

Intake Process

There are THREE cases of operation of an engine. These are as follows:

1. For the spark ignition engine operating at full throttle. This is also similar to the conventional (naturally aspirated) compression ignition engine. At this operating condition, exhaust pressure, p_{ex}, is equal to inlet pressure, p_{in}, that is

$$p_{ex}/p_{in} = 1$$

Intake Process

2. For the spark ignition engine operating at idle and part throttle. At this operating condition, exhaust pressure is greater than inlet pressure, that is

$$p_{ex}/p_{in} > 1$$

There are two possibilities in this case:

(i) Early inlet valve opening. Inlet valve opens before piston reaches TDC.

(ii) Late inlet valve opening. Inlet valve opens when piston reaches near or at TDC.

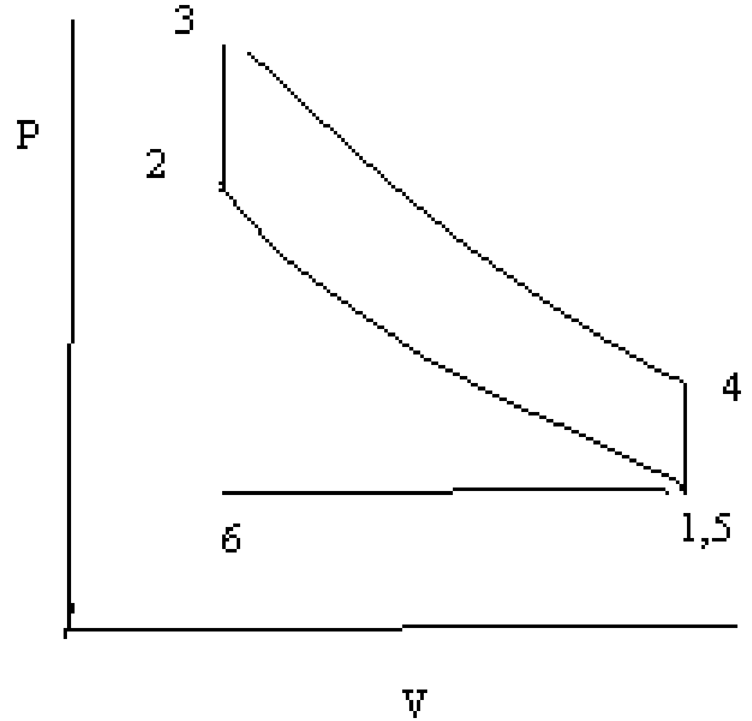
Intake Process

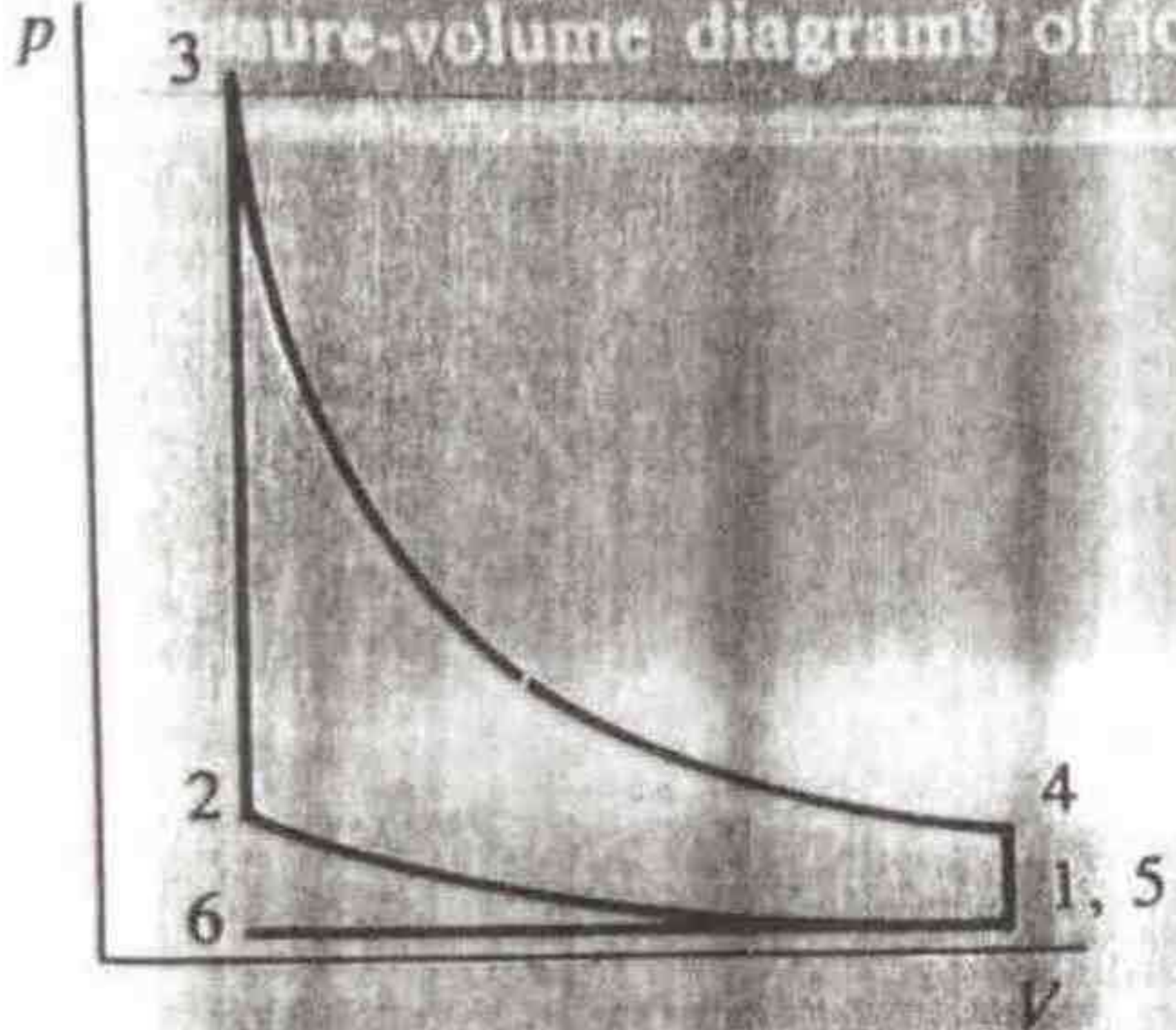
3. For the spark and compression ignition engine operating with a supercharger. At this operating condition, the inlet pressure is greater than the exhaust pressure, that is

$$p_{ex}/p_{in} < 1$$

Case 1: Wide Open Throttle SI or Conventional CI Engine.

- Fig.1





volume diagrams of
p
3
2
6
4
1, 5
V

WOT SI and Conventional CI

Since intake process is at manifold pressure (assumed constant) and equal to p_a

Thus $p_1 = p_a = p_6$ hence

$$W = \int_6^1 p\,dV = p_1\left(V_1 - V_6\right)$$

By definition, $m = V/v$ so that

$$W = m_1 p_1 v_1 - m_e p_6 v_6$$
$$= m_1 p_a v_a - m_e p_e v_e$$

WOT SI and Conventional CI

Substituting in the basic equation for the intake process, for W, and simplifying

$$m_1 h_m = m_a h_a + m_e h_e$$

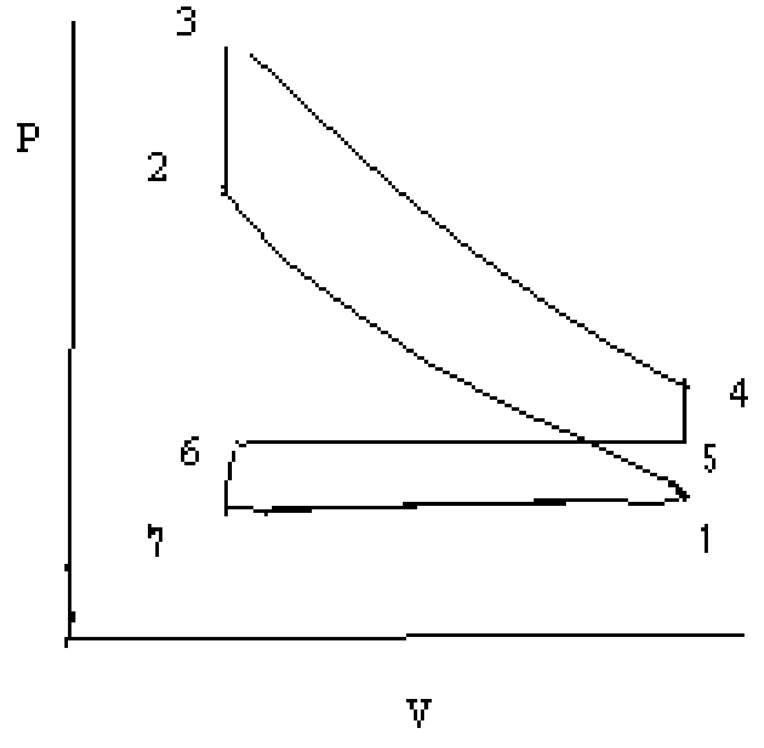
Dividing through by m_1 and remembering that the ratio m_e/m_1 is the residual gas fraction, f, we get

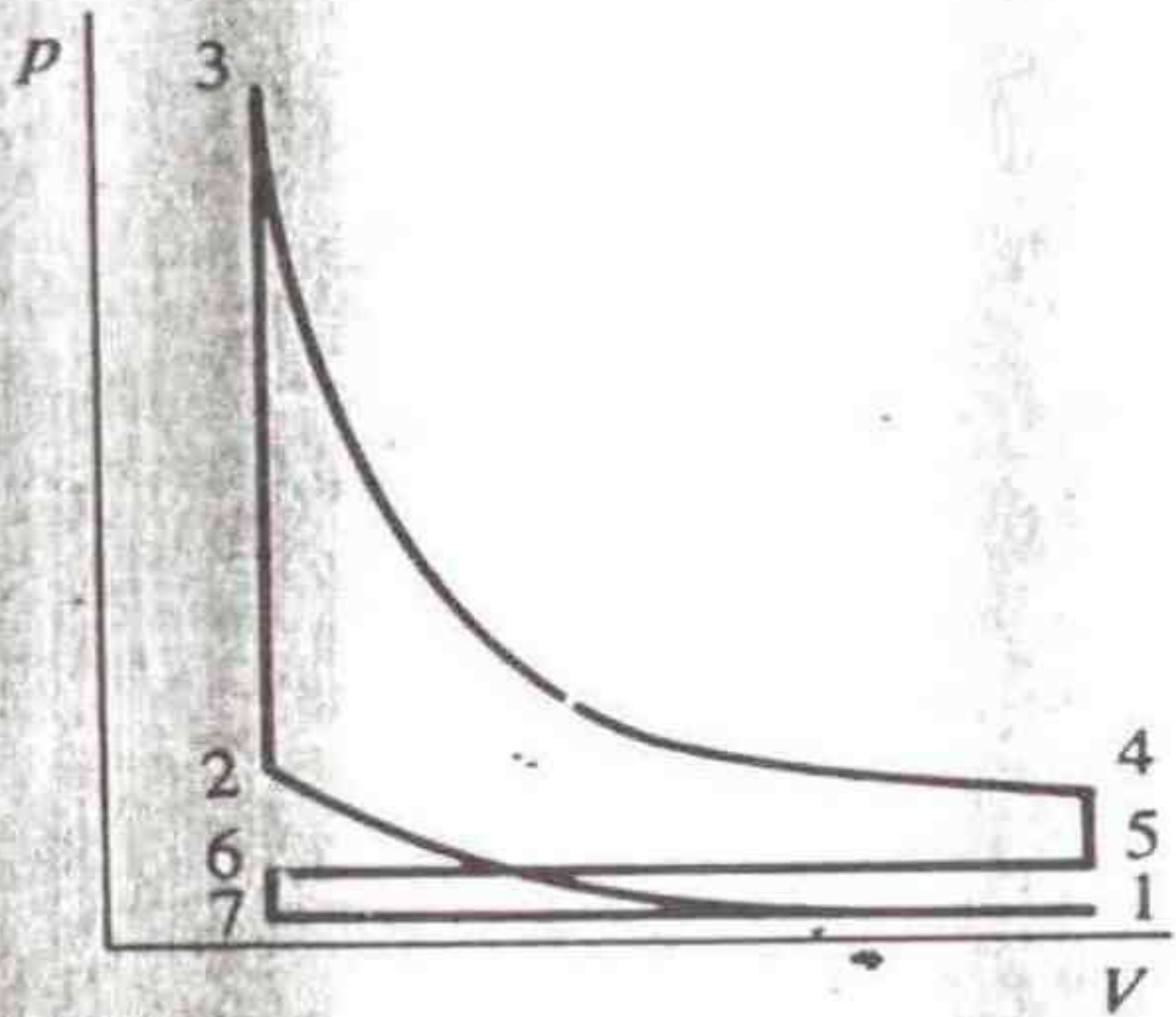
$$h_1 = (1 - f)\, h_a + f h_e$$

This gives the equation of the ideal intake process at wide open throttle for an Otto cycle engine and can be applied to the dual cycle engine as well.

Case 2(a): Part throttle SI engine. Early inlet valve opening.

Fig.2





p
3
2
6
7
4
5
1
V

Part Throttle: Early IVO

If the inlet valve opens before the piston reaches TDC, the residual charge will first expand into the intake manifold and mix with the fresh charge and then reenter the cylinder along with the fresh charge.

Now

$$W = \int_7^1 p\,dV = p_1 \left(V_1 - V_7 \right)$$

$$= p_1 v_1 m_1 – p_1 v_7 m_e$$

Part Throttle: Early IVO

Hence:

$$-(p_1v_1m_1 - p_1v_7m_e) = -m_ah_a + m_1u_1 - m_eu_e$$

Upon simplification, this becomes

$$m_1h_1 = m_ah_a + m_eu_7 + p_1v_7m_e$$

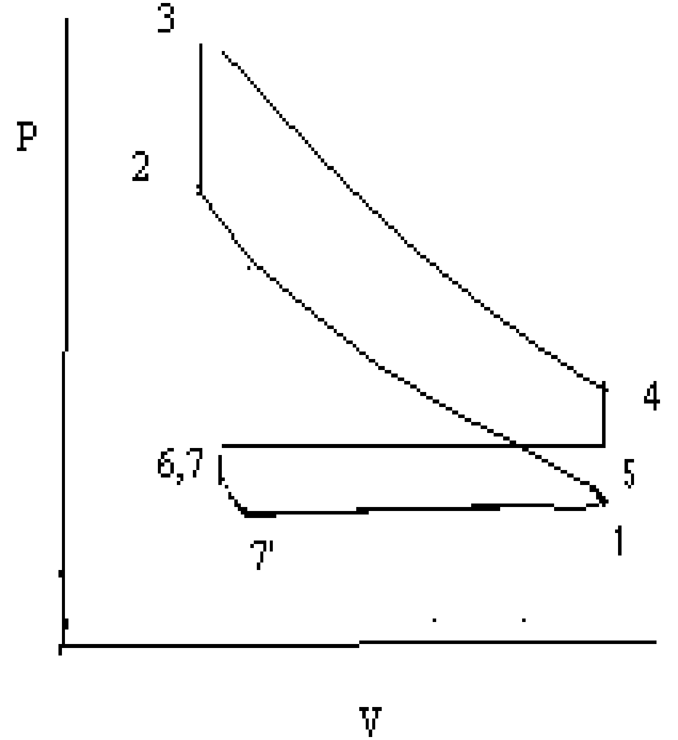
Thus we get

$$h_1 = (1- f)\, h_a + f\,(u_7 + p_7v_7)$$

$$= (1 - f)\, h_a + fh_7$$

Case 2(b): Part throttle SI engine. Late inlet valve opening.

Fig. 3



Part Throttle Late IVO

The residual at the end of the exhaust stroke is at point 6. In this case, the valve opens when the piston reaches the TDC. The piston starts on its intake stroke when the fresh charge begins to enter. However, since the fresh charge is at a lower pressure, mixing will not take place until pressure equalization occurs. Thus before the charge enters, the residual charge expands and does work on the piston in the expansion process, 7-7’. This process, in the ideal case, can be assumed to be isentropic. Once pressure equalization occurs, the mixture of the residual and fresh charge will press against the piston during the rest of the work process, 7’-1.

Part Throttle Late IVO

Now:

$$W = \int_{TDC}^{BDC} pdV = \int_{7}^{7'} pdV + \int_{7'}^{1} pdV$$

During the adiabatic expansion, the work done by the residuals is given by

$$-\Delta U = m_e(u_7 - u_{7'})$$

$$\text{Hence, } W = m_e(u_7 - u_{7'}) + p_1(V_1 - V_{7'})$$

Part Throttle Late IVO

And since $m = V/v$,

$$W = m_e(u_7 - u_{7'}) + m_1 p_1 v_1 - m_e p_{7'} v_{7'}$$

$$\text{Thus, } m_1 h_1 = m_a h_a + m_e h_{7'}$$

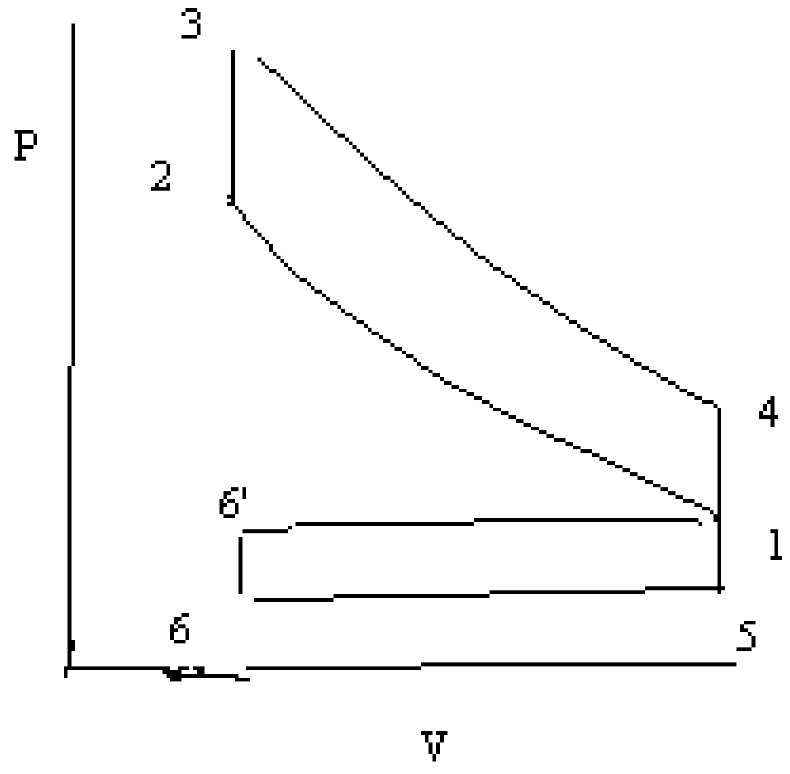
Which reduces to $h_m = (1 - f)\, h_a + f h_{7'}$

This gives the equation for the case where the inlet valve opens late, that is, after the piston reaches the top dead center of the exhaust stroke.

Although the throttle may drop the pressure radically, this has little effect on either the enthalpy of the liquid or the gases, being zero for gases behaving ideally.

Case 3: Supercharged Engine

Fig. 4



Supercharged Engine

Here, the intake pressure is higher than the exhaust pressure. Pressure $p_{6'}$ or p_1 represents the supercharged pressure and p_5 or p_6 the exhaust pressure. Intake starts from point 6'

As before

$$W = \int_{6'}^{1} p dV = p_1 \left(V_1 - V_{6'} \right)$$

$$= p_1 v_1 m_1 - p_1 v_{6'} m_e$$

Supercharged Engine

Hence

$$- (p_1 v_1 m_1 – p_1 v_{6'} m_e) = -m_a h_a + m_1 u_1 - m_e u_e$$

Upon simplification, this becomes

$$m_1 h_1 = m_a h_a + m_e u_{6'} + p_1 v_{6'} m_e$$

Thus we get

$$h_1 = (1- f)\, h_a + f\, (u_{6'} + p_{6'} v_{6'})$$

$$= (1 – f)\, h_a + f h_{6'}$$

Effect of Variation in Specific Heat of Air

A more realistic solution for Air Cycles

Effect of Variation in specific heat and gamma

Because of variation in specific heat and gamma, the cycle analysis will be different.

We cannot use the standard formulas for determining the air standard efficiencies.

We must determine the temperatures and pressures taking into account the variation in c_p and γ and determine the net work and heat supplied or heat supplied and heat rejected to determine the efficiencies.

Correlations for air

A number of correlations are available for determining the specific heat of air (at constant pressure or at constant volume) as function of temperature. Some correlations for gamma are also available.

For example:

$$\gamma = 1.4 - 7.18 \times 10^{-5} T$$

A few of the correlations are given below:

Correlations for air

$c_p = 0.9211 + 0.0002306\ T$ kJ/kg-K

T is in Kelvin

Other properties can be obtained.

A third order equation was proposed by Partha Pratim Saha, 89085,ex student of this course

$$c_p = 26.430213692 + 8.4435671*10^{-3}T - 2.1567692496*10^{-6}T^2 + 1.9461954*10^{-10}T^3 \text{ kJ /kmole K}$$

- T is in Kelvin
- Molecular weight of air is 29

More correlations for air

According to Lucas, the c_p of any gas is given as follows:

$c_p = a_{ij} (T/1000)^{i-1}$

where i = 1 to 7 and j represents the particular species, isooctane, oxygen or nitrogen. The units are kJ $kmole^{-1}$ K^{-1}

Values of a_{ij} are given elsewhere

Another correlation for different gases and air is given below:

$$c_p = C_0 + C_1 \frac{T}{1000} + C_2 \left(\frac{T}{1000} \right)^2 + C_3 \left(\frac{T}{1000} \right)^3 kJ/kgK$$

Gas	C_0	C_1	C_2	C_3
Air	1.05	-0.365	0.85	-0.39
Methane	1.2	3.25	0.75	-0.71
CO_2	0.45	1.67	-1.27	0.39
Steam	1.79	0.107	0.586	-0.20
O_2	0.88	-0.0001	0.54	-0.33
N_2	1.11	-0.48	0.96	-0.42

The next slide gives tabulated data of air as function of temperature.

T(K)	u (kJ/kg)	h (kJ/kg)	ρ (kg/m^3)	c_p (kJ/kg·K)	$\mu \times 10^5$ (kg/m·s)	D_{AB} (cm^2/s)	k (W/m·K)	Pr	Sc
300	214.32	300.43	1.177	1.005	1.853	0.21	0.0261	0.711	0.749
400	286.42	401.26	0.882	1.013	2.294	0.34	0.0330	0.703	0.788
500	359.79	503.30	0.706	1.029	2.682	0.50	0.0395	0.699	0.760
600	435.03	607.27	0.588	1.051	3.030	0.68	0.0456	0.698	0.751
700	512.58	713.50	0.505	1.075	3.349	0.89	0.0513	0.702	0.745
800	592.53	822.15	0.441	1.099	3.643	1.11	0.0569	0.703	0.744
900	674.77	933.10	0.392	1.121	3.918	1.35	0.0625	0.703	0.740
1000	759.14	1046.17	0.353	1.141	4.177	1.61	0.0672	0.709	0.735
1100	845.38	1161.12	0.321	1.160	4.44	1.88	0.0732	0.704	0.736
1200	933.28	1277.72	0.294	1.177	4.69	2.17	0.0782	0.704	0.735
1300	1022.67	1395.81	0.271	1.195	4.93	2.47	0.0837	0.704	0.736
1400	1113.34	1515.18	0.252	1.212	5.17	2.79	0.0891	0.703	0.736
1500	1205.15	1635.69	0.235	1.230	5.40	3.13	0.0946	0.702	0.734
1600	1297.93	1757.18	0.221	1.248	5.63	3.49	0.100	0.703	0.730
1700	1391.62	1879.57	0.208	1.266	5.85	3.86	0.105	0.703	0.728
1800	1486.13	2002.78	0.196	1.286	6.07	4.26	0.111	0.703	0.727
1900	1581.33	2126.71	0.186	1.307	6.29	4.67	0.117	0.703	0.724
2000	1677.21	2251.29	0.176	1.331	6.50	5.10	0.124	0.698	0.724
2100	1773.69	2376.45	0.168	1.359	6.72	5.53	0.131	0.696	0.723
2200	1870.73	2502.19	0.160	1.392	6.93	5.97	0.139	0.694	0.725
2300	1968.28	2628.45	0.153	1.434	7.14	6.42	0.149	0.687	0.727
2400	2066.30	2755.17	0.147	1.487	7.35	6.88	0.161	0.679	0.727
2500	2164.76	2882.33	0.141	1.556	7.57	7.36	0.175	0.673	0.729

Note: For other pressures multiply ρ by the pressure in atmospheres; divide D_{AB} by the pressure in atmospheres; c_p, μ, k and Pr do not change with pressure. D_{AB} is the binary diffusion coefficient for O_2 into N_2 and Sc is based on this D_{AB}.

Source: Data from Keenan, Chao, and Kaye, *Gas Tables*, Wiley, New York, 1983; Kays and Crawford, *Convective Heat and Mass Transfer*, McGraw-Hill, New York, 1980, Table A-1; and Field, M. A., *Combustion of Pulverized Coal*, Cheney & Sons, London, 1967, App. Q.

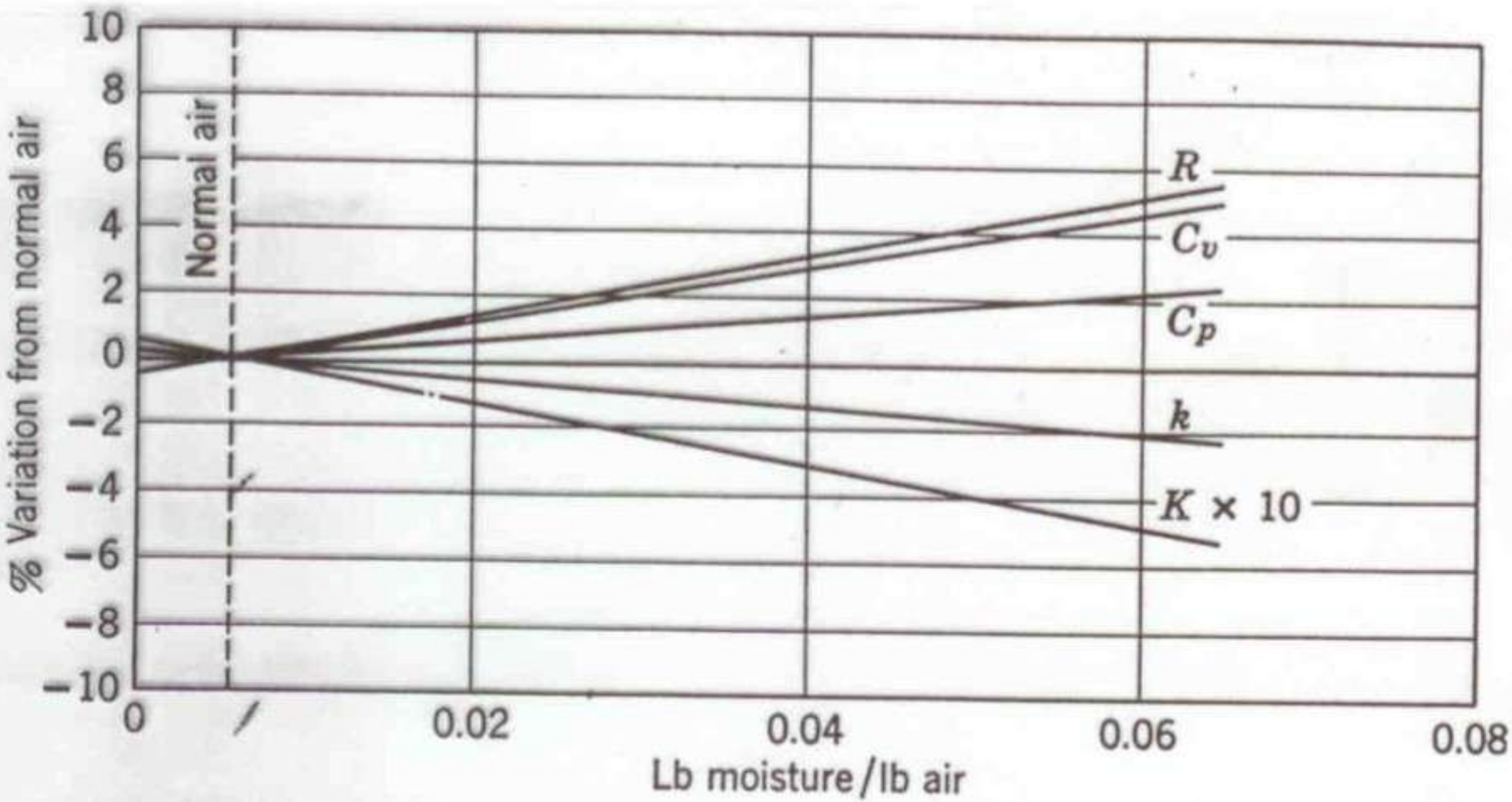


Fig 3-2. Effect of humidity on properties of air: R = universal R/m in eq 2-2; C_v = specific heat at constant volume; C_p = specific heat at constant pressure; k = ratio of specific heats; K = thermal conductivity.

$$\gamma(T) = \frac{c_p(T)}{c_p(T) - R}$$

$$R \ \ for \ air \ is \ 0.287 \, kJ / kgK$$

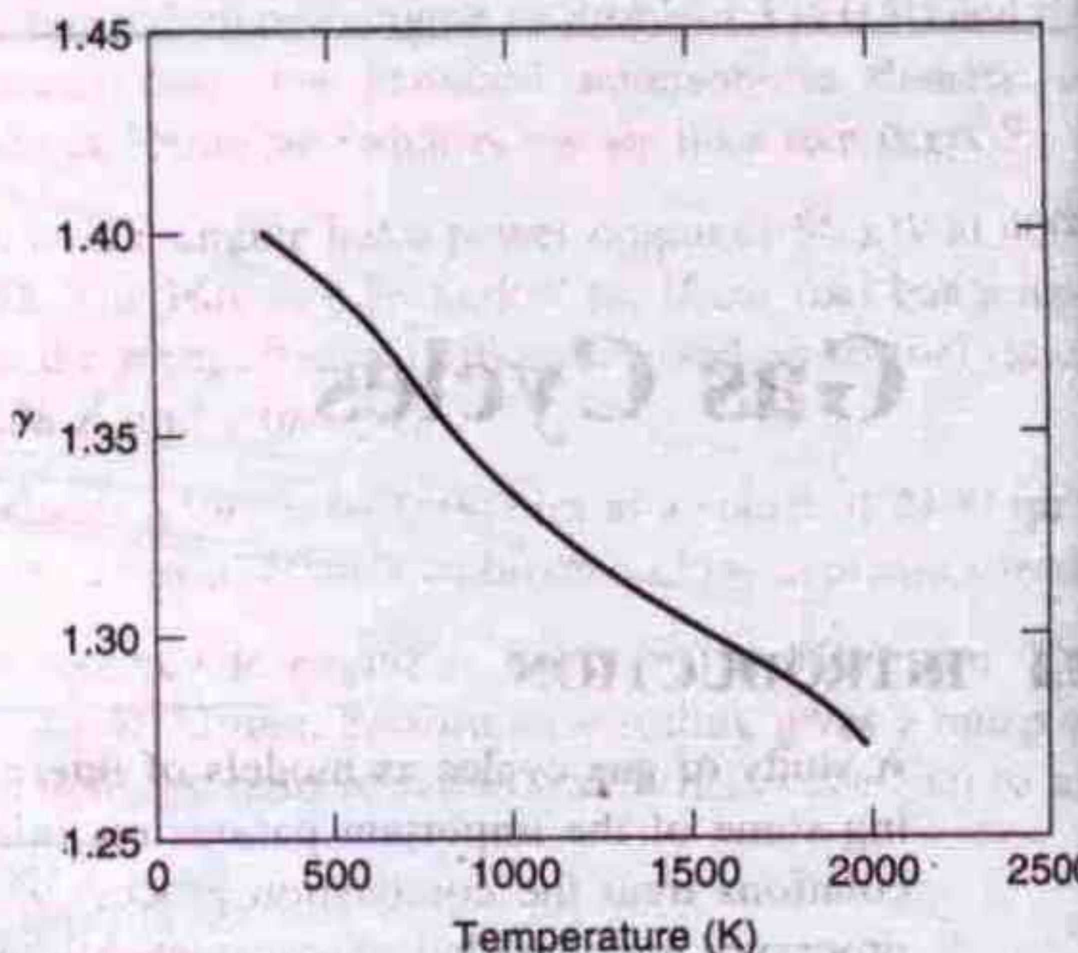


Figure 2-1 Specific heat ratio for air.

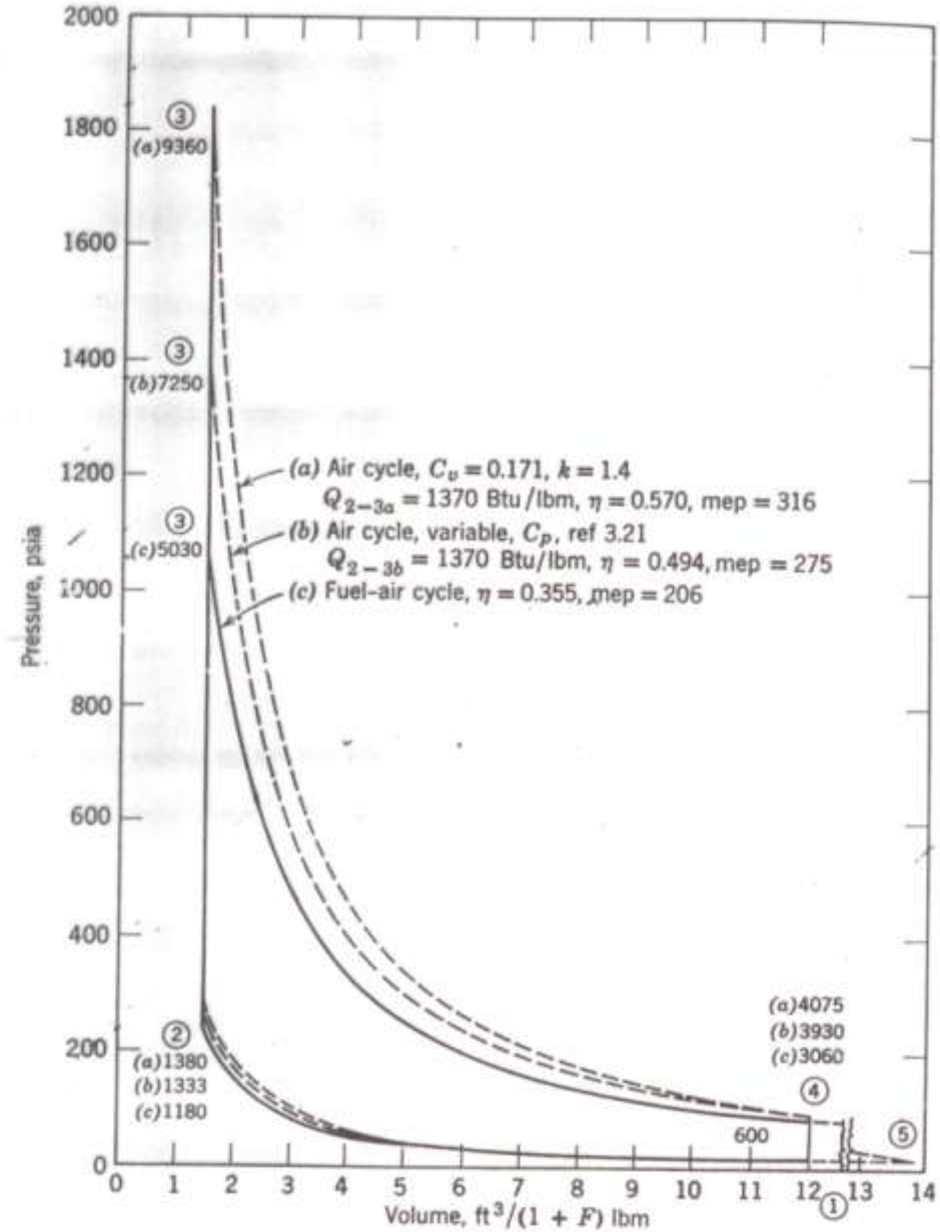


Fig 4-1. Constant-volume fuel-air cycle compared with air cycle: $r = 8$; $F_R = 1.2$; $p_1 = 14.7$; $T_1 = 600$; $f = 0.05$. Fuel is octene, C_8H_{16}. Numbers at each station are temperatures in °R: (a) air cycle; (b) air cycle, variable sp heat; (c) fuel-air cycle (example 4-1).

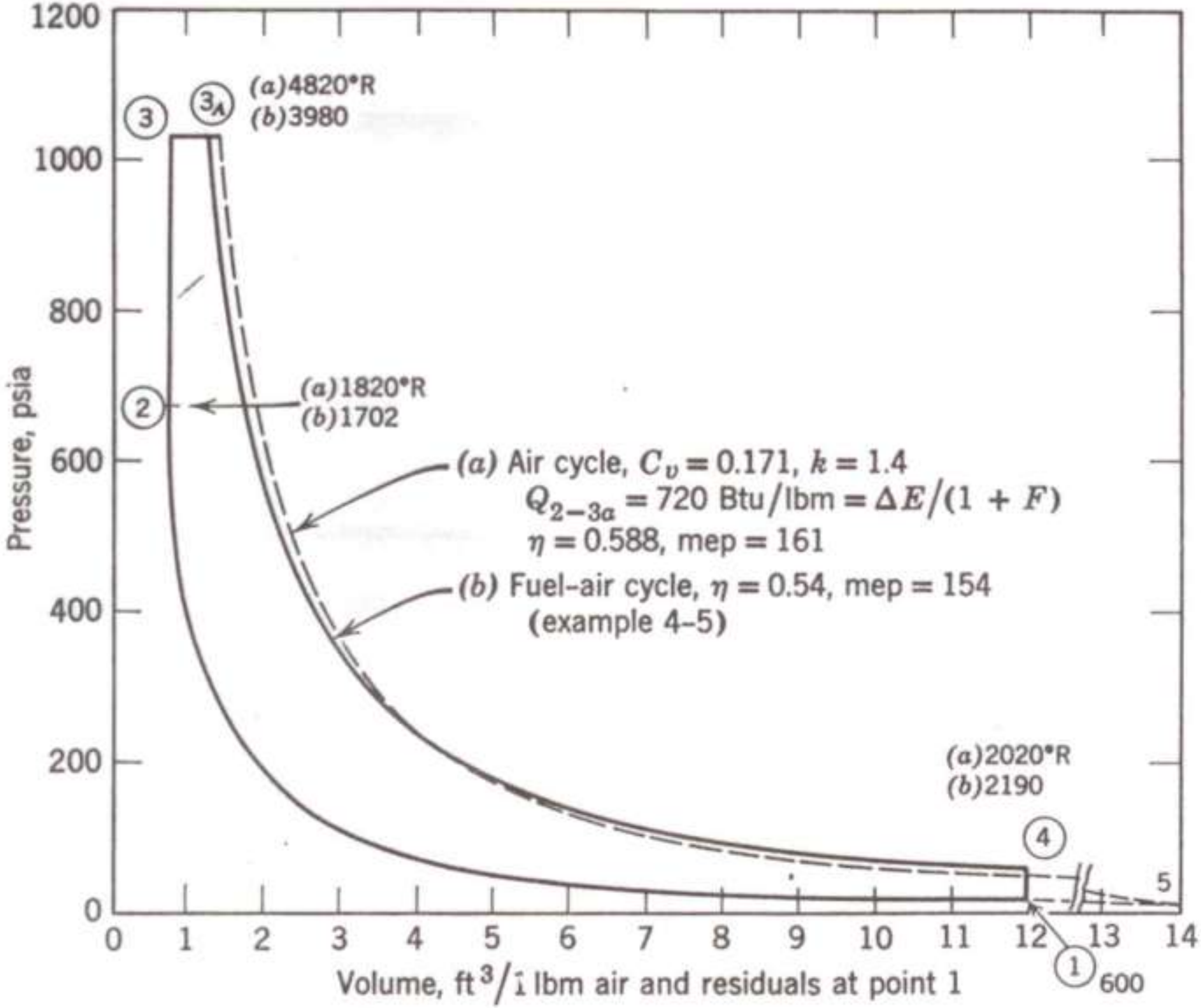


Fig 4-2. Limited-pressure fuel-air cycle compared with air cycle: $r = 16$; $F_R = 0.6$; $p_1 = 14.7$; $T_1 = 600$; $f = 0.03$; $p_3 = 70 \times p_1 = 1030$ psia. Numbers at eac . station are temperatures in °R.