SATHYABAMA
INSTITUTE OF SCIENCE AND TECHNOLOGY
(DEEMED TO BE UNIVERSITY)
Accredited "A" Grade by NAAC | 12B Status by UGC | Approved by AICTE
www.sathyabama.ac.in

SCHOOL OF MECHANICAL ENGINEERING

DEPARTMENT OF AERONAUTICAL

UNIT – III – Aero Engineering Thermodynamics – SAEA1301

III. AIR, VAPOUR & REFRIGERATION CYCLES

Definition of a cycle

A cycle is defined as a repeated series of operations occurring in a certain order. It may be repeated by repeating the processes in the same order. The cycle may be of imaginary perfect engine or actual engine. The former is called ideal cycle and the latter actual cycle. In ideal cycle all accidental heat losses are prevented and the working substance is assumed to behave like a perfect working substance.

Air standard efficiency

To compare the effects of different cycles, it is of paramount importance that the effect of the calorific value of the fuel is altogether eliminated and this can be achieved by considering air (which is assumed to behave as a perfect gas) as the working substance in the engine cylinder. The efficiency of engine using air as the working medium is known as an "Air standard efficiency". This efficiency is often called ideal efficiency. The actual efficiency of a cycle is always less than the air-standard efficiency of that cycle under ideal conditions. This is taken into account by introducing a new term "Relative efficiency" which is defined as the ratio of Actual thermal efficiency to Air standard efficiency.

The analysis of all air standard cycles is based upon the following assumptions:

1. The gas in the engine cylinder is a perfect gas i.e., it obeys the gas laws and has constant specific heats.
2. The physical constants of the gas in the cylinder are the same as those of air at moderate temperatures i.e., the molecular weight of cylinder gas is 29.Cp = 1.005 kJ/kg-K, Cv = 0.718 kJ/kg-K.
3. The compression and expansion processes are adiabatic and they take place without internal friction, i.e., these processes are isentropic.
4. No chemical reaction takes place in the cylinder. Heat is supplied or rejected by bringing a hot body or a cold body in contact with cylinder at appropriate points during the process.
5. The cycle is considered closed with the same 'air' always remaining in the cylinder to repeat the cycle.

Constant volume or otto cycle

This cycle is so named as it was conceived by 'Otto'. On this cycle, petrol, gas and many types of oil engines work. It is the standard of comparison for internal combustion engines.

Figs. 1 (a) and (b) shows the theoretical p-V diagram and T-s diagrams of this cycle respectively.

- The point 1 represents that cylinder is full of air with volume V_1, pressure P_1 and absolute temperature T_1.
- Line 1-2 represents the adiabatic compression of air due to which P_1, V_1 and T_1 change to P_2, V_2 and T_2 respectively.
- Line 2-3 shows the supply of heat to the air at constant volume so that P_2 and T_2 change to P_3 and T_3 (V_3 being the same as V_2).
- Line 3-4 represents the adiabatic expansion of the air. During expansion P_3, V_3 and T_3 change to a final value of P_4, V_4 or V_1 and T_4, respectively.
- Line 4-1 shows the rejection of heat by air at constant volume till original state (point 1) reaches. Consider 1 kg of air (working substance):

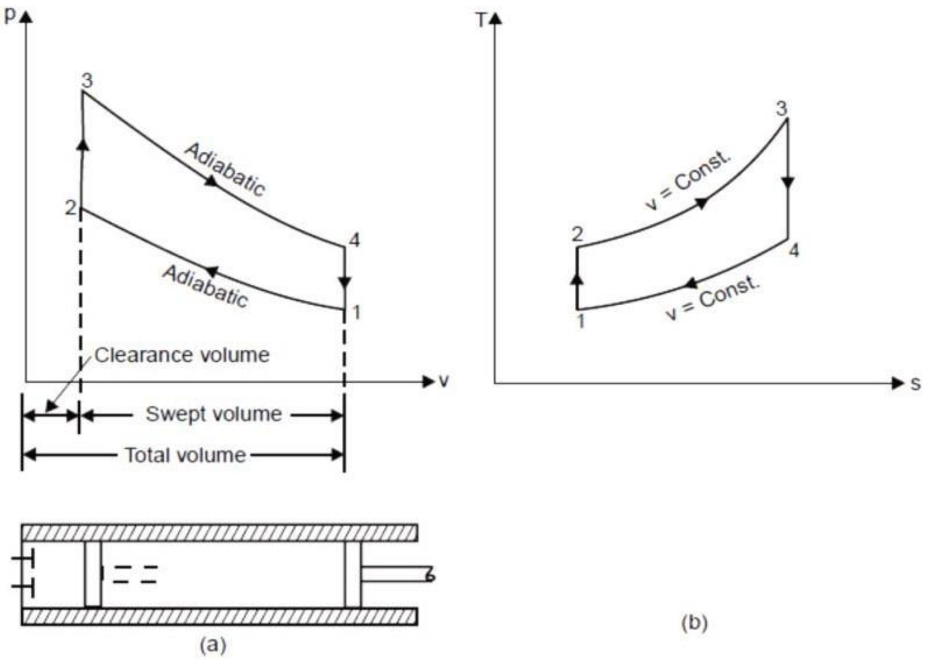


Fig. 1. Constant volume or otto cycle

Heat supplied at constant volume $= c_v(T_3 - T_2)$.

Heat rejected at constant volume $= c_v(T_4 - T_1)$.

But, work done $=$ Heat supplied $-$ Heat rejected

$$= c_v(T_3 - T_2) - c_v(T_4 - T_1)$$

$$\therefore \quad \text{Efficiency} = \frac{\text{Work done}}{\text{Heat supplied}} = \frac{c_v(T_3 - T_2) - c_v(T_4 - T_1)}{c_v(T_3 - T_2)}$$

$$= 1 - \frac{T_4 - T_1}{T_3 - T_2}$$

Let compression ratio, $r_c \, (= r) = \frac{v_1}{v_2}$

and expansion ratio, $r_e \, (= r) = \frac{v_4}{v_3}$

(These *two ratios are same* in this cycle)

As $\frac{T_2}{T_1} = \left(\frac{v_1}{v_2}\right)^{\gamma-1}$

Then, $T_2 = T_1 \cdot (r)^{\gamma-1}$

Similarly, $\frac{T_3}{T_4} = \left(\frac{v_4}{v_3}\right)^{\gamma-1}$

or $T_3 = T_4 \cdot (r)^{\gamma-1}$

Inserting the values of T_2 and T_3 in equation (i), we get

$$\eta_{otto} = 1 - \frac{T_4 - T_1}{T_4 \cdot (r)^{\gamma-1} - T_1 \cdot (r)^{\gamma-1}} = 1 - \frac{T_4 - T_1}{r^{\gamma-1}(T_4 - T_1)}$$

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

This expression is known as the air standard efficiency of the Otto cycle. It is clear from the above expression that efficiency increases with the increase in the value of r, which means we can have maximum efficiency by increasing r to a considerable extent, but due to practical difficulties its value is limited to about 8. The net work done per kg in the Otto cycle can also be expressed in terms of p, v. If p is expressed in bar i.e., 10^5 N/m^2, then work done

$$W = \left(\frac{p_3v_3 - p_4v_4}{\gamma - 1} - \frac{p_2v_2 - p_1v_1}{\gamma - 1}\right) \times 10^2 \text{ kJ}$$

$$\frac{p_3}{p_4} = r^{\gamma} = \frac{p_2}{p_1}$$

$$\frac{p_3}{p_2} = \frac{p_4}{p_1} = r_p$$

where r_p stands for *pressure ratio*.

and
$$v_1 = rv_2 = v_4 = rv_3 \qquad \left[\because \frac{v_1}{v_2} = \frac{v_4}{v_3} = r\right]$$

$$\therefore \qquad W = \frac{1}{\gamma - 1}\left[p_4 v_4 \left(\frac{p_3 v_3}{p_4 v_4} - 1\right) - p_1 v_1 \left(\frac{p_2 v_2}{p_1 v_1} - 1\right)\right]$$

$$= \frac{1}{\gamma - 1}\left[p_4 v_4 \left(\frac{p_3}{p_4 r} - 1\right) - p_1 v_1 \left(\frac{p_2}{p_1 r} - 1\right)\right]$$

$$= \frac{v_1}{\gamma - 1}\left[p_4 (r^{\gamma-1} - 1) - p_1 (r^{\gamma-1} - 1)\right]$$

$$= \frac{v_1}{\gamma - 1}\left[(r^{\gamma-1} - 1)(p_4 - p_1)\right]$$

$$= \frac{p_1 v_1}{\gamma - 1}\left[(r^{\gamma-1} - 1)(r_p - 1)\right]$$

Mean effective pressure (p_m) is given by :

$$p_m = \left[\left(\frac{p_3 v_3 - p_4 v_4}{\gamma - 1} - \frac{p_2 v_2 - p_1 v_1}{\gamma - 1}\right) \div (v_1 - v_2)\right] \text{ bar}$$

$$p_m = \frac{\left[\frac{p_1 v_1}{\gamma - 1}(r^{\gamma-1} - 1)(r_p - 1)\right]}{(v_1 - v_2)}$$

$$= \frac{\frac{p_1 v_1}{\gamma - 1}[(r^{\gamma-1} - 1)(r_p - 1)}{v_1 - \frac{v_1}{r}}$$

$$= \frac{\frac{p_1 v_1}{\gamma - 1}[(r^{\gamma-1} - 1)(r_p - 1)]}{v_1\left(\frac{r - 1}{r}\right)}$$

$$\mathbf{p_m} = \frac{p_1 r[(r^{\gamma-1} - 1)(r_p - 1)]}{(\gamma - 1)(r - 1)}$$

MEP may be thought of as the average pressure acting on a piston during different portions of its cycle.It is the ratio of the work done to stoke volume of the cycle

CONSTANT PRESSURE OR DIESEL CYCLE

This cycle was introduced by Dr. R. Diesel in 1897. It differs from Otto cycle in that heat is supplied at constant pressure instead of at constant volume. Fig.2 shows the p-v and T-s diagrams of this cycle respectively.

This cycle comprises of the following operations:

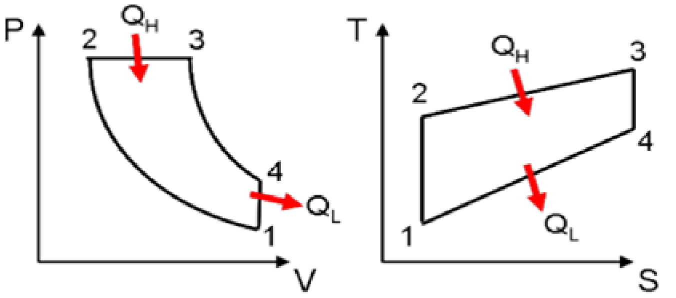


Fig. 2 P-V and T-S diagrams of Ideal Diesel Cycle

(i) 1-2....Adiabatic compression.

(ii) 2-3....Addition of heat at constant pressure.

(iii) 3-4....Adiabatic expansion.

(iv) 4-1... Rejection of heat at constant volume.

Point 1 represents that the cylinder is full of air. Let P_1, V_1 and T_1 be the corresponding pressure, volume and absolute temperature. The piston then compresses the air adiabatically (i.e., pV^r = constant) till the values become P_2, V_2 and T_2 respectively (at the end of the stroke) at point 2. Heat is then added from a hot body at a constant pressure. During this addition of heat let volume increases from V_2 to V_3 and temperature T_2 to T_3, corresponding to point 3. This point (3) is called the point of cut-off. The air then expands adiabatically to the conditions P_4, V_4 and T_4 respectively corresponding to point 4. Finally, the air rejects the heat to the cold body at constant volume till the point 1 where it returns to its original state

Consider 1 kg of air.

Heat supplied at constant pressure $= c_p(T_3 - T_2)$

Heat rejected at constant volume $= c_v(T_4 - T_1)$

Work done $=$ Heat supplied – heat rejected

$$= c_p(T_3 - T_2) - c_v(T_4 - T_1)$$

$$\therefore \qquad \eta_{\text{diesel}} = \frac{\text{Work done}}{\text{Heat supplied}}$$

$$= \frac{c_p(T_3 - T_2) - c_v(T_4 - T_1)}{c_p(T_3 - T_2)}$$

$$= 1 - \frac{(T_4 - T_1)}{\gamma(T_3 - T_2)} \qquad \ldots(i) \quad \left[\because \frac{c_p}{c_v} = \gamma\right]$$

Let compression ratio, $r = \frac{v_1}{v_2}$, and cut-off ratio, $\rho = \frac{v_3}{v_2}$ *i.e.,* $\frac{\text{Volume at cut-off}}{\text{Clearance volume}}$

Now, during *adiabatic compression 1-2,*

$$\frac{T_2}{T_1} = \left(\frac{v_1}{v_2}\right)^{\gamma - 1} = (r)^{\gamma - 1} \quad \text{or} \quad T_2 = T_1 \,.\, (r)^{\gamma - 1}$$

During *constant pressure process 2-3,*

$$\frac{T_3}{T_2} = \frac{v_3}{v_2} = \rho \quad \text{or} \quad T_3 = \rho \,.\, T_2 = \rho \,.\, T_1 \,.\, (r)^{\gamma - 1}$$

During *adiabatic expansion 3-4*

$$\frac{T_3}{T_4} = \left(\frac{v_4}{v_3}\right)^{\gamma - 1}$$

$$= \left(\frac{r}{\rho}\right)^{\gamma - 1} \qquad \left(\because \frac{v_4}{v_3} = \frac{v_1}{v_3} = \frac{v_1}{v_2} \times \frac{v_2}{v_3} = \frac{r}{\rho}\right)$$

$$\therefore \qquad T_4 = \frac{T_3}{\left(\frac{r}{\rho}\right)^{\gamma - 1}} = \frac{\rho \,.\, T_1 (r)^{\gamma - 1}}{\left(\frac{r}{\rho}\right)^{\gamma - 1}} = T_1 \,.\, \rho^{\gamma}$$

By inserting values of T_2, T_3 and T_4 in eqn. (*i*), we get

$$\eta_{\text{diesel}} = 1 - \frac{(T_1 \,.\, \rho^{\gamma} - T_1)}{\gamma\,(\rho \,.\, T_1 \,.\, (r)^{\gamma - 1} - T_1 \,.\, (r)^{\gamma - 1})} = 1 - \frac{(\rho^{\gamma} - 1)}{\gamma (r)^{\gamma - 1} (\rho - 1)}$$

It may be observed that eqn. for efficiency of diesel cycle is different from that of the Otto cycle only in bracketed factor. This factor is always greater than unity, because r > 1. Hence for a given compression ratio, the Otto cycle is more efficient.

Mean effective pressure p_m is given by :

$$p_m = \frac{p_1 v_1 r^{\gamma - 1} [\gamma(\rho - 1) - r^{1-\gamma} (\rho^\gamma - 1)]}{(\gamma - 1) v_1 \left(\frac{r-1}{r}\right)}$$

$$\mathbf{p_m} = \frac{p_1 r^\gamma [\gamma(\rho - 1) - r^{1-\gamma} (\rho^\gamma - 1)]}{(\gamma - 1)(r - 1)} \ .$$

DUAL COMBUSTION CYCLE

This cycle (also called the limited pressure cycle or mixed cycle) is a combination of Otto and Diesel cycles, in a way, that heat is added partly at constant volume and partly at constant pressure ; the advantage of which is that more time is available to fuel (which is injected into the engine cylinder before the end of compression stroke) for combustion. Because of lagging characteristics of fuel this cycle is invariably used for diesel and hot spot ignition engines.

The dual combustion cycle (Fig 3) consists of the following operations :

(i) 1-2—Adiabatic compression

(ii) 2-3—Addition of heat at constant volume

(iii) 3-4—Addition of heat at constant pressure

(iv) 4-5—Adiabatic expansion

(v) 5-1—Rejection of heat at constant volume.

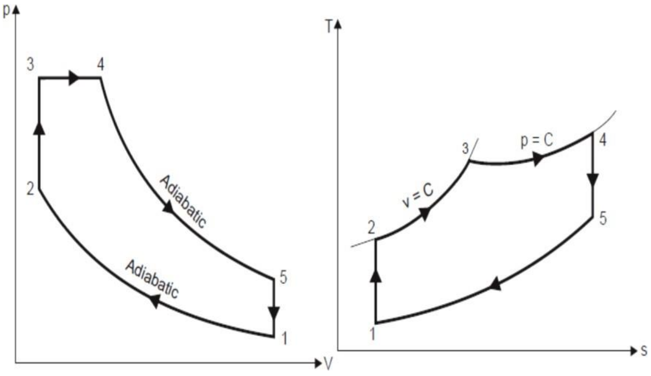


Fig.3. DUAL COMBUSTION CYCLE

Consider 1 kg of air.

Total heat supplied = Heat supplied during the operation 2-3 + heat supplied during the operation 3-4

$$= c_v(T_3 - T_2) + c_p(T_4 - T_3)$$

Heat rejected during operation 5-1 $= c_v(T_5 - T_1)$

Work done = Heat supplied − heat rejected

$$= c_v(T_3 - T_2) + c_p(T_4 - T_3) - c_v(T_5 - T_1)$$

$$\eta_{\text{dual}} = \frac{\text{Work done}}{\text{Heat supplied}} = \frac{c_v(T_3 - T_2) + c_p(T_4 - T_3) - c_v(T_5 - T_1)}{c_v(T_3 - T_2) + c_p(T_4 - T_3)}$$

$$= 1 - \frac{c_v(T_5 - T_1)}{c_v(T_3 - T_2) + c_p(T_4 - T_3)}$$

$$= 1 - \frac{c_v(T_5 - T_1)}{(T_3 - T_2) + \gamma(T_4 - T_3)} \qquad \ldots(i) \qquad \left(\because \ \gamma = \frac{c_p}{c_v}\right)$$

Compression ratio, $r = \dfrac{v_1}{v_2}$

During *adiabatic compression process 1-2,*

$$\frac{T_2}{T_1} = \left(\frac{v_1}{v_2}\right)^{\gamma-1} = (r)^{\gamma-1} \qquad \ldots(ii)$$

During *constant volume heating process,*

$$\frac{p_3}{T_3} = \frac{p_2}{T_2}$$

$\dfrac{T_3}{T_2} = \dfrac{p_3}{p_2} = \beta$, where β is known as **pressure or explosion ratio.**

$$T_2 = \frac{T_3}{\beta} \qquad \ldots(iii)$$

During *adiabatic expansion process,*

$$\frac{T_4}{T_5} = \left(\frac{v_5}{v_4}\right)^{\gamma-1}$$

$$= \left(\frac{r}{\rho}\right)^{\gamma-1} \qquad \ldots(iv)$$

$$\left(\because \ \frac{v_5}{v_4} = \frac{v_1}{v_4} = \frac{v_1}{v_2} \times \frac{v_2}{v_4} = \frac{v_1}{v_2} \times \frac{v_3}{v_4} = \frac{r}{\rho}, \rho \text{ being the cut-off ratio}\right)$$

During *constant pressure heating process,*

$$\frac{v_3}{T_3} = \frac{v_4}{T_4}$$

$$T_4 = T_3 \frac{v_4}{v_3} = \rho\, T_3 \qquad \ldots(v)$$

Putting the value of T_4 in the eqn. (iv), we get

$$\frac{\rho T_3}{T_5} = \left(\frac{r}{\rho}\right)^{\gamma - 1} \quad \text{or} \quad T_5 = \rho \, . \, T_3 \, . \left(\frac{\rho}{r}\right)^{\gamma - 1}$$

Putting the value of T_2 in eqn. (ii), we get

$$\frac{\dfrac{T_3}{\beta}}{T_1} = (r)^{\gamma - 1}$$

$$T_1 = \frac{T_3}{\beta} \cdot \frac{1}{(r)^{\gamma - 1}}$$

Now inserting the values of T_1, T_2, T_4 and T_5 in eqn. (i), we get

$$\eta_{\text{dual}} = 1 - \frac{\left[\rho \, . \, T_3 \left(\dfrac{\rho}{r}\right)^{\gamma - 1} - \dfrac{T_3}{\beta} \cdot \dfrac{1}{(r)^{\gamma - 1}}\right]}{\left[\left(T_3 - \dfrac{T_3}{\beta}\right) + \gamma(\rho T_3 - T_3)\right]} = 1 - \frac{\dfrac{1}{(r)^{\gamma - 1}}\left(\rho^{\gamma} - \dfrac{1}{\beta}\right)}{\left(1 - \dfrac{1}{\beta}\right) + \gamma(\rho - 1)}$$

$$\eta_{\text{dual}} = 1 - \frac{1}{(r)^{\gamma - 1}} \cdot \frac{(\beta \, . \, \rho^{\gamma} - 1)}{[(\beta - 1) + \beta\gamma(\rho - 1)]}$$

$$\mathbf{p_m} = \frac{p_1 (r)^{\gamma} [\beta(\rho - 1) + (\beta - 1) - r^{1 - \gamma}(\beta\rho^{\gamma} - 1)]}{(\gamma - 1)(r - 1)}$$

COMPARISON OF OTTO, DIESEL AND DUAL COMBUSTION CYCLES

Following are the important variable factors which are used as a basis for comparison of the cycles:

- Compression ratio.
- Maximum pressure
- Heat supplied
- Heat rejected
- Net work

Some of the above mentioned variables are fixed when the performance of Otto, Diesel and dual combustion cycles is to be compared.

Efficiency Versus Compression Ratio

Fig shows the comparison for the air standard efficiencies of the Otto, Diesel and Dual combustion cycles at various compression ratios and with given cut-off ratio for the Diesel and Dual combustion cycles. It is evident from the Fig that the air standard efficiencies increase with the increase in the compression ratio. For a given compression ratio Otto cycle is the most efficient while the Diesel cycle is the least efficient.

Note. The maximum compression ratio for the petrol engine is limited by detonation. In their respective ratio ranges, the Diesel cycle is more efficient than the Otto cycle.

For the Same Compression Ratio and the Same Heat Input

A comparison of the cycles (Otto, Diesel and Dual) on the p-v and T-s diagrams for the same compression ratio and heat supplied is shown in the Fig.4

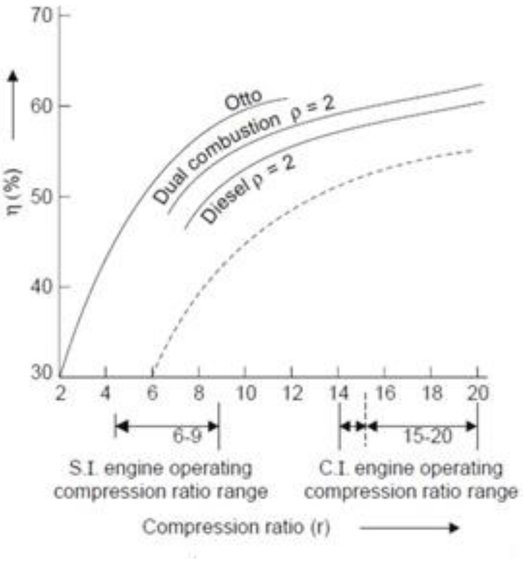


Fig 4. Efficiency Versus Compression Ratio

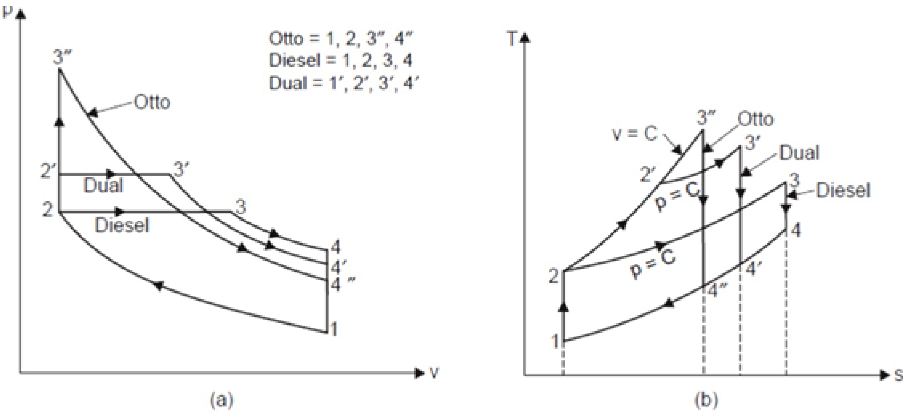


Fig 5. Combined PV diagram of all cycles

We know that, $$\eta = 1 - \frac{\text{Heat rejected}}{\text{Heat supplied}} \qquad ...(13.13)$$

Since all the cycles reject their heat at the same specific volume, process line from state 4 to 1, the quantity of heat rejected from each cycle is represented by the appropriate area under the line 4 to 1 on the T-s diagram. As is evident from the eqn. the cycle which has the least heat rejected will have the highest efficiency. Thus, Otto cycle is the most efficient and Diesel cycle is the least efficient of the three cycles.

hotto >hdual > hdiesel

For Constant Maximum Pressure and Heat Supplied

Fig. 6 shows the Otto and Diesel cycles on p-v and T-s diagrams for constant maximum pressure and heat input respectively.

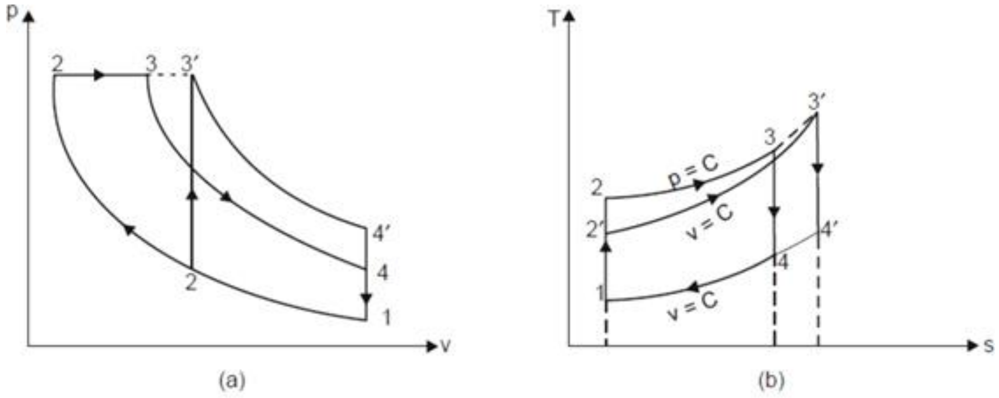
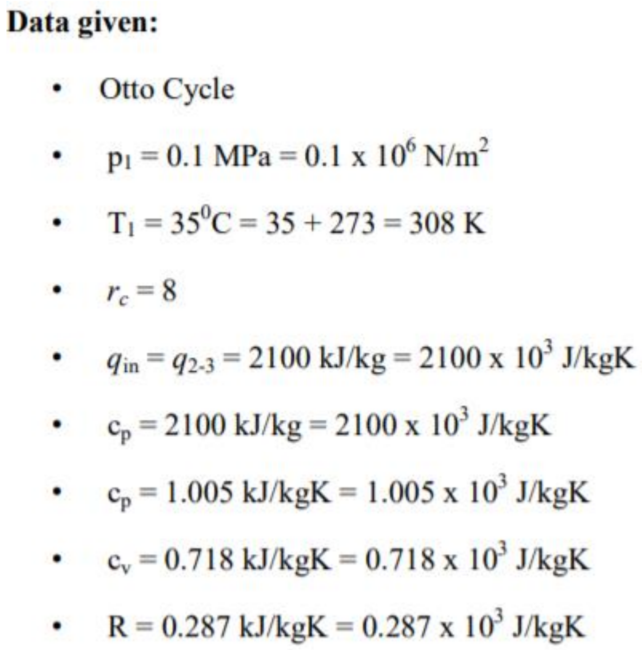


Fig 6. Constant Maximum Pressure and Heat Supplied

For the maximum pressure the points 3 and 3′ must lie on a constant pressure line.

On T-s diagram the heat rejected from the Diesel cycle is represented by the area under the line 4 to 1 and this area is less than the Otto cycle area under the curve 4′ to 1 ; hence the Diesel cycle is more efficient than the Otto cycle for the condition of maximum pressure and heat supplied.

1. An engine working on the Otto cycle is supplied with air at 0.1 MPa, 350C. The compression ratio is 8. Heat supplied is 2100 kJ/kg. Calculate the maximum pressure and temperature of the cycle, the cycle efficiency, and the mean effective pressure. (For air, cp = 1.005kJ/kgK, cv = 0.718 kJ/kgK, and R = 0.287 kJ/kgK)



Data given:

- Otto Cycle
- $p_1 = 0.1$ MPa $= 0.1 \times 10^6$ N/m^2
- $T_1 = 35^0$C $= 35 + 273 = 308$ K
- $r_c = 8$
- $q_{in} = q_{2\text{-}3} = 2100$ kJ/kg $= 2100 \times 10^3$ J/kgK
- $c_p = 2100$ kJ/kg $= 2100 \times 10^3$ J/kgK
- $c_p = 1.005$ kJ/kgK $= 1.005 \times 10^3$ J/kgK
- $c_v = 0.718$ kJ/kgK $= 0.718 \times 10^3$ J/kgK
- $R = 0.287$ kJ/kgK $= 0.287 \times 10^3$ J/kgK

To calculate:

(i) Maximum pressure (p_3)

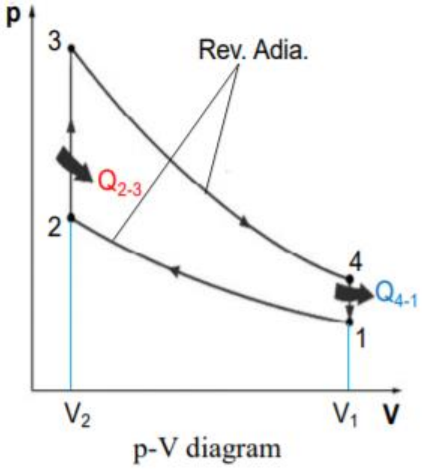
(ii) Maximum temperature (T_3)

(iii) Cycle efficiency (η)

(iv) MEP (p_m)

Solution:

Given that: $r_c = \frac{v_1}{v_2} = 8$



p-V diagram

Fig. 6.1

Cycle efficiency:

$$\eta = 1 - \frac{1}{r_c^{\gamma-1}} = 1 - \frac{1}{8^{1.4-1}} = 0.5647(56.47\%)$$

Also, Cycle efficiency:

$$\eta = \frac{w_{net}}{q_{2-3}} \Rightarrow w_{net} = \eta \times q_{2-3}$$

$$\therefore w_{net} = 0.5647 \times 2100 \times 10^3 = 1185.87 \times 10^3\, J/kg$$

At State 1:

$$p_1 v_1 = RT_1 \Rightarrow v_1 = \frac{RT_1}{p_1} = \frac{0.287 \times 10^3 \times 308}{0.1 \times 10^6}$$

$$\therefore v_1 = 0.88 m^3/kg$$

Given that:

$$r_c = \frac{v_1}{v_2} = 8 \Rightarrow v_2 = \frac{v_1}{8} = \frac{0.88}{8}$$

$$\therefore v_2 = 0.11 m^3/kg$$

Mean effective pressure:

$$p_m = \frac{w_{net}}{v_1 - v_2} = \frac{1185.87 \times 10^3}{0.88 - 0.11} = 1540.09 \times 10^3\, N/m^2$$

Given that: $c_p = 1.005$ kJ/kgK, $c_v = 0.718$ kJ/kgK

$$\gamma = \frac{c_p}{c_v} = \frac{1.005}{0.718} = 1.4$$

For isentropic process 1-2:

$$\frac{T_2}{T_1} = \left(\frac{v_1}{v_2}\right)^{\gamma-1} = (r_c)^{\gamma-1} \Rightarrow T_2 = T_1 r_c^{\gamma-1}$$

$$\therefore T_2 = 308 \times 8^{1.4-1} = 707.6K(434.6^0 C)$$

and

$$\frac{p_2}{p_1} = \left(\frac{v_1}{v_2}\right)^{\gamma} = (r_c)^{\gamma} \Rightarrow p_2 = p_1 r_c^{\gamma}$$

$$\therefore p_2 = 0.1 \times 10^6 \times 8^{1.4} = 1.84 \times 10^6 N/m^2$$

Given that: $q_{2\text{-}3} = 2100 \times 10^3$ J/kg

$$q_{2-3} = c_v (T_3 - T_2) \Rightarrow 2100 \times 10^3 = 0.718 \times 10^3 (T_3 - 707.6)$$

$$\therefore T_3 = 3632.39K(3359.39^0 C)$$

For constant volume process 2-3:

$$\frac{p_3}{p_2} = \frac{T_3}{T_2} \Rightarrow p_3 = p_2\left(\frac{T_3}{T_2}\right) = 1.84 \times 10^6 \left(\frac{3632.39}{707.6}\right)$$

$$\therefore p_3 = 9.46 \times 10^6 N/m^2$$

2. An engine of 250 mm bore and 375 mm stroke works on constant volume cycle. The clearance volume is 0.00263 m3 . The initial pressure and temperature are 1 bar and 500C. If the maximum pressure is 25 bar, determine (i) the air standard efficiency of the cycle and (ii) the mean effective pressure.

Data given:

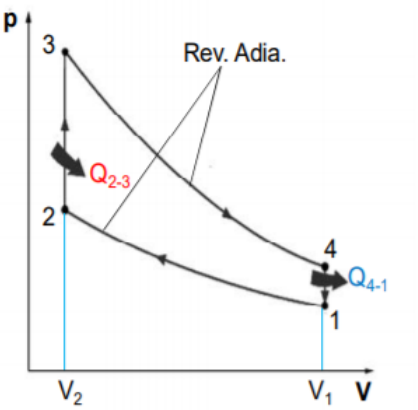
Constant Volume Cycle (Otto Cycle)

- $d = 250$ mm $= 250 \times 10^{-3}$ m

- $L = 375$ mm $= 375 \times 10^{-3}$ m
- $v_2 = v_3 = 0.00263$ m^3
- $p_1 = 1$ bar $= 1 \times 10^5$ N/m^2
- $T_1 = 50^0$C $= 50 + 273 = 323$ K
- $p_3 = 25$ bar $= 25 \times 10^5$ N/m^2

For air, assume that:

- $c_p = 1.005$kJ/kgK $= 1.005 \times 10^3$ J/kgK
- $c_v = 0.718$ kJ/kgK $= 0.718 \times 10^3$ J/kgK
- $R = 0.287$ kJ/kgK $= 0.287 \times 10^3$ J/kgK
- $\gamma = 1.4$



p-V diagram

Fig. 6.2

To determine:

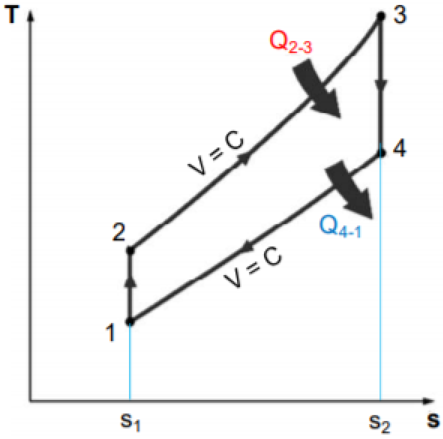
(i) Air standard efficiency (η_{Otto})

(ii) Mean effective pressure (p_m)

Solution:

Stroke volume, V$_s$:

$$V_s = \frac{\pi d^2 L}{4} = \frac{\pi \times \left(250 \times 10^{-3}\right)^2 \times 375 \times 10^{-3}}{4} = 0.0184 m^3$$



T-s diagram

Fig. 6.3

Total volume, V:

$$V = V_c + V_s = 0.00263 + 0.0184 = 0.02103 m^3$$

Compression ratio, r$_c$:

$$r_c = \frac{V_1}{V_2} = \frac{0.02103}{0.00263} = 7.996$$

Air standard efficiency, η:

$$\eta = 1 - \frac{1}{r_c^{\gamma-1}} = 1 - \frac{1}{7.996^{1.4-1}} = 0.5646 (56.46\%)$$

For isentropic process 1-2:

$$\frac{p_2}{p_1} = \left(\frac{v_1}{v_2}\right)^{\gamma} = (r_c)^{\gamma} \Rightarrow p_2 = p_1 r_c^{\gamma}$$

$$\therefore p_2 = 1\times10^5 \times 7.996^{1.4} = 18.37\times10^5\, N/m^2$$

Given that: $p_3 = 25 \times 10^5$ N/m^2

Explosion ratio, α:

$$\alpha = \frac{p_3}{p_2} = \frac{25}{18.37} = 1.36$$

Mean effective pressure, p_m:

$$p_m = \frac{p_1 r_c (\alpha - 1)\left(r_c^{\gamma-1} - 1\right)}{(\gamma - 1)(r_c - 1)}$$

$$p_m = \frac{1\times10^5 \times 7.996(1.36-1)\left(7.996^{1.4-1} - 1\right)}{(1.4-1)(7.996-1)}$$

$$\therefore p_m = 1.334\times10^5\, N/m^2$$

3. The compression ratio of an air standard diesel cycle is 16. The temperature and pressure at the beginning of isentropic compression are 150C and 0.1 MPa respectively. During the constant pressure process, the heat is added until the temperature reaches 14800C. Determine (i) the cutoff ratio (ii) the heat supplied per kg of air, (iii) the cycle efficiency, and (iv) the m.e.p

   Data given:

   - Diesel Cycle
   - $r_c = (v_1 / v_2) = 16$
   - $T_1 = 15^0$C = 15 + 273 = 288 K
   - $p_1 = 0.1$ MPa = 0.1×10^6 N/m^2
   - $T_3 = 1480^0$C = 1480 + 273 = 1753 K

For air, assume that

- $c_p = 1.005 \text{kJ/kgK} = 1.005 \times 10^3$ J/kgK
- $c_v = 0.718$ kJ/kgK $= 0.718 \times 10^3$ J/kgK
- $R = 0.287$ kJ/kgK $= 0.287 \times 10^3$ J/kgK
- $\gamma = 1.4$

To calculate:

(i) Cut-off ratio (v_3/v_2)

(ii) Heat supplied per kg of air ($q_{2\text{-}3}$)

(iii) Cycle efficiency (η)

(iv) MEP (p_m)

Solution:

For isentropic process 1-2:

$$\frac{T_2}{T_1} = \left(\frac{v_1}{v_2}\right)^{\gamma-1} = \left(r_c\right)^{\gamma-1} \Rightarrow T_2 = T_1 r_c^{\gamma-1}$$

$$\therefore T_2 = 288 \times 16^{1.4-1} = 873.05K\left(600.05^0 C\right)$$

For constant pressure process 2-3:

$$Cut-off\ \ ratio: \rho = \frac{V_3}{V_2} = \frac{T_3}{T_2} = \frac{1753}{873.05} = 2.01$$

Heat Supplied, $q_{2\text{-}3}$:

$$q_{2-3} = c_p\left(T_3 - T_2\right) = 1.005 \times 10^3 \times \left(1753 - 873.05\right)$$

$$\therefore q_{2-3} = 884.35 \times 10^3 J/kg$$

For Isentropic expansion process 3-4:

$$\frac{T_4}{T_3} = \left(\frac{\rho}{r_c}\right)^{\gamma-1} \Rightarrow T_4 = T_3\left(\frac{\rho}{r_c}\right)^{\gamma-1} = 1753 \times \left(\frac{2.01}{16}\right)^{1.4-1}$$

$$\therefore T_4 = 764.56K\left(491.56^0 C\right)$$

Heat rejected, $q_{4\text{-}1}$:

$$q_{4-1} = c_v\left(T_4 - T_1\right) = 0.718\times10^3\times\left(764.56-288\right)$$
$$\therefore q_{4-1} = 342.17\times10^3 J/kg$$

Cycle efficiency, η:

$$\eta = 1-\frac{q_{4-1}}{q_{2-3}} = 1-\frac{342.17}{884.35} = 0.6131(61.31\%)$$

At State 1:

$$p_1v_1 = RT_1 \Rightarrow v_1 = \frac{RT_1}{p_1} = \frac{0.287\times10^3\times288}{0.1\times10^6} = 0.827m^3/kg$$

Mean effective pressure, p_m:

$$p_m = \frac{w_{net}}{v_s} = \frac{q_{2-3}-q_{4-1}}{v_1-v_2} = \frac{q_{2-3}-q_{4-1}}{v_1\left(1-\frac{1}{r_c}\right)} = \frac{884.35-342.17}{0.827\left(1-\frac{1}{16}\right)}$$
$$\therefore p_m = 6.993\times10^5 N/m^2$$

INTERNAL COMBUSTION ENGINES

INTRODUCTION

As the name implies or suggests, the internal combustion engines (briefly written as IC engines) are those engines in which the combustion of fuel takes place inside the engine cylinder. These are petrol, diesel, and gas engines.

CLASSIFICATION OF IC ENGINES

The internal combustion engines may be classified in many ways, but the following are important from the subject point of view

1. According to the type of fuel used

   (a) Petrol engines. (b) Diesel engines or oil engines, and (c) Gas engines.

2. According to the method of igniting the fuel

(a) Spark ignition engines (briefly written as S.1. engines), (b) Compression ignition engines (briefly written as C.I. engines), and (c) Hot spot ignition engines

3. According to the number of strokes per cycle
   (a) Four stroke cycle engines, and (b) Two stroke cycle engines.

4. According to the cycle of operation
   (a) Otto. cycle (also known as constant volume cycle) engines, (b) Diesel cycle (also known as constant pressure cycle) engines, and (c) Dual combustion cycle (also known as semi-diesel cycle) engines.

5. According to the speed of the engine
   (a) Slow speed engines, (b) Medium speed engines, (c) High speed engines.

6. According to the cooling system
   (a) Air-cooled engines. (b) Water-cooled engines. (c) Evaporative cooling engines.

7. According to the method of fuel injection
   (a) Carburetor engines, (b) Air injection engines, (c) Airless or solid injection engines.

8. According to the number of cylinders
   (a) Single cylinder engines (b) Multi-cylinder engines.

9. According to the arrangement of cylinders
   (a) Vertical engines, (b) Horizontal engines, (c) Radial engines, (d) In-line multi-cylinder engines, (e)V-type multi-cylinder engines, (j) Opposite-cylinder engines, (g) Opposite- piston engines

MAIN COMPONENTS OF IC ENGINES

As a matter of fact, an IC engine consists of hundreds of different parts, which are important for its proper working. The description of all these parts is beyond the scope. However, the main components, which are important from academic point of view, are shown and are discussed below:

1. Cylinder. It is one of the most important part of the engine, in which the piston moves to and fro in order to develop power. Generally, the engine cylinder has to withstand a high pressure (more than 50 bar) and temperature (more than 2000°C). Thus the materials for an engine cylinder should be such that it can retain sufficient strength at such a high pressure and temperature. For ordinary engines, the cylinder is made of ordinary cast iron. But for heavy duty engines, it is made of steel alloys or aluminium alloys. In case of multiple cylinder engines, the cylinders are cast in one block known as cylinder block.Sometimes, a liner or sleeve is inserted into the cylinder, which can be replaced when worn out. As the material required for liner is comparatively small, it can be made of alloy cast iron having long life and sufficient resistance to rapid wear and tear to the fast moving reciprocating parts.

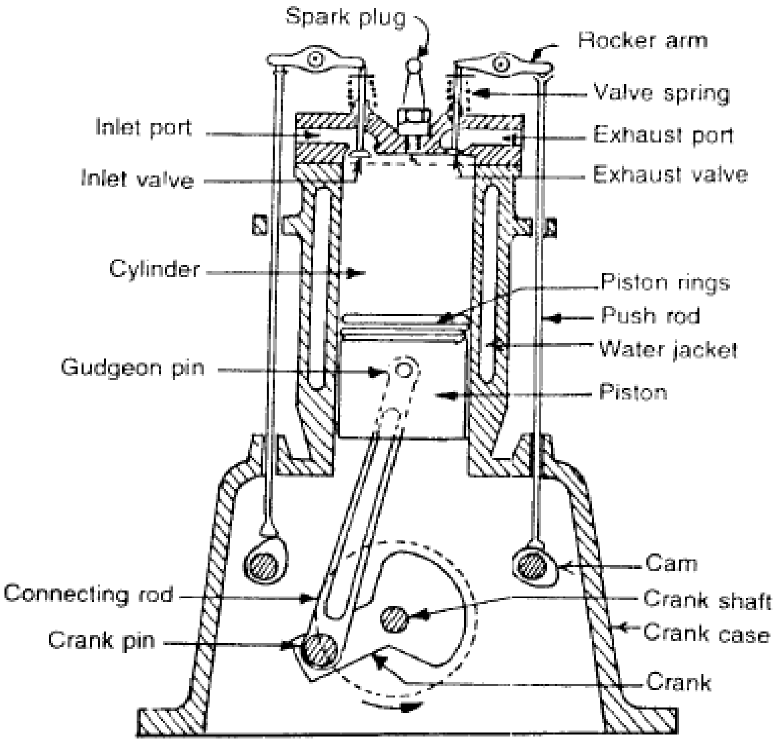


Fig. 7. MAIN COMPONENTS OF IC ENGINES

2. Cylinder head: It is fitted on one end of the cylinder, and acts as a cover to close the

cylinder bore. Generally, the cylinder head contains inlet and exit valves for admitting fresh charge and exhausting the burnt gases. In petrol engines, the cylinder head also contains a spark plug for igniting the fuel-air mixture, towards the end of compression stroke. But in diesel engines, the cylinder head contains nozzle *(i.e.* fuel valve) for injecting the fuel into the cylinder. The cylinder head is, usually, cast as one piece and bolted to one end of the cylinder. Generally, the cylinder block and cylinder head are made from the same material. A copper or asbestos gasket is provided between the engine cylinder and cylinder head to make an air-tight joint.

3. Piston: It is considered as the heart of an I.C. engine, whose main function is to transmit the force exeI1ed by the burning of charge to the connecting rod. The pistons are generally made of aluminium alloys which are light in weight. They have good heat conducting property and also greater strength at higher temperatures.

4. Piston rings: These are circular rings and made of special steel alloys which retain elastic properties even at high temperatures. The piston rings are housed in the circumferential grooves provided on the outer surface of the piston. Generally, there are two sets of rings mounted for the piston. The function of the upper rings is to provide air tight seal to prevent leakage of the burnt gases into the lower portion. Similarly, the function of the lower rings is to provide effective seal to prevent leakage of the oil into the engine cylinder.

5. Connecting rod: It is a link between the piston and crankshaft, whose main function is

to transmit force from the piston to the crankshaft. Moreover, it converts reciprocating motion of the piston into circular motion of the crankshaft, in the working stroke. The upper *(i.e.* smaller) end of the connecting rod is fitted to the piston and the lower *(i.e.* bigger) end to the crank. The special steel alloys or aluminium alloys are used for the manufacture of connecting rods. A special care is required for the design and manufacture of connecting rod, as it is subjected to alternatively compressive and tensile stresses as well as bending stresses.

6. Crankshaft: It is considered as the backbone of an l.c. engine whose function is to

convert the reciprocating motion of the piston into the rotary motion with the help of connecting rod. This shaft contains one or more eccentric portions called cranks. That part of the crank, to which bigger end of the connecting rod is fitted, is called crank pin.It has been experienced that too many main bearings create difficulty of correct alignment. Special steel alloys are used for the manufacture of crankshaft. A special care is required for the design and manufacture of crankshaft.

7. Crank case: It is a cast iron case, which holds the cylinder and crankshaft of an I.c.

engine. It also serves as a sump for the lubricating oil. The lower portion of the crank case is known as bed plate, which is fixed with the help of bolts.

8. Flywheel: It is a big wheel, mounted on the crankshaft, whose function is to maintain its speed constant. It is done by storing excess energy during the power stroke, which is returned during other strokes.

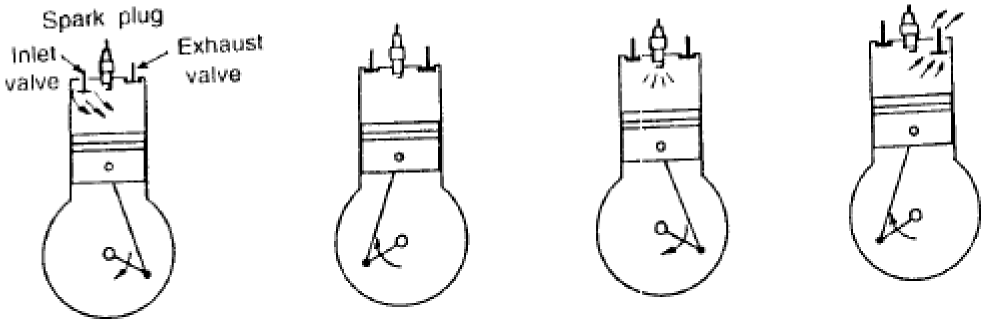
FOUR STROKE CYCLE PETROL ENGINES

It is also known as Otto cycle. It requires four strokes of the piston to complete one cycle of operation in the engine cylinder. The four strokes of a petrol engine sucking fuel-air mixture (petrol mixed with proportionate quantity of air in the carburetor known as charge) are described below:

1. Suction or charging stroke: In this stroke, the inlet valve opens and charge is sucked

into the cylinder as the piston moves downward from top dead centre *(T.D.C.).* It continues till the piston reaches its bottom dead centre *(B.D.* C.) as shown in (a).

2. Compression stroke: In this stroke, both the inlet and exhaust valves are closed and the charge is compressed as the piston moves upwards from *B.D.* C. to *TD.* C. As a result of compression, the pressure and temperature of the charge increases considerably (the actual values depend upon the compression ratio). This completes one revolution of the crank shaft. The compression stroke is shown in (b).



(*a*) Suction or charging stroke. (*b*) Compression stroke. (*c*) Expansion or working stroke. (*d*) Exhaust stroke.

Four-stroke cycle petrol engine.

Fig 8. FOUR STROKE CYCLE PETROL ENGINES

3. Expansion or working stroke Shortly before the piston reaches *T.D.*C. (during compression stroke), the charge is ignited with the help of a spark plug. It suddenly increases the pressure and temperature of the products of combustion but the volume, practically, remains constant. Due to the rise in pressure, the piston is pushed down with a great force. The hot burnt gases expand due to high speed of the piston. During this expansion, some of the heat energy produced is transformed into mechanical work. It may be noted that during this working stroke, as shown in *(c),* both the valves are closed and piston moves from *T.D.C.* to *B.D.C*

4. Exhaust stroke: In this stroke, the exhaust valve is open as piston moves from *B.D.C.* to *T.D.*C. This movement of the piston pushes out the products of combustion, from the engine cylinder and are exhausted through the exhaust valve into the atmosphere, as shown in *(d).* This completes the cycle, and the engine cylinder is ready to suck the charge again.

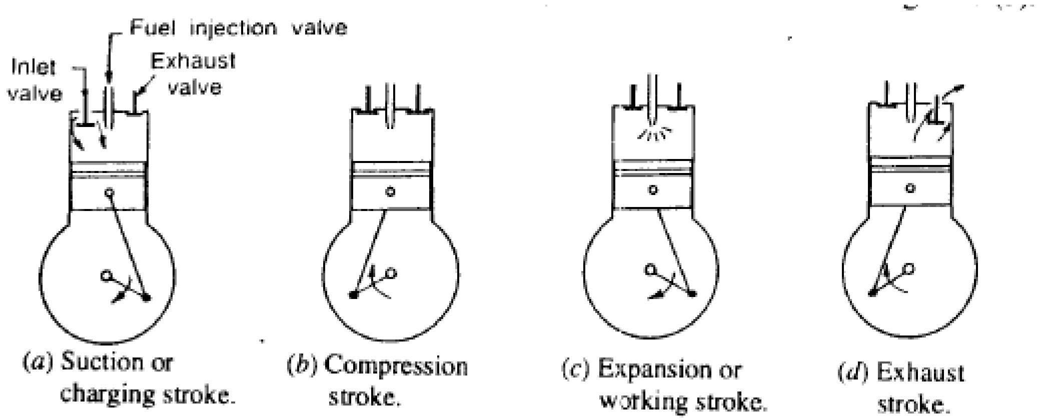
FOUR-STROKE CYCLE DIESEL ENGINE

It is also known as *compression ignition engine* because the ignition takes p\ace due to the heat produced in the engine cylinder at the end of compression stroke. The four strokes of a diesel engine sucking pure air are described below:

1. Suction or charging stroke: In this stroke, the inlet valve opens and pure air is sucked

into the cylinder as the piston moves downwards from the top dead centre *(TDC).* It continues till the piston reaches its bottom dead centre *(BDC)* as shown *(a).*

2. Compression stroke: In this stroke, both the valves are closed and the air is compressed as the piston moves upwards from *BDC* to *TDC.* As a result of compression, pressure and temperature of the air increases considerably (the actual value depends upon the compression ratio). This completes one revolution of the crank shaft. The compression stroke is shown in *(b).*



(*a*) Suction or charging stroke. (*b*) Compression stroke. (*c*) Expansion or working stroke. (*d*) Exhaust stroke.

Fig 9. FOUR-STROKE CYCLE DIESEL ENGINE

3. Expansion or working stroke: Shortly before the piston reaches the *TDC* (during the compression stroke), fuel oil is injected in the form of very fine spray into the engine cylinder, through the nozzle, known as fuel injection valve. At this moment temperature of the compressed air is sufficiently high to ignite the fuel. It suddenly increases the pressure and temperature of the products of combustion. The fuel oil is continuously injected for a fraction of the revolution. The fuel oil is assumed to be burnt at constant pressure. Due to increased pressure, the piston is pushed down with a great force. The hot burnt gases expand due to high speed of the piston. During this expansion, some of the heat energy is transformed into mechanical work. It may be noted that during this working stroke, both the valves are closed and the piston moves from *TDC* to *BDC.*

4. Exhaust stroke: In this stroke, the exhaust valve is open as the piston moves from *BDC* to *TDC.* This movement of the piston pushes out the products of combustion from the engine cylinder through the exhaust valve into the atmosphere. This completes the cycle and the engine cylinder is ready to suck the fresh air again.

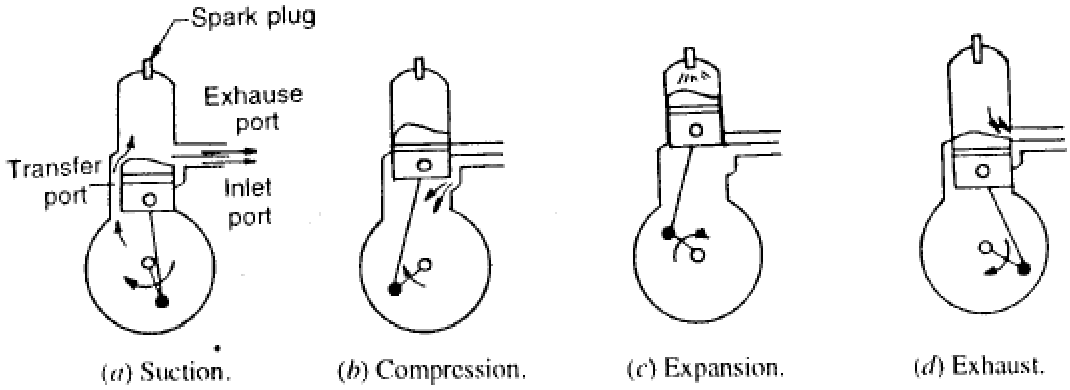
TWO-STROKE CYCLE PETROL ENGINE

A two-stroke cycle petrol engine was devised by Duglad Clerk in I RHO. In this cycle, the suction, compression, expansion and exhaust takes place during two strokes of the piston. It means that there is one working stroke after every revolution of the crank shaft. A two stroke engine has ports instead of valves. All the four stages of a two stroke petrol engine are described below:

1. Suction stage: In this stage, the piston, while going down towards *BDC*, uncovers both the transfer port and the exhaust port The fresh fuel-air mixture flows into the engine cylinder from the crank case, as shown *(a).*

2. Compression stage: In this stage, the piston, while moving up, first covers the transfer port and then exhaust port. After that the fuel is compressed as the piston moves upwards as shown *(b).* In this stage, the inlet port opens and fresh fuel-air mixture enters into the crank case.

3. Expansion stage: Shortly before this piston reaches the *TDC* (during compression stroke), the charge is ignited with the help of a spark plug. It suddenly increases the pressure and temperature of the products of combustion. But the volume, practically, remains constant. Due to rise in the pressure, the piston is pushed downwards with a great force as shown in *(c).* The hot burnt gases expand due to high speed of the piston. During this expansion, some of the heat energy produced is transformed into mechanical work.



(*a*) Suction. (*b*) Compression. (*c*) Expansion. (*d*) Exhaust.

Fig. 10. TWO-STROKE CYCLE PETROL ENGINE

1. **Exhaust stage:** In this stage, the exhaust port is opened as the piston moves downwards. The products of combustion, from the engine cylinder are exhausted through the exhaust port into the atmosphere, as shown *(d).* This completes the cycle and the engine cylinder is ready to suck the charge again

TWO STROKE AND FOUR STROKE CYCLE ENGINES

In a two-stroke engine, the working cycle is completed in two strokes of the piston or one revolution of the crankshaft. This is achieved by carrying out the suction and compression processes in one stroke (or more precisely in inward stroke), expansion and exhaust processes in the second stroke (or more precisely in outward stroke). In a four-stroke engine, the working cycle is completed in four-strokes of the piston or two-revolutions of the crankshaft. This is achieved by carrying out suction, compression, expansion and exhaust processes in each stroke. It will be interesting to know that from the thermodynamic point of view, there is no difference between two-stroke and four-stroke cycle engines. The difference is purely mechanical.

Advantages and Disadvantage of Two-stroke over Four-stroke Cycle Engines

Advantages

1. A two stroke cycle engine gives twice the number of power strokes than the four stroke cycle engine at the same engine speed. Theoretically, a two-stroke cycle engine should develop twice the power as that of a four-stroke cycle engine. But in actual practice, a two-stroke cycle engine develops 1.7 to 1.8 times greater value for slow speed engines the power developed by four-stroke cycle engine of the same dimensions and speed. This is due to lower compression ratio and effective stroke being less than the theoretical stroke.
2. For the same power developed, a two-stroke cycle engine is lighter, less bulky and occupies less floor area. Thus it makes, a two-stroke cycle engine suitable for marine engines and other light vehicles.
3. As the number of working strokes in a two-stroke cycle engine are twice than the four-stroke cycle engine, so the turning moment of a two-stroke cycle engine is more uniform. Thus it makes a two-stroke cycle engine to have a lighter flywheel and foundations. This also leads to a higher mechanical efficiency of a two-stroke cycle engine.
4. The initial cost of a two-stroke cycle engine is considerably less than a four-stroke cycle engine.
5. The mechanism of a two-stroke cycle engine is much simpler than a four-stroke cycle engine.
6. The two-stroke cycle engines are much easier to start.

Disadvantages

1. Thermal efficiency of a two-stroke cycle engine is less than that a four-stroke cycle engine, because a two-stroke cycle engine has less compression ratio than that of a four-stroke cycle engine.
2. Overall efficiency of a two stroke cycle engine is also less than that of a four-stroke cycle engine because in a two-stroke cycle, inlet and exhaust ports remain open simultaneously for some time. In spite of careful design, a small quantity of charge is lost from the engine cylinder.
3. In case of a two-stroke cycle engine, the number of power strokes is twice as those of a four-stroke cycle engine. Thus the capacity of the cooling system must be higher. Beyond a certain limit, the cooling capacity offers a considerable difficulty. Moreover, there is a greater wear and tear in a two-stroke cycle engine.
4. The consumption of lubricating oil is large in a two-stroke cycle engine because of high operating temperature.
5. The exhaust gases in a two-stroke cycle engine create noise, because of short time available for their exhaust.

comparison of petrol and diesel engines (SI and CI engines)

Following points are important for the comparison of petrol engines and diesel engines:

Table.1. comparison of petrol and diesel engines (SI and CI engines)

Petrol Engines	Diesel Engines
• A petrol engine draws a mixture of petrol and air during suction stroke. • The carburetor is employed to mix air and petrol in the required proportion and to supply it to the engine during suction stroke • Pressure at the end of compression is about 10 bar	• A diesel engine draws only air during suction stroke • The injector or atomizer is employed to inject the fuel at the end of compression stroke. • Pressure at the end of compression is about 35 bar. • The fuel is injected in the form of

• The charge *(i.e.* petrol and air mixture) is ignited with the help of spark plug • The combustion of fuel takes place approximately at constant volume. In other words, it works on Otto cycle • A petrol engine has compression ratio approximately from 6 to 10. • The starting' is easy due to low compression ratio. • As the compression ratio is low, the petrol engines are lighter and cheaper. • The running cost of a petrol engine is high because of the higher cost of petrol. • The maintenance cost is less. • The thermal efficiency is up to about 26%. • Overheating trouble is more due to low thermal efficiency. • These are high speed engines. • The petrol engines arc generally employed in light duty vehicles such as scooters, motorcycles, cars. These are also used in aero planes	fine spray. The temperature of the compressed air (about 600"C at a pressure of about *35*bar) is sufficiently high to ignite the fuel. • The combustion of fuel takes place approximately at constant pressure. In other words. It works on Diesel cycle. • A diesel engine has compression ratio approximately from 15 to 25. • The starting is little difficult due. to high compression ratio. • As the compression ratio is high. the diesel engine;; are heavier and costlier. • The running cost of diesel engine is low because of the lower cost of diesel. • The maintenance cost is more. • The thermal efficiency is up to about 40% • Overheating trouble is less due to high thermal efficiency • These are relatively low speed engines. • The diesel engines are generally employed in heavy duty vehicles like buses. trucks, and earth moving machines etc.

Rankine Cylce:

Rankine cycle is the theoretical cycle on which the steam turbine (or engine) works.

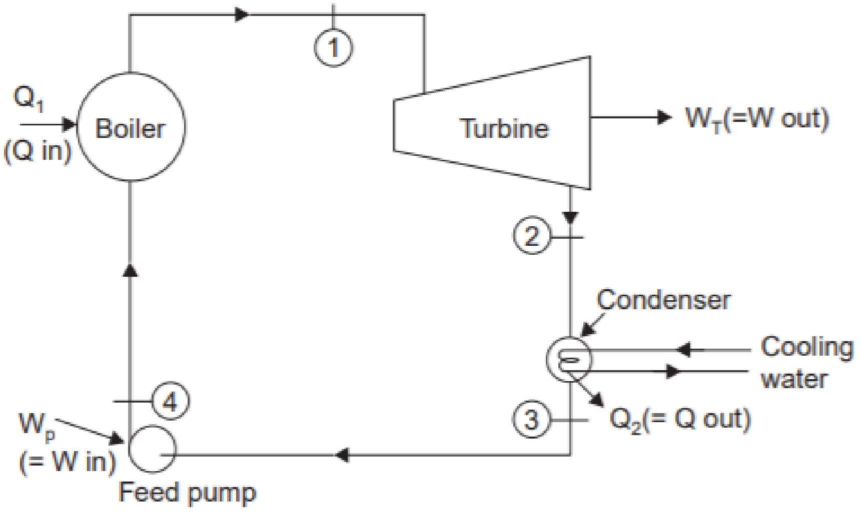


Fig.11. Rankine cycle.

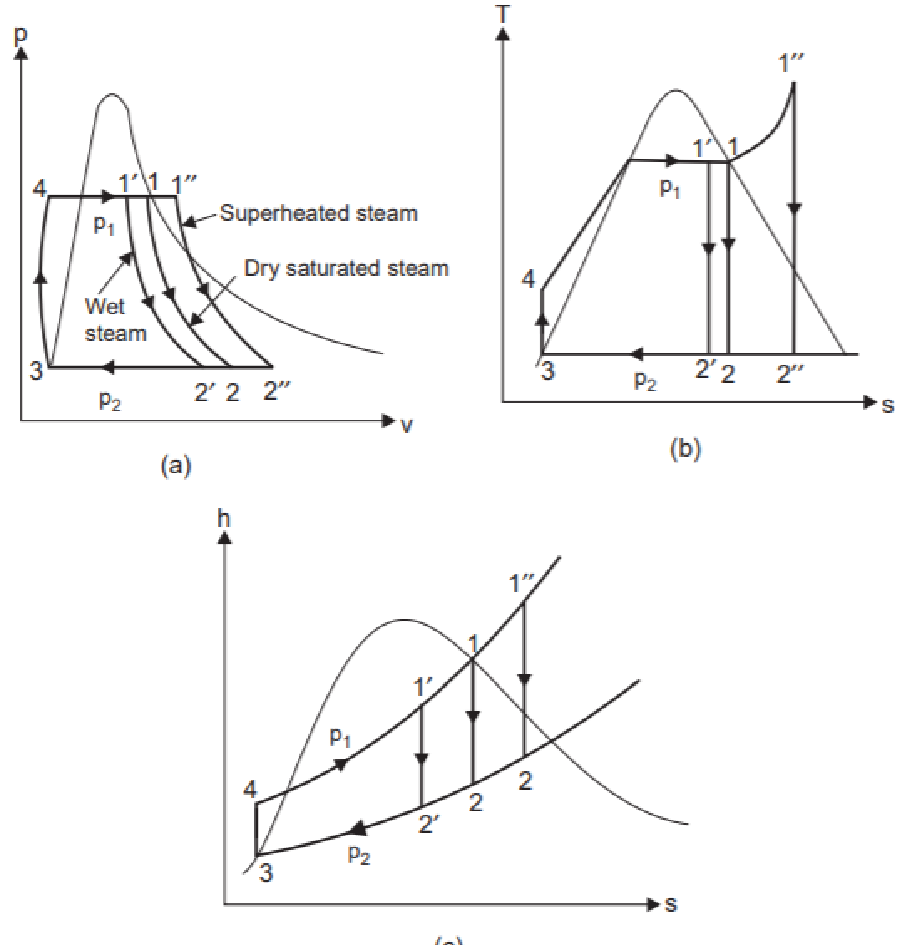


Fig.12. (a) p-v diagram ; (b) T-s diagram ; (c) h-s diagram for Rankine cycle

The Rankine cycle is shown in Fig. It comprises of the following processes : Process 1-2 : Reversible adiabatic expansion in the turbine (or steam engine). Process 2-3 : Constant-pressure transfer of heat in the condenser. Process 3-4 : Reversible adiabatic pumping process in the feed pump. Process 4-1 : Constant-pressure transfer of heat in the boiler. Fig. 12.3 shows the Rankine cycle on p-v, T-s and h-s diagrams (when the saturated steam enters the turbine, the steam can be wet or superheated also).

Considering 1 kg of fluid :

Applying steady flow energy equation (S.F.E.E.) to boiler, turbine, condenser and pump :

(i) **For boiler** (as control volume), we get

$$h_{f_4} + Q_1 = h_1$$

$$\therefore \quad Q_1 = h_1 - h_{f_4}$$

(*ii*) **For turbine** (as control volume), we get

$$h_1 = W_T + h_2, \text{ where } W_T = \text{turbine work}$$

$$\therefore \quad W_T = h_1 - h_2$$

(*iii*) **For condenser**, we get

$$h_2 = Q_2 + h_{f_3}$$

$$\therefore \quad Q_2 = h_2 - h_{f_3}$$

(*iv*) **For the feed pump**, we get

$$h_{f_3} + W_P = h_{f_4}, \quad \text{where, } W_P = \text{Pump work}$$

$$\therefore \quad W_P = h_{f_4} - h_{f_3}$$

Now, efficiency of Rankine cycle is given by

$$\eta_{\text{Rankine}} = \frac{W_{\text{net}}}{Q_1} = \frac{W_T - W_P}{Q_1}$$

$$= \frac{(h_1 - h_2) - (h_{f_4} - h_{f_3})}{(h_1 - h_{f_4})}$$

The feed pump handles liquid water which is incompressible which means with the increase in pressure its density or specific volume undergoes a little change. Using general property relation for reversible adiabatic compression, we get

$$Tds = dh - vdp$$

$$\because \quad ds = 0$$

$$\therefore \quad dh = vdp$$

or $$\Delta h = v\,\Delta p \quad \text{...... (since change in specific volume is negligible)}$$

or $$h_{f_4} - h_{f_3} = v_3\,(p_1 - p_2)$$

When p is in bar and v is in m^3/kg, we have

$$h_{f_4} - h_{f_3} = v_3\,(p_1 - p_2) \times 10^5 \text{ J/kg}$$

The feed pump term ($h_{f_4} - h_{f_3}$) being a small quantity in comparison with turbine work, W_T, is usually neglected, *especially when the boiler pressures are low.*

Then, $$\eta_{\text{Rankine}} = \frac{h_1 - h_2}{h_1 - h_{f_4}}$$

Effect of Operating Conditions on Rankine Cycle Efficiency:

The Rankine cycle efficiency can be improved by : (i) Increasing the average temperature at which heat is supplied. (ii) Decreasing/reducing the temperature at which heat is rejected.

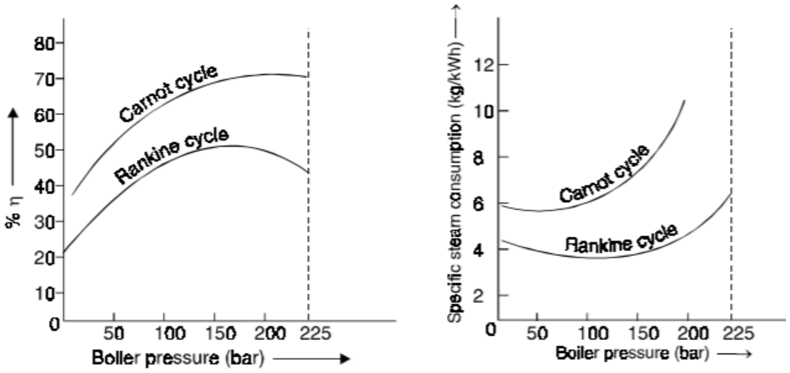


Fig. 13. Effect of Operating Conditions on Rankine Cycle Efficiency:

This can be achieved by making suitable changes in the conditions of steam generation or condensation, as discussed below :

1. **Increasing boiler pressure.** It has been observed that by increasing the boiler pressure (other factors remaining the same) the cycle tends to rise and reaches a maximum value at a boiler pressure of about 166 bar.

2. **Superheating.** All other factors remaining the same, if the steam is superheated before allowing it to expand the Rankine cycle efficiency may be increased. The use of superheated steam also ensures longer turbine blade life because of the absence of erosion from high velocity water particles that are suspended in wet vapour.

3. **Reducing condenser pressure**. The thermal efficiency of the cycle can be amply improved by reducing the condenser pressure (hence by reducing the temperature at which heat is rejected), especially in high vacuums. But the increase in efficiency is obtained at the increased cost of condensation apparatus.

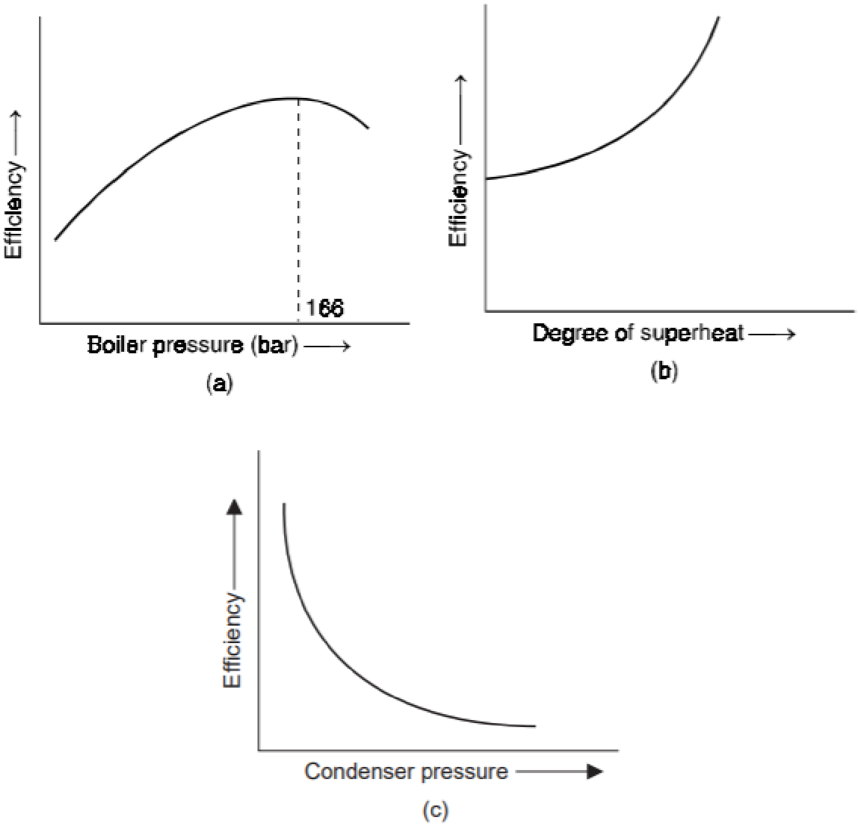


Fig.14. Effect of operating conditions on the thermal efficiency of the Rankine cycle.

The thermal efficiency of the Rankine cycle is also improved by the following methods : (i) By regenerative feed heating. (ii) By reheating of steam. (iii) By water extraction. (iv) By using binary-vapour

Problem 4: In a steam power cycle, the steam supply is at 15 bar and dry and saturated. The condenser pressure is 0.4 bar. Calculate the Carnot and Rankine efficiencies of the cycle. Neglect pump work.

Solution. Steam supply pressure, $p_1 = 15$ bar, $x_1 = 1$

Condenser pressure, $p_2 = 0.4$ bar

Carnot and Rankine efficiencies :

From steam tables :

At 15 bar : $t_s = 198.3°C$, $h_g = 2789.9$ kJ/kg, $s_g = 6.4406$ kJ/kg K

At 0.4 bar : $t_s = 75.9°C$, $h_f = 317.7$ kJ/kg, $h_{fg} = 2319.2$ kJ/kg,

$$s_f = 1.0261 \text{ kJ/kg K}, \quad s_{fg} = 6.6448 \text{ kJ/kg K}$$

$$T_1 = 198.3 + 273 = 471.3 \text{ K}$$

$$T_2 = 75.9 + 273 = 348.9 \text{ K}$$

$$\therefore \quad \eta_{\text{carnot}} = \frac{T_1 - T_2}{T_1} = \frac{471.3 - 348.9}{471.3}$$

$$= \mathbf{0.259 \text{ or } 25.9\%. \text{ (Ans.)}}$$

$$\eta_{\text{Rankine}} = \frac{\text{Adiabatic or isentropic heat drop}}{\text{Heat supplied}} = \frac{h_1 - h_2}{h_1 - h_{f_2}}$$

where $h_2 = h_{f_2} + x_2\, h_{fg_2} = 317.7 + x_2 \times 2319.2$ ...(*i*)

Value of x_2 :

As the steam expands isentropically,

$$\therefore \quad s_1 = s_2$$

$$6.4406 = s_{f_2} + x_2\, s_{fg_2} = 1.0261 + x_2 \times 6.6448$$

$$\therefore \quad x_2 = \frac{6.4406 - 1.0261}{6.6448} = 0.815$$

$$\therefore \quad h_2 = 317.7 + 0.815 \times 2319.2 = 2207.8 \text{ kJ/kg} \qquad \text{[From eqn. } (i)]$$

Hence, $\eta_{\text{Rankine}} = \dfrac{2789.9 - 2207.8}{2789.9 - 317.7} = \mathbf{0.2354 \text{ or } 23.54\%.}$ **(Ans.)**

Problem 5: In a steam turbine steam at 20 bar, 360°C is expanded to 0.08 bar. It then enters a condenser, where it is condensed to saturated liquid water. The pump feeds back the water into the boiler. Assume ideal processes, find per kg of steam the net work and the cycle efficiency.

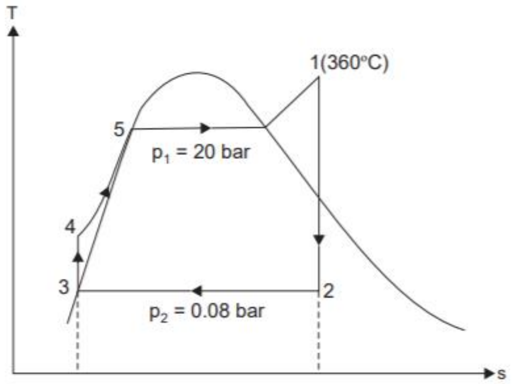


Fig.14.1

Solution. Boiler pressure, $p_1 = 20$ bar (360°C)

Condenser pressure, $p_2 = 0.08$ bar

From steam tables :

At 20 bar (p_1), 360°C : $h_1 = 3159.3$ kJ/kg

$s_1 = 6.9917$ kJ/kg-K

At 0.08 bar (p_2) : $h_3 = h_{f(p_2)} = 173.88$ kJ/kg,

$s_3 = s_{f(p_2)} = 0.5926$ kJ/kg-K

$h_{fg(p_2)} = 2403.1$ kJ/kg, $\quad s_{g(p_2)} = 8.2287$ kJ/kg-K

$v_{f(p_2)} = 0.001008$ m³/kg $\quad \therefore \quad s_{fg(p_2)} = 7.6361$ kJ/kg-K

Now $\quad s_1 = s_2$

$$6.9917 = s_{f(p_2)} + x_2\, s_{fg(p_2)} = 0.5926 + x_2 \times 7.6361$$

$$\therefore \qquad x_2 = \frac{0.69917 - 0.5926}{7.6361} = 0.838$$

$$\therefore \qquad h_2 = h_{f(p_2)} + x_2\, h_{fg(p_2)}$$

$$= 173.88 + 0.838 \times 2403.1 = 2187.68 \text{ kJ/kg.}$$

Net work, W_{net} :

$$W_{net} = W_{turbine} - W_{pump}$$

$$W_{pump} = h_{f_4} - h_{f(p_2)} \;(= h_{f_3}) = v_{f(p_2)}\;(p_1 - p_2)$$

$$= 0.00108 \text{ (m}^3\text{/kg)} \times (20 - 0.08) \times 100 \text{ kN/m}^2$$

$$= 2.008 \text{ kJ/kg}$$

[and $\quad h_{f_4} = 2.008 + h_{f(p_2)} = 2.008 + 173.88 = 175.89$ kJ/kg]

$$W_{turbine} = h_1 - h_2 = 3159.3 - 2187.68 = 971.62 \text{ kJ/kg}$$

$\therefore \qquad W_{net} = 971.62 - 2.008 =$ **969.61 kJ/kg. (Ans.)**

Cycle efficiency, η_{cycle} :

$$Q_1 = h_1 - h_{f_4} = 3159.3 - 175.89 = 2983.41 \text{ kJ/kg}$$

$\therefore \qquad \eta_{cycle} = \dfrac{W_{net}}{Q_1} = \dfrac{969.61}{2983.41} =$ **0.325 or 32.5%. (Ans.)**

Problem 6: A Rankine cycle operates between pressures of 80 bar and 0.1 bar. The maximum cycle temperature is 600°C. If the steam turbine and condensate pump efficiencies are 0.9 and 0.8 respectively, calculate the specific work and thermal efficiency. Relevant steam table extract is given below.

p(bar)	*t(°C)*	*Specific volume (m³/kg)*		*Specific enthalpy (kJ/kg)*			*Specific entropy (kJ/kg K)*		
		v_f	v_g	h_f	h_{fg}	h_g	s_f	s_{fg}	s_g
0.1	*45.84*	*0.0010103*	*14.68*	*191.9*	*2392.3*	*2584.2*	*0.6488*	*7.5006*	*8.1494*
80	*295.1*	*0.001385*	*0.0235*	*1317*	*1440.5*	*2757.5*	*3.2073*	*2.5351*	*5.7424*

At 80 bar, 600°C :

$h_1 = 3642$ kJ / kg ;

$s_1 = 7.0206$ kJ / kg K.

Since $s_1 = s_2$,

$$\therefore\ 7.0206 = s_{f_2} + x_2\, s_{fg_2}$$

$$= 0.6488 + x_2 \times 7.5006$$

$$x_2 = \frac{7.0206 - 0.6488}{7.5006} = 0.85$$

Now, $h_2 = h_{f_2} + x_2\, h_{fg_2}$

$$= 191.9 + 0.85 \times 2392.3$$

$$= 2225.36 \text{ kJ/kg}$$

Actual turbine work

$$= \eta_{turbine} \times (h_1 - h_2)$$

$$= 0.9\,(3642 - 2225.36) = 1275 \text{ kJ/kg}$$

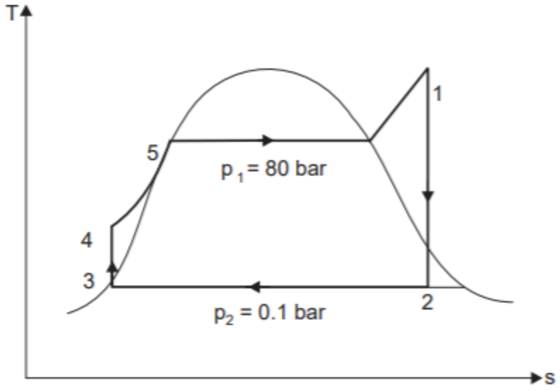


Fig. 14.2

Pump work $= v_{f(p_2)}\,(p_1 - p_2)$

$$= 0.0010103\,(80 - 0.1) \times \frac{10^5}{10^3} \text{ kN/m}^2 = 8.072 \text{ kJ/kg}$$

Actual pump work $= \dfrac{8.072}{\eta_{pump}} = \dfrac{8.072}{0.8} = 10.09$ kJ/kg

Specific work $(W_{net}) = 1275 - 10.09 =$ **1264.91 kJ / kg. (Ans.)**

Thermal efficiency $= \dfrac{W_{net}}{Q_1}$

where, $Q_1 = h_1 - h_{f_4}$

But $h_{f_4} = h_{f_3}$ + pump work = 191.9 + 10.09 = 202 kJ/kg

$\therefore$ Thermal efficiency, $\eta_{th} = \dfrac{1264.91}{3642 - 202} =$ **0.368 or 36.8 %. (Ans.)**

Principles of Refrigeration and Airconditioning:

INTRODUCTION

For specific applications, efficiencies of both living and non-living beings depend to a great extent on the physical environment. The nature keeps conditions in the physical environment in the dynamic state ranging from one extreme to the other. Temperature, humidity, pressure and air motion are some of the important environment variables that at any location keep changing throughout the year. Adaptation to these many a times unpredictable variations are not possible and thus working efficiently is not feasible either for the living beings or the non-living ones. Thus for any specific purpose, control of the environment is essential. Refrigeration and air-conditioning is the subject which deals with the techniques to control the environments of the living and non-living subjects and thus provide them comforts to enable them to perform better and have longer lives.

DEFINITIONS

Refrigeration:

Refrigeration is defined as a method of reducing the temperature of a system below that of the surroundings and maintaining it at the lower temperature by continuously extracting the heat from it.

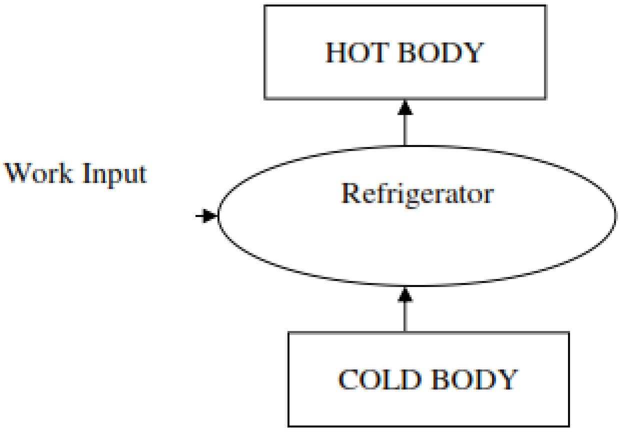


Fig.15. Refrigerator cycles

The principle of refrigeration is based on second law of thermodynamics. It states that heat does not flow from a low temperature body to a high temperature body without the help of an external work. In refrigeration process, the heat is continuously removed from a system at lower temperature and transfers it to the surroundings at a higher temperature. This operation according to second law of thermodynamics can only be performed by the aid of the external work. Therefore in a refrigeration system, power is to be supplied to remove heat continuously from the refrigerator to keep it cool at a temperature less than the surroundings. The refrigeration cycle is based on reversible Carnot cycle.

Refrigeration effect:

The rate at which the heat is absorbed in a cycle of from the interior space to be cooled is called refrigeration effect. It is defined as the quantity of heat removed to the time taken. It is also called as the capacity of a refrigerator.

Ton of Refrigeration (or) Unit of Refrigeration (TR):

The standard unit of refrigeration is *ton refrigeration* or simply *ton* denoted by TR. It is equivalent to the rate of heat transfer needed to produce 1 ton (2000 lbs) of ice at 32 ^{0}F from water at 32 ^{0}F in one day, i.e., 24 hours. The enthalpy of solidification of water from and at 32 ^{0}F in British thermal unit is 144 Btu/lb. Thus

$$1\ \text{TR} = \frac{2000\,\text{lb} \times 144\,\text{Btu/lb}}{24\,\text{hr}}$$

$$= 12000\ \text{Btu/hr} = 200\ \text{Btu/min}$$

In general, 1 TR means 200 Btu of heat removal per minute. Thus if a refrigeration system is capable of cooling at the rate of 400 Btu/min, it is a 2 ton machine. A machine of 20 ton rating is capable of cooling at a rate of 20 × 200 = 4000 Btu/min. This unit of refrigeration is currently in use in the USA, the UK and India. In many countries, the standard MKS unit of kcal/hr is used. In the MKS it can be seen that

$$\begin{aligned} 1\ \text{TR} = 12000\ \text{Btu/hr} &= \frac{12000}{3.968} = 3024.2\ \text{kcal/hr} \\ &= 50.4\ \text{kcal/min} \approx 50\ \text{kcal/min} \end{aligned}$$

If Btu ton unit is expressed into SI system, it is found to be **210 kJ/min or 3.5 kW.**

Co-efficient of Performance (COP):

The Co-efficient of Performance is defined as the ratio of heat absorbed in a system to the work supplied.

The theoretical Coefficient of Performance (Carnot), (COP a standard measure of refrigeration efficiency of an ideal refrigeration system) depends on two key system temperatures: evaporator temperature T_e and condenser temperature T_c

COP is given as: $COP_{Carnot} = T_e / (T_c - T_e)$

This expression also indicates that higher COP_{Carnot} is achieved with higher evaporator temperatures and lower condenser temperatures. But COP is only a ratio of temperatures, and does not take into account the type of compressor. Hence the COP normally used in industry is calculated as follows:

$$COP = \frac{\text{Cooling effect (kW)}}{\text{Power input to compressor (kW)}}$$

Where the cooling effect is the difference in enthalpy across the evaporator and expressed as kW.

Ice making capacity:

It is the ability of the refrigeration system to make ice. In other words, it is the capacity of refrigeration system to remove heat from water to make ice.

Relative COP:

It is the ratio of actual COP to the theoretical COP of a refrigerator. Actual COP is measured during a test and theoretical COP is obtained by applying the laws of thermodynamics.

REFRIGERATOR AND HEAT PUMP

The vapor compression refrigeration cycle is a common method for transferring heat from a low temperature to a high temperature.

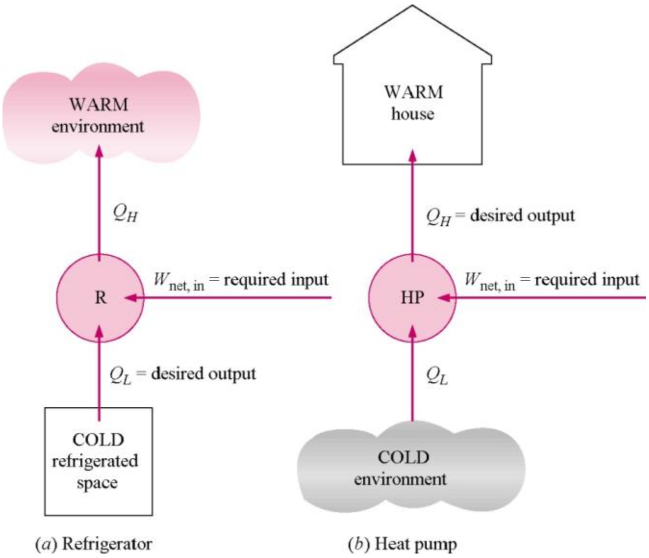


Fig. 16 Refrigerator and Heat pump

The above figure shows the objectives of refrigerators and heat pumps. The purpose of a refrigerator is the removal of heat, called the cooling load, from a low temperature medium. The purpose of a heat pump is the transfer of heat to a high temperature medium, called the heating load. When we are interested in the heat energy removed from a low temperature space, the device is called a refrigerator. When we are interested in the heat energy supplied to the high temperature space, the device is called a heat pump. In general, the term "heat pump" is used to describe the cycle as heat energy is removed from the low temperature space and rejected to the high temperature space.

The performance of refrigerators and heat pumps is expressed in terms of *coefficient of performance* (COP), defined as

$$COP_R = \frac{\text{Desired output}}{\text{Required input}} = \frac{\text{Cooling effect}}{\text{Work input}} = \frac{Q_L}{W_{net,in}}$$

$$COP_{HP} = \frac{\text{Desired output}}{\text{Required input}} = \frac{\text{Heating effect}}{\text{Work input}} = \frac{Q_H}{W_{net,in}}$$

Both COP_R and COP_{HP} can be larger than 1. Under the same operating conditions, the COPs are related by

$$COP_{HP} = COP_R + 1$$

TYPES OF REFRIGERATION

Refrigeration is classified as based on working substance used

- Air refrigeration system (Bell-Coleman cycle)
- Water refrigeration system
- Ice refrigeration system
- Refrigeration by special fluid (low boiling point fluids – Refrigerants) (Reversed Carnot cycle)
  - Vapour compression refrigeration system (VCR)
  - Vapour absorbtion refrigeration system (VAR)
  - Vapour adsorbtion refrigeration system and etc.,

Simple Vapour Compression Refrigeration System (VCR)

It consists of the following essential parts:

Compressor

The low pressure and temperature vapour refrigerant from evaporator is drawn into the compressor through the inlet or suction valve A, where it is compressed to a high pressure and temperature. This high pressure and temperature vapour refrigerant is discharged into the condenser through the delivery or discharge valve B.

Condenser

The condenser or cooler consists of coils of pipe in which the high pressure and temperature vapour refrigerant is cooled and condensed. The refrigerant, while passing through the condenser, gives up its latent heat to the surrounding condensing medium which is normally air or water.

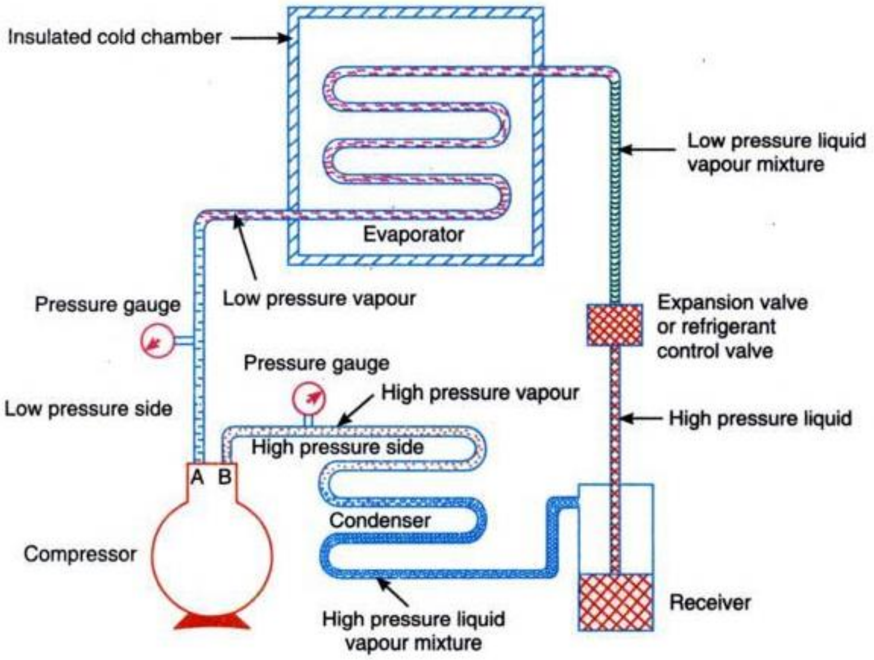


Fig.17. Simple Vapour Compression Refrigeration System (VCR)

Receiver

The condensed liquid refrigerant from the condenser is stored in a vessel known as receiver from where it is supplied to the evaporator through the expansion valve or refrigerant control valve.

Expansion Valve

It is also called throttle valve or refrigerant control valve. The function of the expansion valve is to allow the liquid refrigerant under high pressure and temperature to pass at a controlled rate after reducing its pressure and temperature. Some of the liquid refrigerant evaporates as it passes through the expansion valve, but the greater portion is vaporized in the evaporator at the low pressure and temperature

Evaporator

An evaporator consists of coils of pipe in which the liquid-vapour. Refrigerant at low pressure and temperature is evaporated and changed into vapour refrigerant at low pressure and temperature. In evaporating, the liquid vapour refrigerant absorbs its latent heat of vaporization from the medium (air, water or brine) which is to be cooled.

The Simple Vapor Compression Refrigeration Cycle

The vapor compression refrigeration cycle has four components: evaporator, compressor, condenser, and expansion (or throttle) valve. The most widely used refrigeration cycle is the *vapor-compression refrigeration cycle.* In an ideal or simple vapor-compression refrigeration cycle, the refrigerant enters the compressor as a saturated vapor and is cooled to the saturated liquid state in the condenser. It is then throttled to the evaporator pressure and vaporizes as it absorbs heat from the refrigerated space.

The ideal vapor compression cycle consists of four processes.

Ideal Vapor-Compression Refrigeration Cycle

Process	Description
1-2	Isentropic compression
2-3	Constant pressure heat rejection in the condenser
3-4	Throttling in an expansion valve
4-1	Constant pressure heat addition in the evaporator

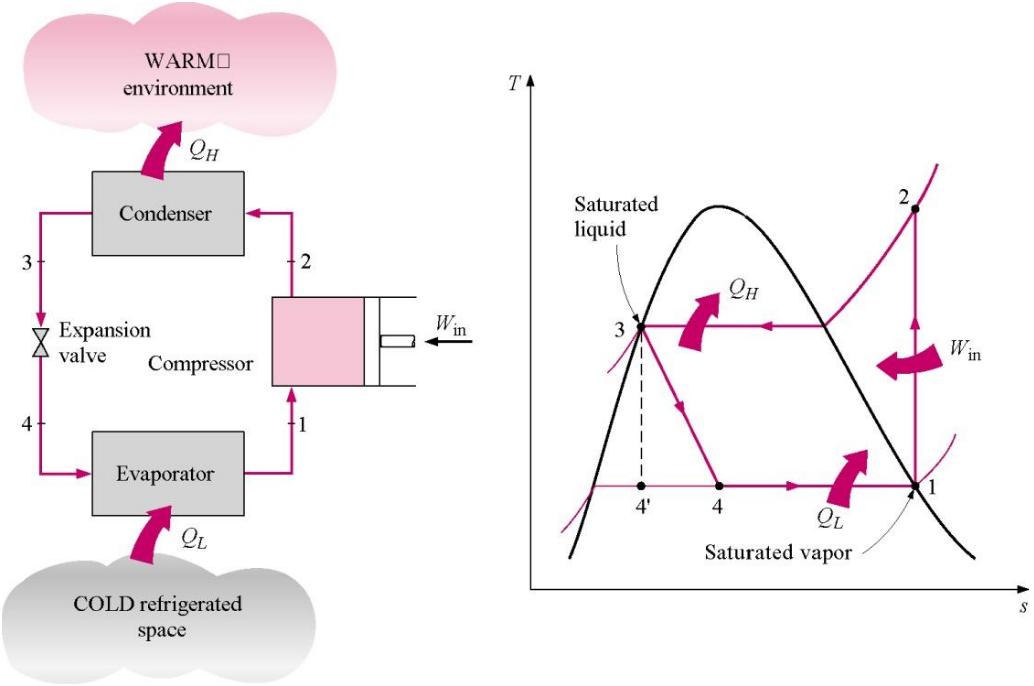


Fig.18 Simple Vapor Compression Refrigeration Cycle

The P-h diagram is another convenient diagram often used to illustrate the refrigeration cycle.

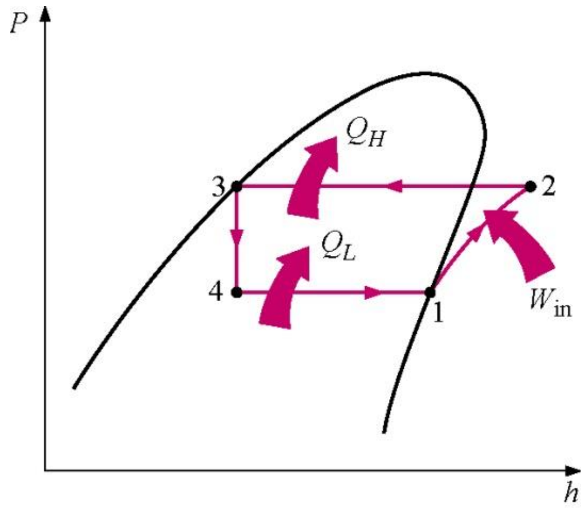


Fig. 19. P-h diagram of Simple Vapor Compression Refrigeration Cycle

The ordinary household refrigerator is a good example of the application of this cycle.

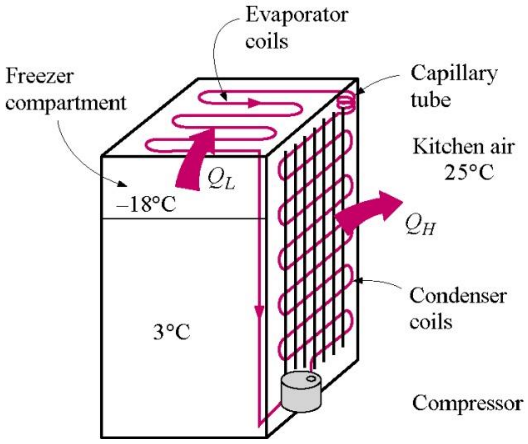


Fig. 20. Ordinary household refrigerator

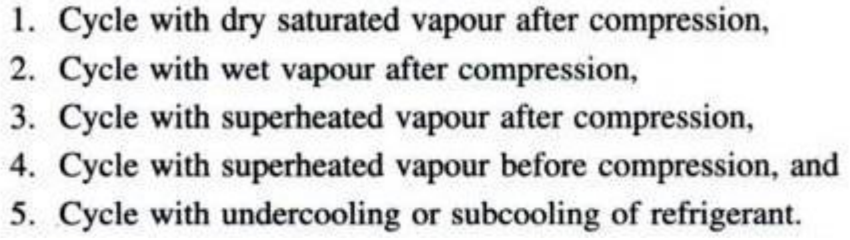
Results of First and Second Law Analysis for Steady-Flow

Component	Process	First Law Result
Compressor	s = Const.	$\dot{W}_{in} = \dot{m}(h_2 - h_1)$
Condenser	P = Const.	$\dot{Q}_H = \dot{m}(h_2 - h_3)$
Throttle Valve	Δs > 0 $\dot{W}_{net} = 0$ $\dot{Q}_{net} = 0$	$h_4 = h_3$
Evaporator	P = Const.	$\dot{Q}_L = \dot{m}(h_1 - h_4)$

$$COP_R = \frac{\dot{Q}_L}{\dot{W}_{net,in}} = \frac{h_1 - h_4}{h_2 - h_1}$$

$$COP_{HP} = \frac{\dot{Q}_H}{\dot{W}_{net,in}} = \frac{h_2 - h_3}{h_2 - h_1}$$

Methods to enhance the COP of simple vapour compression refrigeration system



1. Cycle with dry saturated vapour after compression,
2. Cycle with wet vapour after compression,
3. Cycle with superheated vapour after compression,
4. Cycle with superheated vapour before compression, and
5. Cycle with undercooling or subcooling of refrigerant.

VAPOUR ABSORPTION SYSTEM

Introduction

In a vapour absorption system the refrigerant is absorbed on leaving the evaporator, the absorbing medium being a solid or liquid. In order that the sequence of events should be continuous it is necessary for the refrigerant to be separated from the absorbent and subsequently condensed before being returned to the evaporator. The separation is accomplished by the application of direct heat in a ‘generator’. The solubility of the refrigerant and absorbent must be suitable and the plant which uses ammonia as the refrigerant and water as absorbent will be described.

Simple Vapour Absorption System

for a simple absorption system. The solubility of ammonia in water at low temperatures and pressures is higher than it is at higher temperatures and pressures. The ammonia vapour leaving the evaporator at point 2 is readily absorbed in the low temperature hot solution in the absorber. This process is accompanied by the rejection of heat. The ammonia in water solution is pumped to the higher pressure and is heated in the generator. Due to reduced solubility of ammonia in water at the higher pressure and temperature, the vapour is removed from the solution. The vapour then passes to the condenser and the weakened ammonia in water solution is returned to the absorber.

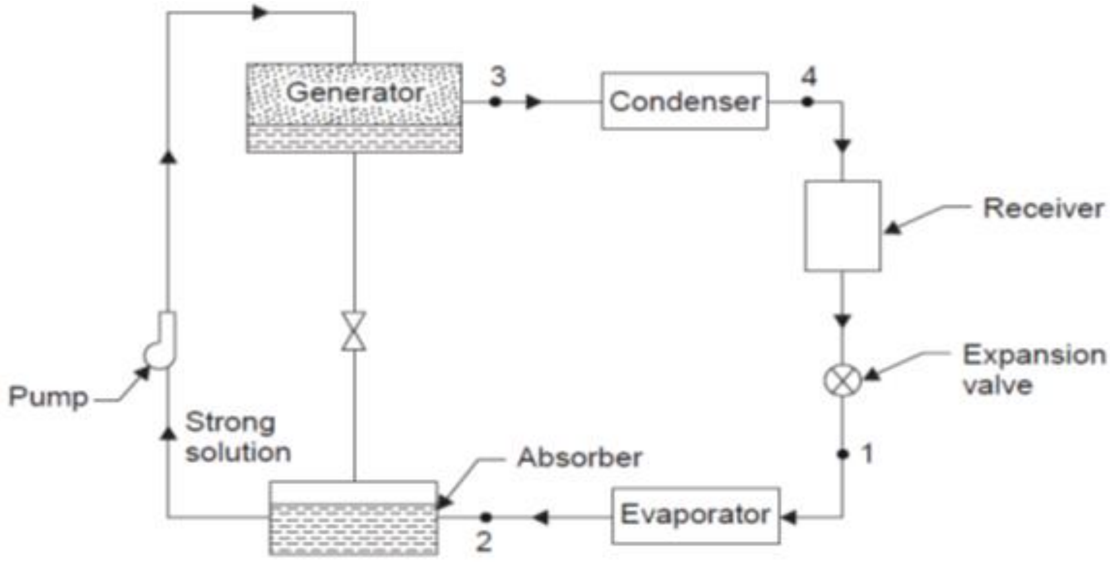
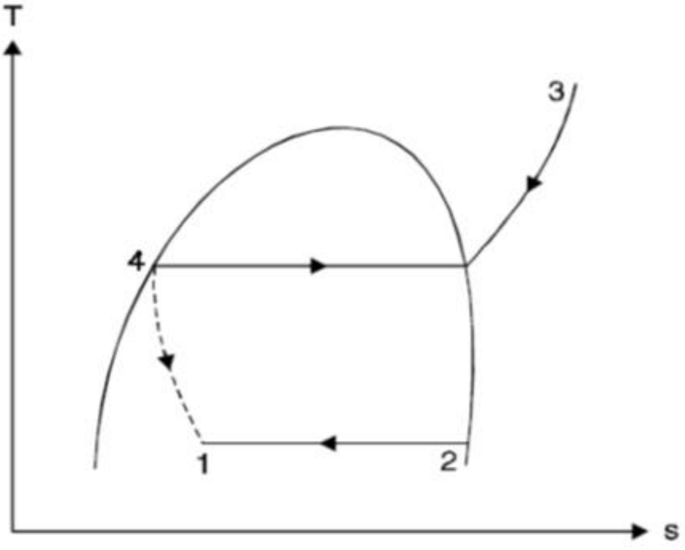


Fig 21. Simple Vapour Absorption System



Simple vapour absorption system—*T-s* diagram.

Fig 22. T-s diagram of Simple Vapour Absorption System

In this system the work done on compression is less than in vapour compression cycle (since pumping a liquid requires much less work than compressing a vapour between the same pressures) but a heat input to the generator is required. The heat may be supplied by any convenient form e.g. steam or gas heating.

Practical Vapour Absorption System

Although a simple vapour absorption system can provide refrigeration yet its operating efficiency is low. The following accessories are fitted to make the system more practical and improve the performance and working of the plant.

1. Heat exchanger. 2. Analyser. 3. Rectifier.

1. **Heat exchanger.** A heat exchanger is located between the generator and the absorber.

The strong solution which is pumped from the absorber to the generator must be heated ; and the weak solution from the generator to the absorber must be cooled. This is accomplished by a heat exchanger and consequently cost of heating the generator and cost of cooling the absorber are reduced.

2. **Analyser.** An analyser consists of a series of trays mounted above the generator. Its main function is to remove partly some of the unwanted water particles associated with ammonia vapour going to condenser. If these water vapours are permitted to enter condenser they may enter the expansion valve and freeze ; as a result the pipe line may get choked.

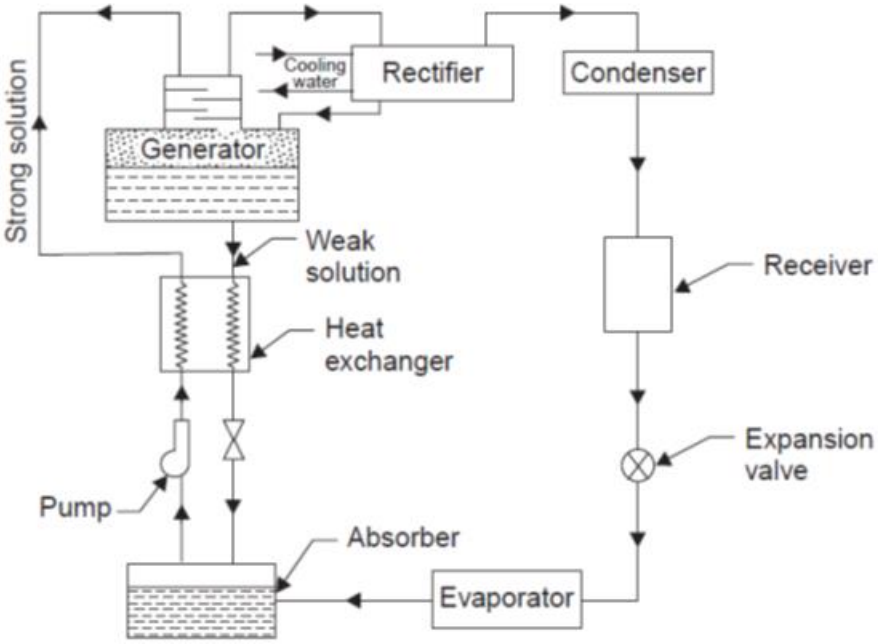


Fig 23. Practical Vapour Absorption System

3. **Rectifier.** A rectifier is a water-cooled heat exchanger which condenses water vapour and some ammonia and sends back to the generator. Thus final reduction or elemination of the percentage of water vapour takes place in a rectifier. The co-efficient of performance (C.O.P.) of this system is given by :

$$\text{C.O.P.} = \frac{\text{Heat extracted from the evaporator}}{\text{Heat supplied in the generator + Work done by the liquid pump}} .$$

PSYCHROMETRY AND AIR – CONDITIONING

Psychrometric properties, Use of psychrometric chart, Psychrometric process – Sensible heat exchange process, Latent heat exchange process, Adiabatic mixing, Evaporative cooling, Property calculations of air-vapour mixtures.

Principles of air-conditioning, Types of air conditioning systems – summer, winter, year round air conditioners, Concept of RSHF, GSHF, ESHF, Simple problems.

5.17.1 CONCEPT OF PSYCHROMETRY AND PSYCHROMETRICS

Air comprises of fixed gases principally, nitrogen and oxygen with an admixture of water vapour in varying amounts. In atmospheric air water is always present and its relative weight averages less than 1% of the weight of atmospheric air in temperate climates and less than 3% by weight under the most extreme natural climatic conditions, it is nevertheless one of most important factors in human comfort and has significant effects on many materials. Its effect on human activities is in fact altogether disproportionate to its relative weights. *The art of measuring* the *moisture content of air is termed* “**psychrometry**”. *The science which investigates the thermal properties of moist air, considers the measurement and control of the moisture content of air, and studies the effect of atmospheric moisture on material and human comfort may properly be termed* “**psychrometrics’**’.

DEFINITIONS

Some of the more important definitions are given below :

1. **Dry air.** The international joint committee on Psychrometric Data has adopted the following exact composition of air expressed in mole fractions (Volumetric) Oxygen 0.2095, Nitrogen 0.7809, Argon 0.0093, Carbon dioxide 0.0003. Traces of rare gases are neglected. Molecular weight of air for all air conditioning calculations will be taken as 28.97. Hence the gas constant, *Rair* = 0.287 kJ/kg K Dry air is never found in practice. Air always contains some moisture. Hence the common designation “air” *usually means moist* air. The term ‘*dry air*’ is used to indicate the *water free contents* of air having any degree of moisture.

2. **Saturated air.** Moist air is said to be saturated when its condition is such that it can co-exist in natural equilibrium with an associated condensed moisture phase presenting a flat surface to it. For *a given temperature, a given quantity of air can be saturated with a fixed quantity of moisture.* At higher temperatures, it requires a larger quantity of moisture to saturate it. At saturation, vapour pressure of moisture in air corresponds to the saturation pressure given in steam tables corresponding to the given temperature of air.

3. **Dry-bulb temperature (DBT).** It *is the temperature of air as registered by an ordinary thermometer* (*tdb*).

4. **Wet-bulb temperature (WBT).** *It is the temperature registered by a thermometer when the bulb is covered by a wetted wick and is exposed to a current of rapidly moving air* (*twb*).

5. **Adiabatic saturation temperature.** *It is the temperature at which the water or ice can saturate air by evaporating adiabatically into it*. It is numerically equivalent to the measured wet bulb temperature (as corrected, if necessary for radiation and conduction) (*twb*).

6. **Wet bulb depression.** It *is the difference between dry-bulb and wet bulb temperatures* (*tdb – twb*).

7. **Dew point temperature (DPT).** *It is the temperature to which air must be cooled at constant pressure in order to cause condensation of any of its water vapour*. It is equal to steam table saturation temperature corresponding to the actual partial pressure of water vapour in the air (*tdp*).

8. **Dew point depression.** *It* is *the difference between the dry bulb and dew point temperatures* (*tdb – tdp*).

9. **Specific humidity (Humidity ratio).** *It is the ratio of the mass of water vapour per unit mass of dry air in the mixture of vapour and air,* it is generally expressed as grams of water per kg of dry air. For a given barometric pressure it is a function of dew point temperature alone.

10. **Relative humidity (RH), (φ).** *It is the ratio of the partial pressure of water vapour in the mixture to the saturated partial pressure at the dry bulb temperature, expressed as percentage.*

11. **Sensible heat.** *It is the heat that changes the temperature of a substance when added*

to or abstracted from it.

12. **Latent heat.** *It is the heat that does not affect the temperature but changes the state*

of substance when added to or abstracted from it.

13. **Enthalpy.** *It is the combination energy which represents the sum of internal and flow energy in a steady flow process*. It is determined from an arbitrary datum point for the air mixture and is expressed as kJ per kg of dry air (*h*).

Note. When air is saturated DBT, WBT, DPT are *equal*.

TEXT / REFERENCE BOOKS

1. Nag P.K., “Engineering Thermodynamics”, Tata McGraw Hill Education, 2009.
2. Yunus A. Cengel, Michael A. Boles, “Thermodynamics: An Engineering Approach”, McGraw Hill Education, 2014.
3. Rajput R.K., “Engineering Thermodynamics”, Laxmi Publications, 2010.
4. Khurmi R.S., Gupta J.K, “Thermal Engineering”, S Chand, 2006.
5. P.L. Ballaney,”Thermal Engineering”, Khanna Publisher, 2005.

SATHYABAMA
INSTITUTE OF SCIENCE AND TECHNOLOGY
(DEEMED TO BE UNIVERSITY)
Accredited "A" Grade by NAAC | 12B Status by UGC | Approved by AICTE
www.sathyabama.ac.in

SCHOOL OF MECHANICAL ENGINEERING

DEPARTMENT OF AERONAUTICAL

UNIT – II – Aero Engineering Thermodynamics – SAEA1301

II. SECOND LAW OF THERMODYNAMICS AND ITS APPLICATIONS

Limitations of first law of thermodynamics and introduction to second law

It has been observed that energy can flow from a system in the form of heat or work. The first law of thermodynamics sets no limit to the amount of the total energy of a system which can be caused to flow out as work. A limit is imposed, however, as a result of the principle enunciated in the second law of thermodynamics which states that heat will flow naturally from one energy reservoir to another at a lower temperature, but not in opposite direction without assistance. This is very important because a heat engine operates between two energy reservoirs at different temperatures. Further the first law of thermodynamics establishes equivalence between the quantity of heat used and the mechanical work but does not specify the conditions under which conversion of heat into work is possible, neither the direction in which heat transfer can take place. This gap has been bridged by the second law of thermodynamics.

Performance of heat engines and reversed heat engines

A heat engine is used to produce the maximum work transfer from a given positive heat transfer. The measure of success is called the thermal efficiency of the engine and is defined by the ratio :

$$\text{Thermal efficiency, } \eta_{th} = \frac{W}{Q_1}$$

where, W = Net work transfer from the engine, and Q_1 = Heat transfer to engine.

For a reversed heat engine acting as a refrigerator when the purpose is to achieve the maximum heat transfer from the cold reservoir, the measure of success is called the co-efficient of performance (C.O.P.). It is defined by the ratio :

$$\text{Co-efficient of performance, } (\text{C.O.P.})_{ref.} = \frac{Q_2}{W}$$

where, Q_1 = Heat transfer to hot reservoir, and W = Net work transfer to the heat pump.

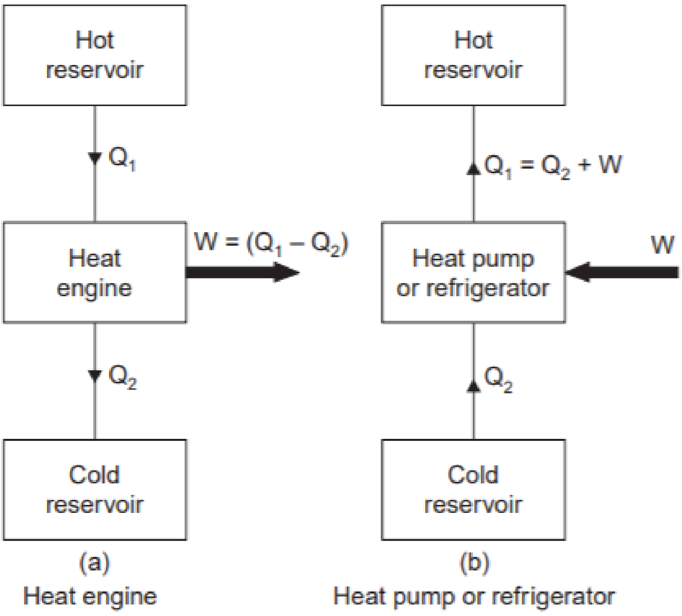


Fig 1. heat engines and reversed heat engines

In all the above three cases application of the first law gives the relation $Q_1 - Q_2 = W$, and this can be used to rewrite the expressions for thermal efficiency and co-efficient of performance solely in terms of the heat transfers.

$$\eta_{th} = \frac{Q_1 - Q_2}{Q_1}$$

$$(C.O.P.)_{ref} = \frac{Q_2}{Q_1 - Q_2}$$

$$(C.O.P.)_{heat\ pump} = \frac{Q_1}{Q_1 - Q_2}$$

It may be seen that η_{th} is always less than unity and $(C.O.P.)_{heat\ pump}$ is always greater than unity.

Reversible processes:

A reversible process should fulfill the following conditions :

1. The process should not involve friction of any kind.

2. Heat transfer should not take place with finite temperature difference.

3. The energy transfer as heat and work during the forward process should be identically equal to energy transfer as heat and work during the reversal of the process.

4. There should be no free or unrestricted expansion.

5. There should be no mixing of the fluids. 6. The process must proceed in a series of equilibrium states.

Some examples of ideal reversible processes are :

(i) Frictionless adiabatic expansion or compression ;
(ii) Frictionless isothermal expansion or compression ;
(iii) Condensation and boiling of liquids.

Some examples of irreversible processes are :

(i) Combustion process;
(ii) Mixing of two fluids ;
(iii) All processes involving friction;
(iv) Flow of electric current through a resistance ;
(v) Heat flow from a higher temperature to lower temperature.

Reversible processes are preferred because the devices which produce work such as engines and turbines, reversible process of the working fluid delivers more work than the corresponding irreversible processes. Also in case of fans, compressors, refrigerators and pumps less power input is required when reversible processes are used in place of corresponding irreversible ones. In thermodynamic analysis concept of reversibility, though hypothetical, is very important because a reversible process is the most efficient process. Only reversible processes can be truely represented on property diagrams. Thermodynamic reversibility can only be approached but can never be achieved. Thus the main task of the engineer is to design the system which will evolve approximate reversible processes.

Statements of second law of thermodynamics:

The second law of thermodynamics has been enunciated meticulously by Clausius, Kelvin and Planck in slightly different words although both statements are basically identical. Each statement is based on an irreversible process. The first considers transformation of heat between two thermal reservoirs while the second considers the transformation of heat into work.

Clausius Statement:

“It is impossible for a self-acting machine working in a cyclic process unaided by any external agency, to convey heat from a body at a lower temperature to a body at a higher temperature”

Kelvin-Planck Statement:

"It is impossible to construct an engine, which while operating in a cycle produces no other effect except to extract heat from a single reservoir and do equivalent amount of work".

Equivalence of Clausius Statement to the Kelvin-Planck Statement:

Consider a higher temperature reservoir T_1 and low temperature reservoir T_2. Fig. shows a heat pump which requires no work and transfers an amount of Q_2 from a low temperature to a higher temperature reservoir (in violation of the Clausius statement). Let an amount of heat Q_1 (greater than Q_2) be transferred from high temperature reservoir to heat engine which devolops a net work, $W = Q_1 - Q_2$ and rejects Q_2 to the low temperature reservoir. Since there is no heat interaction with the low temperature, it can be eliminated. The combined system of the heat engine and heat pump acts then like a heat engine exchanging heat with a single reservoir, which is the violation of the Kelvin-Planck statement.

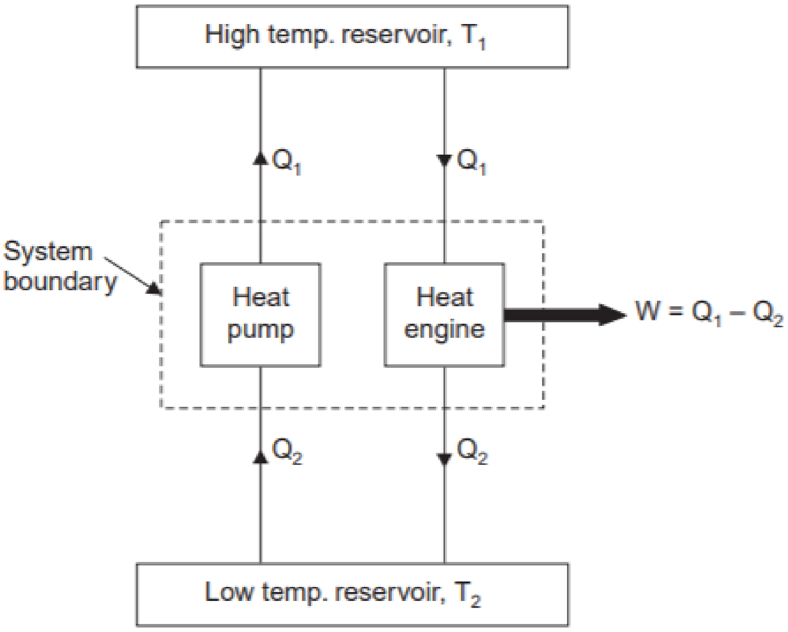


Fig.2. Equivalence of Clausius statement to Kelvin-Planck statement

Clausius inequality:

When a reversible engine uses more than two reservoirs the third or higher numbered reservoirs will not be equal in temperature to the original two. Consideration of expression for efficiency of the engine indicates that for maximum efficiency, all the heat transfer should take place at maximum or minimum reservoir temperatures. Any intermediate reservoir used will, therefore, lower the efficiency of the heat engine. Practical engine cycles often involve continuous changes of temperature during heat transfer. A relationship among processes in which these sort of changes occur is necessary. The ideal approach to a cycle in which temperature continually changes is to consider the system to be in communication with a large number of reservoirs in procession. Each reservoir is considered to have a temperature differing by a small amount from the previous one. In such a model it is possible to imagine that each reservoir is replaced by a reversible heat engine in communication with standard reservoirs at same temperature T_0. Fig. shows one example to this substitution.

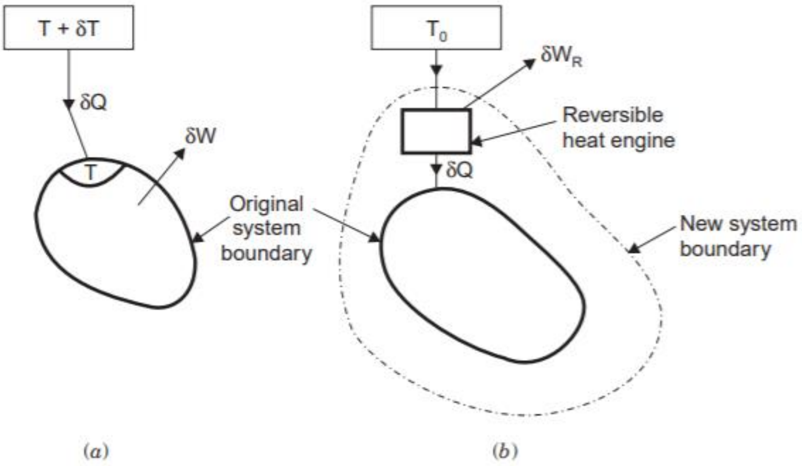


Fig. 3. Clausius inequality

The system to which the heat transfer is effected is neither concerned with the source of energy it receives nor with the method of transfer, save that it must be reversible. Associated with the small heat transfer dQ to the original system is a small work transfer dW and for this system the first law gives,

$$\sum_{\text{cycle}} (\delta Q - \delta W) = 0$$

Now consider the engine replacing the reservoirs and apply the second law to the new system in Fig. (b). If the new system is not a perpetual motion machine of second kind, no positive work transfer is possible with a single reservoir.

Therefore, $$\sum_{\text{cycle}} (\delta W - \delta W_R) \leq 0$$

But by the definition of thermodynamic temperature in equation

$$\frac{\delta W_R}{\delta Q} = \frac{\delta Q_0 - \delta Q}{\delta Q} = \frac{T_0 - T}{T}$$

$$T_0 \sum_{\text{cycle}} \left(\frac{\delta Q}{T}\right) \leq 0 \text{ but } T_0 \neq 0 \text{ and therefore ;}$$

$$\sum_{\text{cycle}} \left(\frac{\delta Q}{T}\right) \leq 0$$

This is known as Clausius inequality.

Carnot cycle:

The cycle was first suggested by a French engineer Sadi Carnot in 1824 which works on reversible cycle and is known as Carnot cycle. Any fluid may be used to operate the Carnot cycle which is performed in an engine cylinder the head of which is supposed alternatively to be perfect conductor or a perfect insulator of a heat. Heat is caused to flow into the cylinder by the application of high temperature energy source to the cylinder head during expansion, and to flow from the cylinder by the application of a lower temperature energy source to the head during compression.

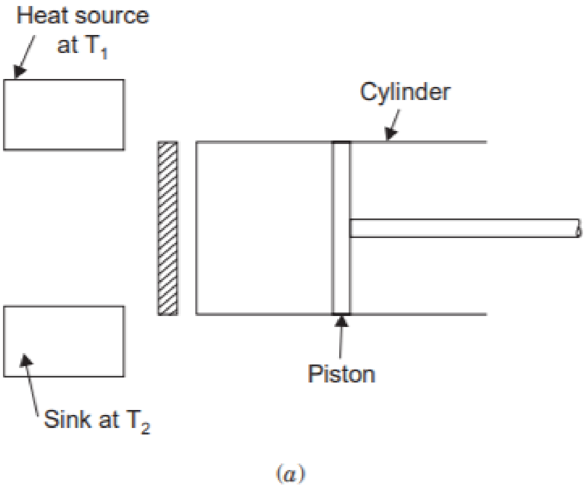
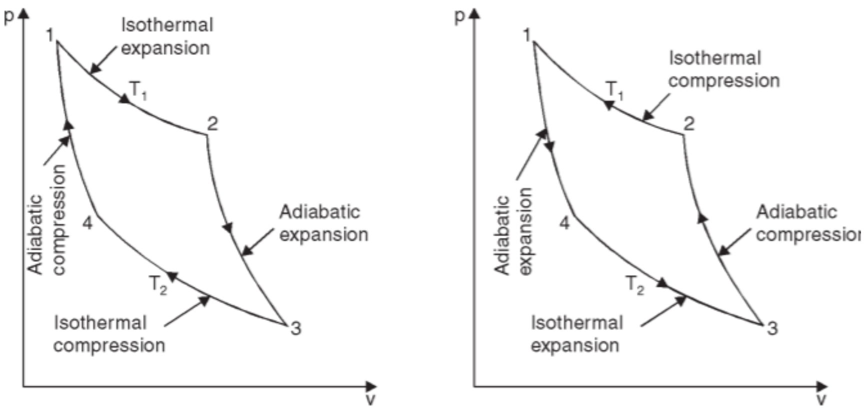


Fig. 4. Carnot Engine



(*b*)
Carnot engine cycle

(*c*)
Carnot heat pump cycle

Fig. 5. Carnot Cycle

The assumptions made for describing the working of the Carnot engine are as follows :

(i) The piston moving in a cylinder does not develop any friction during motion.

(ii) The walls of piston and cylinder are considered as perfect insulators of heat.

(iii)The cylinder head is so arranged that it can be a perfect heat conductor or perfect heat insulator.

(iv) The transfer of heat does not affect the temperature of source or sink.

(v) Working medium is a perfect gas and has constant specific heat.

(vi) Compression and expansion are reversible.

Following are the four stages of Carnot cycle :

Stage 1. (Process 1-2). Hot energy source is applied. Heat Q_1 is taken in whilst the fluid expands isothermally and reversibly at constant high temperature T_1.

Stage 2. (Process 2-3). The cylinder becomes a perfect insulator so that no heat flow takes place. The fluid expands adiabatically and reversibly whilst temperature falls from T_1 to T_2.

Stage 3. (Process 3-4). Cold energy source is applied. Heat Q_2 flows from the fluid whilst it is compressed isothermally and reversibly at constant lower temperature T_2.

Stage 4. (Process 4-1). Cylinder head becomes a perfect insulator so that no heat flow occurs. The compression is continued adiabatically and reversibly during which temperature is raised from T_2 to T_1. The work delivered from the system during the cycle is represented by the enclosed area of the cycle. Again for a closed cycle, according to first law of the thermodynamics the work obtained is equal to the difference between the heat supplied by the source (Q_1) and the heat rejected to the sink (Q_2).

$$\therefore \qquad W = Q_1 - Q_2$$

$$\text{Also, thermal efficiency, } \eta_{th} = \frac{\text{Work done}}{\text{Heat supplied by the source}} = \frac{Q_1 - Q_2}{Q_1}$$

$$= 1 - \frac{Q_2}{Q_1} \left(= 1 - \frac{T_2}{T_1} \right) \qquad \begin{bmatrix} \because \quad Q_1 = m\,c_p\,T_1 \\ Q_2 = m\,c_p\,T_2 \\ \text{where, } m = \text{mass of fluid.} \end{bmatrix}$$

Such an engine since it consists entirely of reversible processes, can operate in the reverse direction so that it follows the cycle shown in Fig. (b) and operates as a heat pump. Q2 is being taken in at the lower temperature T2 during the isothermal expansion (process 4-3) and heat Q1 is being rejected at the upper temperature T1 (process 2-1). Work W will be needed to drive the pump. Again, the enclosed area represents this work which is exactly equal to that flowing from it when used as engine.

The Carnot cycle cannot be performed in practice because of the following reasons :

1. It is impossible to perform a frictionless process.
2. It is impossible to transfer the heat without temperature potential.
3. Isothermal process can be achieved only if the piston moves very slowly to allow heat transfer so that the temperature remains contant.

Adiabatic process can be achieved only if the piston moves as fast as possible so that the heat transfer is negligible due to very short time available. The isothermal and adiabatic processes take place during the same stroke therefore the piston has to move very slowly for part of the stroke and it has to move very fast during remaining stroke. This variation of motion of the piston during the same stroke is not possible.

Carnot’s theorem:

“It states that of all engines operating between a given constant temperature source and a given constant temperature sink, none has a higher efficiency than a reversible engine”.

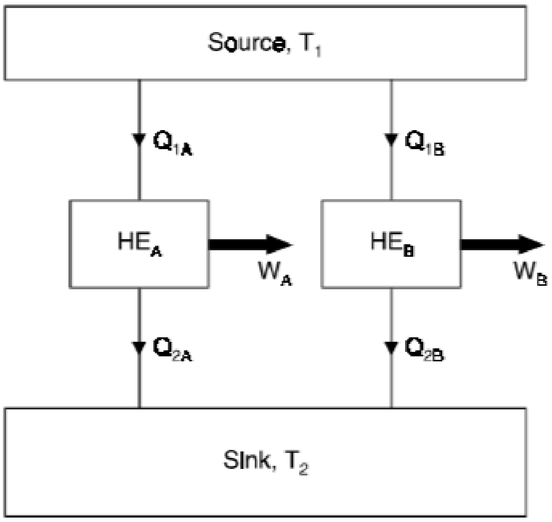


Fig.6. Two cyclic heat engines HE_A and HE_B operating between the same source and sink, of which HE_B is reversible.

HE_A and HE_B are the two engines operating between the given source at temperature T_1 and the given sink at temperature T_2.

Let HE_A be any heat engine and HE_B be any reversible heat engine. We have to prove that efficiency of HE_B is more than that of HE_A. Let us assume that $\eta_A > \eta_B$. Let the rates of working of the engines be such that,

$$Q_{1A} = Q_{1B} = Q_1$$
$$\eta_A > \eta_B$$
$$\frac{W_A}{Q_{1A}} > \frac{W_B}{Q_{1B}}$$
$$W_A > W_B$$

Now, let HE_B be reversed. Since HE_B is a reversible heat engine, the magnitudes of heat and work transfer quantities will remain the same, but their directions will be reversed, as shown in Fig. Since $W_A > W_B$, some part of W_A (equal to W_B) may be fed to drive the reversed heat engine $\exists H_B$. Since $Q_{1A} = Q_{1B} = Q_1$, the heat discharged by $\exists H_B$ may be supplied to HE_A. The source may, therefore, be eliminated. The net result is that HE_A and $\exists H_B$ together constitute a heat engine which, operating in a cycle produces network $W_A – W_B$ while exchanging heat with a single reservoir at T_2. This violates the Kelvin-Planck statement of the second law. Hence the assumption that $\eta_A > \eta_B$ is wrong.

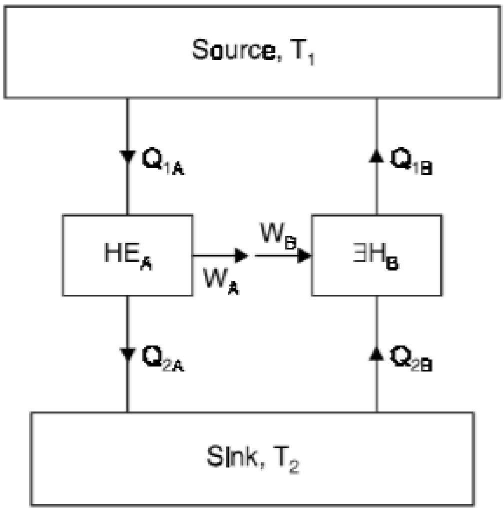


Fig.7. HEB is reversed

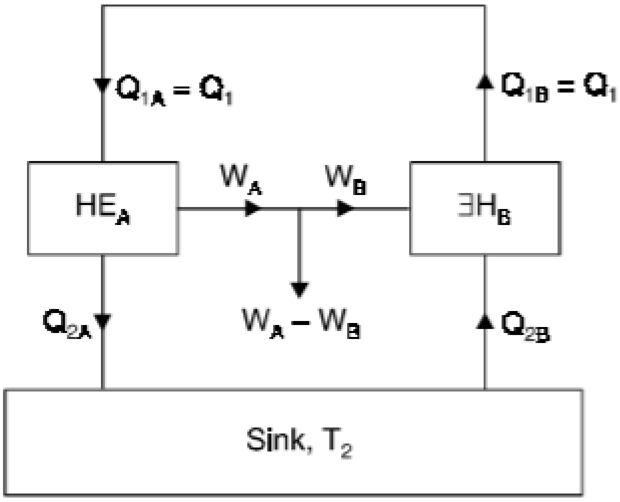


Fig.8. HEA and ƎHB together violate the Kelvin-Planck statement.

Therefore,

$$\eta_B \geq \eta_A.$$

Corollary of carnot's theorem:

''The efficiency of all reversible heat engines operating between the same temperature levels is the same".

Let both the heat engines HE_A and HE_B be reversible. Let us assume $\eta_A > \eta_B$. if HE_B is reversed to run say, as a heat pump using some part of the work output (W_A) of engine HE_A, we see that the combined system of heat pump HE_B and engine HE_A, becomes a PMM2. So η_A cannot be greater than η_B. Similary, if we assume $\eta_B > \eta_A$ and reverse the engine HE_A, we observe that η_B cannot be greater than η_A ∴ $\eta_A = \eta_B$. Since the efficiencies of all reversible engines operating between the same heat reservoirs are the same, the efficiency of a reversible engine is independent of the nature or amount of the working substance undergoing the cycle.

Efficiency of the reversible heat engine:

The efficiency of a reversible heat engine in which heat is received solely at T_1 is found to be,

$$\eta_{rev.} = \eta_{max} = 1 - \left(\frac{Q_2}{Q_1}\right)_{rev.} = 1 - \frac{T_2}{T_1}$$

$$\eta_{rev.} = \frac{T_1 - T_2}{T_1}$$

From the above expression, it may be noted that as T_2 decreases and T_1 increases, the efficiency of the reversible cycle increases.

Since η is always less than unity, T_2 is always greater than zero and + ve. The C.O.P. of a refrigerator is given by,

$$(\text{C.O.P.})_{ref.} = \frac{Q_2}{Q_1 - Q_2} = \frac{1}{\frac{Q_1}{Q_2} - 1}$$

For a *reversible refrigerator*, using

$$\frac{Q_1}{Q_2} = \frac{T_1}{T_2}$$

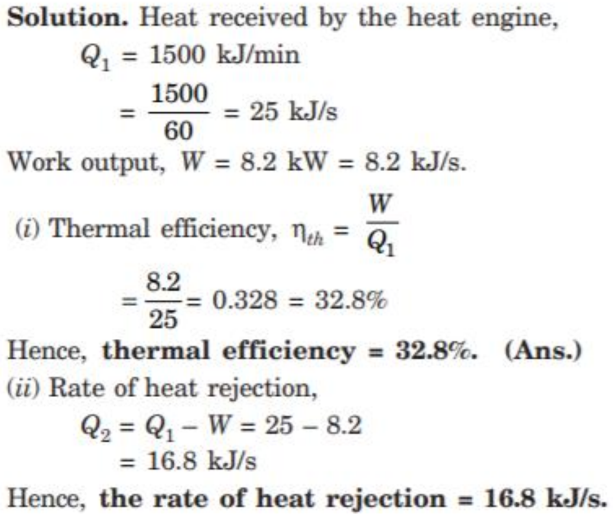
$$(\text{C.O.P.})_{rev.} = \frac{1}{\frac{T_1}{T_2} - 1}$$

$$\therefore \qquad [(\text{C.O.P.})_{ref.}]_{rev.} = \frac{T_2}{T_1 - T_2}$$

Similarly, for a *reversible heat pump*

$$[(\text{C.O.P.})_{heat\ pump}]_{rev.} = \frac{T_1}{T_1 - T_2}$$

Problem 1: A heat engine receives heat at the rate of 1500 kJ/min and gives an output of 8.2 kW. Determine : (i) The thermal efficiency ; (ii) The rate of heat rejection.



Solution. Heat received by the heat engine,

$$Q_1 = 1500 \text{ kJ/min}$$

$$= \frac{1500}{60} = 25 \text{ kJ/s}$$

Work output, $W = 8.2 \text{ kW} = 8.2 \text{ kJ/s}$.

(*i*) Thermal efficiency, $\eta_{th} = \frac{W}{Q_1}$

$$= \frac{8.2}{25} = 0.328 = 32.8\%$$

Hence, **thermal efficiency = 32.8%. (Ans.)**

(*ii*) Rate of heat rejection,

$$Q_2 = Q_1 - W = 25 - 8.2$$

$$= 16.8 \text{ kJ/s}$$

Hence, **the rate of heat rejection = 16.8 kJ/s.**

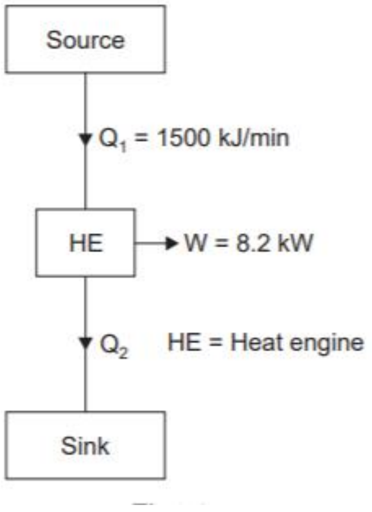
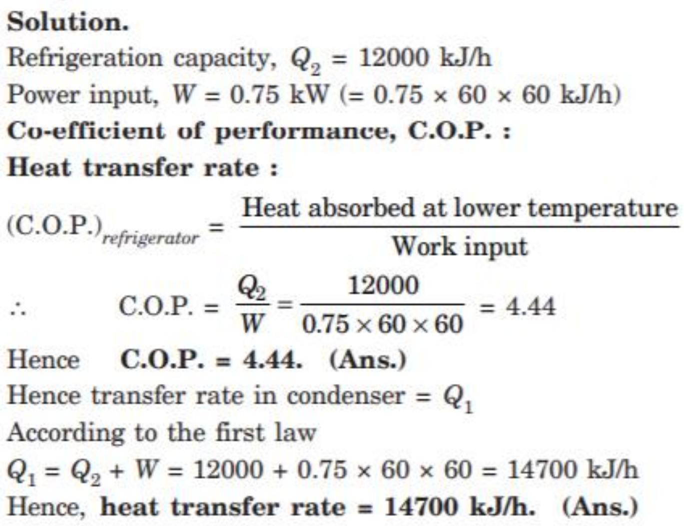


Fig 8.1

Problem 2: Find the co-efficient of performance and heat transfer rate in the condenser of a refrigerator in kJ/h which has a refrigeration capacity of 12000 kJ/h when power input is 0.75 kW.

Solution:



Solution.

Refrigeration capacity, $Q_2 = 12000 \text{ kJ/h}$

Power input, $W = 0.75 \text{ kW } (= 0.75 \times 60 \times 60 \text{ kJ/h})$

Co-efficient of performance, C.O.P. :

Heat transfer rate :

$$(\text{C.O.P.})_{refrigerator} = \frac{\text{Heat absorbed at lower temperature}}{\text{Work input}}$$

$$\therefore \quad \text{C.O.P.} = \frac{Q_2}{W} = \frac{12000}{0.75 \times 60 \times 60} = 4.44$$

Hence **C.O.P. = 4.44. (Ans.)**

Hence transfer rate in condenser = Q_1

According to the first law

$$Q_1 = Q_2 + W = 12000 + 0.75 \times 60 \times 60 = 14700 \text{ kJ/h}$$

Hence, **heat transfer rate = 14700 kJ/h. (Ans.)**

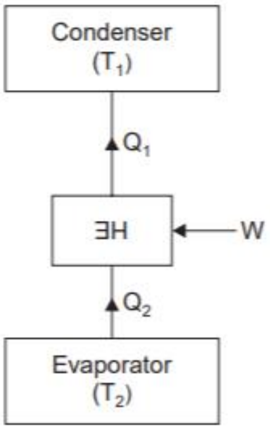


Fig. 8.2

Problem 3: A domestic food refrigerator maintains a temperature of – 12°C. The ambient air temperature is 35°C. If heat leaks into the freezer at the continuous rate of 2 kJ/s determine the least power necessary to pump this heat out continuously.

Solution. Freezer temperature,

$$T_2 = -12 + 273 = 261 \text{ K}$$

Ambient air temperature,

$$T_1 = 35 + 273 = 308 \text{ K}$$

Rate of heat leakage into the freezer = 2 kJ/s

Least power required to pump the heat :

The refrigerator cycle removes heat from the freezer at the same rate at which heat leaks into it (Fig. 5.12).

For minimum power requirement

$$\frac{Q_2}{T_2} = \frac{Q_1}{T_1}$$

$$\therefore \qquad Q_1 = \frac{Q_2}{T_2} \times T_1 = \frac{2}{261} \times 308 = 2.36 \text{ kJ/s}$$

$$\therefore \qquad W = Q_1 - Q_2$$

$$= 2.36 - 2 = 0.36 \text{ kJ/s} = 0.36 \text{ kW}$$

Hence, *least power required to pump the heat continuously*

= **0.36 kW. (Ans.)**

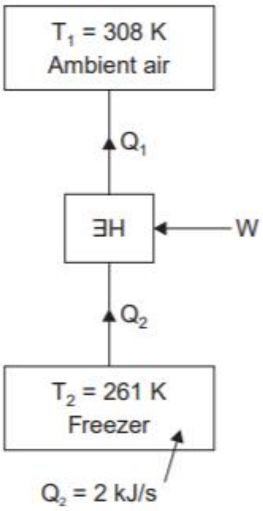


Fig. 8.3

Problem 4: An inventor claims that his engine has the following specifications :

Temperature limits ...... 750°C and 25°C

Power developed ...... 75 kW

Fuel burned per hour ...... 3.9 kg

Heating value of the fuel ...... 74500 kJ/kg

State whether his claim is valid or not

Solution. Temperature of source, $T_1 = 750 + 273 = 1023$ K

Temperature of sink, $T_2 = 25 + 273 = 298$ K

We know that the thermal efficiency of Carnot cycle is the maximum between the specified temperature limits.

Now, $$\eta_{carnot} = 1 - \frac{T_2}{T_1} = 1 - \frac{298}{1023} = 0.7086 \quad \text{or} \quad 70.86\%$$

The actual thermal efficiency claimed,

$$\eta_{thermal} = \frac{\text{Work done}}{\text{Heat supplied}} = \frac{75 \times 1000 \times 60 \times 60}{3.9 \times 74500 \times 1000} = 0.9292 \quad \text{or} \quad 92.92\%.$$

Since $\eta_{thermal} > \eta_{carnot}$, therefore claim of the inventor is **not valid** (or possible). **(Ans.)**

Problem 5: A cyclic heat engine operates between a source temperature of 1000°C and a sink temperature of 40°C. Find the least rate of heat rejection per kW net output of the engine ?

Solution. Temperature of source,

$$T_1 = 1000 + 273 = 1273 \text{ K}$$

Temperature of sink,

$$T_2 = 40 + 273 = 313 \text{ K}$$

Least rate of heat rejection per kW net output :

For a reversible heat engine, the rate of heat rejection will be minimum (Fig. 5.13)

$$\eta_{max} = \eta_{rev.} = 1 - \frac{T_2}{T_1}$$

$$= 1 - \frac{313}{1273} = 0.754$$

Now $\frac{W_{net}}{Q_1} = \eta_{max} = 0.754$

$\therefore$ $Q_1 = \frac{W_{net}}{0.754} = \frac{1}{0.754} = 1.326 \text{ kW}$

Now $Q_2 = Q_1 - W_{net} = 1.326 - 1 = 0.326 \text{ kW}$

Hence, **the least rate of heat rejection = 0.326 kW. (Ans.)**

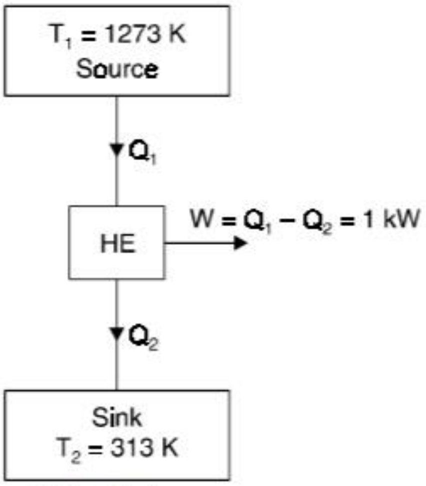


Fig. 8.4

Problem 6: A reversible heat engine operates between two reservoirs at temperatures 700°C and 50°C. The engine drives a reversible refrigerator which operates between reservoirs at temperatures of 50°C and – 25°C. The heat transfer to the engine is 2500 kJ and the net work output of the combined engine refrigerator plant is 400 kJ. (i) Determine the heat transfer to the refrigerant and the net heat transfer to the reservoir at 50°C ; (ii) Reconsider (i) given that the efficiency of the heat engine and the C.O.P. of the refrigerator are each 45 per cent of their maximum possible values.

Solution:

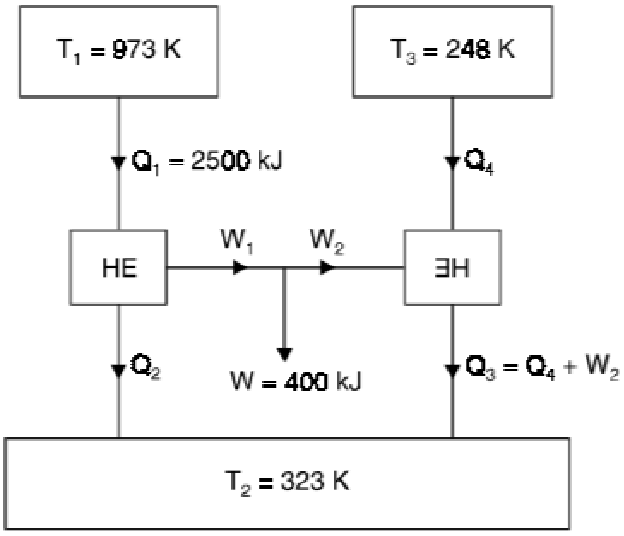


Fig. 8.5

Temperature, $T_1 = 700 + 273 = 973$ K

Temperature, $T_2 = 50 + 273 = 323$ K

Temperature, $T_3 = -25 + 273 = 248$ K

The heat transfer to the heat engine, $Q_1 = 2500$ kJ

The network output of the combined engine refrigerator plant,

$$W = W_1 - W_2 = 400 \text{ kJ.}$$

(*i*) Maximum efficiency of the heat engine cycle is given by

$$\eta_{max} = 1 - \frac{T_2}{T_1} = 1 - \frac{323}{973} = 0.668$$

Again, $$\frac{W_1}{Q_1} = 0.668$$

∴ $$W_1 = 0.668 \times 2500 = 1670 \text{ kJ}$$

$$(\text{C.O.P.})_{max} = \frac{T_3}{T_2 - T_3} = \frac{248}{323 - 248} = 3.306$$

Also, $$\text{C.O.P.} = \frac{Q_4}{W_2} = 3.306$$

Since, $$W_1 - W_2 = W = 400 \text{ kJ}$$

$$W_2 = W_1 - W = 1670 - 400 = 1270 \text{ kJ}$$

∴ $$Q_4 = 3.306 \times 1270 = 4198.6 \text{ kJ}$$

$$Q_3 = Q_4 + W_2 = 4198.6 + 1270 = 5468.6 \text{ kJ}$$

$$Q_2 = Q_1 - W_1 = 2500 - 1670 = 830 \text{ kJ.}$$

Heat rejection to the 50°C reservoir

$$= Q_2 + Q_3 = 830 + 5468.6 = \mathbf{6298.6 \text{ kJ.}} \quad \textbf{(Ans.)}$$

(*ii*) Efficiency of actual heat engine cycle,

$$\eta = 0.45\ \eta_{max} = 0.45 \times 0.668 = 0.3$$

∴ $$W_1 = \eta \times Q_1 = 0.3 \times 2500 = 750 \text{ kJ}$$

∴ $$W_2 = 750 - 400 = 350 \text{ kJ}$$

C.O.P. of the actual refrigerator cycle,

$$\text{C.O.P.} = \frac{Q_4}{W_2} = 0.45 \times 3.306 = 1.48$$

∴ $$Q_4 = 350 \times 1.48 = \mathbf{518 \text{ kJ.}} \quad \textbf{(Ans.)}$$

$$Q_3 = 518 + 350 = 868 \text{ kJ}$$

$$Q_2 = 2500 - 750 = 1750 \text{ kJ}$$

Heat rejected to 50°C reservoir

$$= Q_2 + Q_3 = 1750 + 868 = \mathbf{2618 \text{ kJ.}} \quad \textbf{(Ans.)}$$

Problem 7: A reversible heat pump is used to maintain a temperature of 0°C in a refrigerator when it rejects the heat to the surroundings at 25°C. If the heat removal rate from the refrigerator is 1440 kJ/min, determine the C.O.P. of the machine and work input required. (ii)

If the required input to run the pump is developed by a reversible engine which receives heat at 380°C and rejects heat to atmosphere, then determine the overall C.O.P. of the system.

Solution. (i) Temperature, T1 = 25 + 273 = 298 K Temperature, T2 = 0 + 273 = 273 K

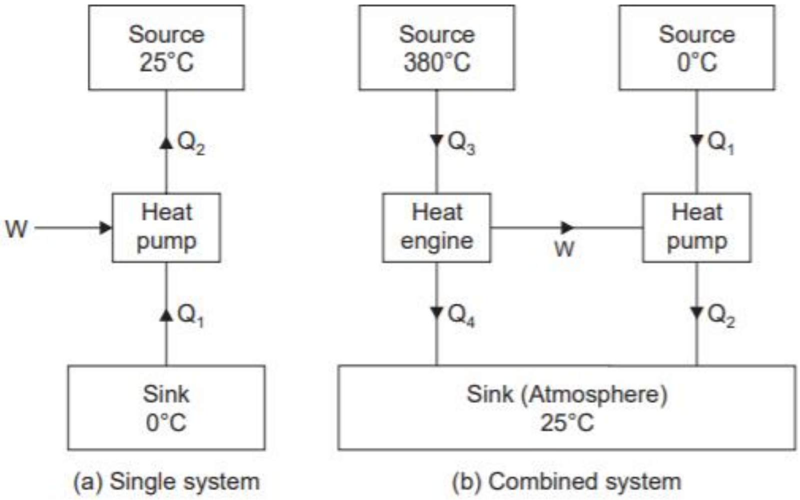


Fig. 8.6

Heat removal rate from the refrigerator,

$$Q_1 = 1440 \text{ kJ/min} = 24 \text{ kJ/s}$$

Now, co-efficient of performance, for reversible heat pump,

$$\textbf{C.O.P.} = \frac{T_1}{T_1 - T_2} = \frac{298}{(298 - 273)} = \textbf{11.92.} \quad \textbf{(Ans.)}$$

$$\therefore \qquad (\text{C.O.P.})_{ref.} = \frac{T_2}{T_1 - T_2} = \frac{273}{298 - 273} = 10.92$$

Now, $$10.92 = \frac{Q_1}{W} = \frac{24}{W}$$

$$\therefore \qquad W = 2.2 \text{ kW}$$

i.e., **Work input required = 2.2 kW. (Ans.)**

$$Q_2 = Q_1 + W = 24 + 2.2 = 26.2 \text{ kJ/s}$$

(*ii*) Refer Fig. 5.15 (*b*).

The overall C.O.P. is given by,

$$\text{C.O.P.} = \frac{\text{Heat removed from the refrigerator}}{\text{Heat supplied from the source}}$$

$$= \frac{Q_1}{Q_3}$$

For the reversible engine, we can write

$$\frac{Q_3}{T_3} = \frac{Q_4}{T_4}$$

or $$\frac{Q_4 + W}{T_3} = \frac{Q_4}{T_4}$$

or $$\frac{Q_4 + 2.2}{(380 + 273)} = \frac{Q_4}{(25 + 273)}$$

or $$\frac{Q_4 + 2.2}{653} = \frac{Q_4}{298}$$

or $$298(Q_4 + 2.2) = 653\ Q_4$$

or $$Q_4(653 - 298) = 298 \times 2.2$$

or $$Q_4 = \frac{298 \times 2.2}{(653 - 298)} = 1.847 \text{ kJ/s}$$

$\therefore$ $$Q_3 = Q_4 + W = 1.847 + 2.2 = 4.047 \text{ kJ/s}$$

Substituting this value in eqn. (*i*), we get

$$\textbf{C.O.P.} = \frac{24}{4.047} = \mathbf{5.93.} \quad \textbf{(Ans.)}$$

If the purpose of the system is to supply the heat to the sink at 25°C, then

$$\textbf{Overall C.O.P.} = \frac{Q_2 + Q_4}{Q_3} = \frac{26.2 + 1.847}{4.047} = \mathbf{6.93.} \quad \textbf{(Ans.)}$$

Entropy:

Introduction

In heat engine theory, the term entropy plays a vital role and leads to important results which by other methods can be obtained much more laboriously. It may be noted that all heat is not equally valuable for converting into work. Heat that is supplied to a substance at high temperature has a greater possibility of conversion into work than heat supplied to a substance at a lower temperature. “Entropy is a function of a quantity of heat which shows the possibility of conversion of that heat into work. The increase in entropy is small when heat is added at a high temperature and is greater when heat addition is made at a lower temperature. Thus for maximum entropy, there is minimum availability for conversion into work and for minimum entropy there is maximum availability for conversion into work.”

Entropy—a Property of a System

Let us consider a system undergoing a reversible process from state 1 to state 2 along path L and then from state 2 to the original state 1 along path M. Applying the Clausius theorem to this reversible cyclic process, we have

$$\oint_R \frac{\delta Q}{T} = 0$$

Hence when the system passes through the cycle 1-L-2-M-1, we have

$$\int_{1(L)}^{2} \frac{\delta Q}{T} + \int_{2(M)}^{1} \frac{\delta Q}{T} = 0$$ 1

Now consider another reversible cycle in which the system changes from state 1 to state 2 along path L, but returns from state 2 to the original state 1 along a different path N. For this reversible cyclic process, we have,

$$\int_{1(L)}^{2} \frac{\delta Q}{T} + \int_{2(N)}^{1} \frac{\delta Q}{T} = 0$$ 2

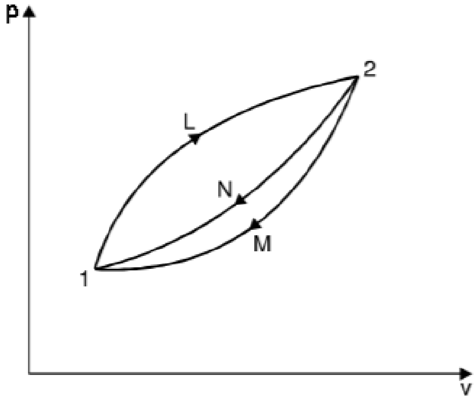


Fig.9. Reversible cyclic process between two fixed end states

Subtracting equation (1) from equation (2), we have

$$\int_{2(M)}^{1} \frac{\delta Q}{T} - \int_{2(N)}^{1} \frac{\delta Q}{T} = 0$$

$$\int_{1}^{2(M)} \frac{\delta Q}{T} = \int_{1}^{2(N)} \frac{\delta Q}{T}$$

As no restriction is imposed on paths L and M, except that they must be reversible, the quantity δQ/T is a function of the initial and final states of the system and is independent of the path of the process. Hence it represents a property of the system. This property is known as the “entropy”.

Maxwell Equation:

The first law applied to a closed system undergoing a reversible process states

$$dQ = du + pdv$$

According to second law,

$$ds = \left(\frac{dQ}{T}\right)_{\text{rev.}}$$

Combining these equations, we get

$$Tds = du + pdv$$

or

$$du = Tds - pdv$$

The properties h, f and g may also be put in terms of T, s, p and v as follows :

$$dh = du + pdv + vdp = Tds + vdp$$

Helmholtz free energy function,

$$df = du - Tds - sdT$$

$$= -\ pdv - sdT$$

Gibb's free energy function,

$$dg = dh - Tds - sdT = vdp - sdT$$

Each of these equations is a *result of the two laws of thermodynamics.*

Since du, dh, df and dg are the *exact differentials,* we can express them as

$$du = \left(\frac{\partial u}{\partial s}\right)_v ds + \left(\frac{\partial u}{\partial v}\right)_s dv,$$

$$dh = \left(\frac{\partial h}{\partial s}\right)_p ds + \left(\frac{\partial h}{\partial p}\right)_s dp,$$

$$df = \left(\frac{\partial f}{\partial v}\right)_T dv + \left(\frac{\partial f}{\partial T}\right)_v dT,$$

$$dg = \left(\frac{\partial g}{\partial p}\right)_T dp + \left(\frac{\partial g}{\partial T}\right)_p dT.$$

Comparing these equations, we may equate the corresponding co-efficients. For example, from the two equations for du, we have

$$\left(\frac{\partial u}{\partial s}\right)_v = T \text{ and } \left(\frac{\partial u}{\partial v}\right)_s = -p$$

The complete group of such relations may be summarised as follows :

$$\left(\frac{\partial u}{\partial s}\right)_v = T = \left(\frac{\partial h}{\partial s}\right)_p$$

$$\left(\frac{\partial u}{\partial v}\right)_s = -p = \left(\frac{\partial f}{\partial v}\right)_T$$

$$\left(\frac{\partial h}{\partial p}\right)_s = v = \left(\frac{\partial g}{\partial p}\right)_T$$

$$\left(\frac{\partial f}{\partial T}\right)_v = -s = \left(\frac{\partial g}{\partial T}\right)_p$$

Also,

$$\boxed{\begin{aligned}
\left(\frac{\partial T}{\partial v}\right)_s &= -\left(\frac{\partial p}{\partial s}\right)_v \\
\left(\frac{\partial T}{\partial p}\right)_s &= \left(\frac{\partial v}{\partial s}\right)_p \\
\left(\frac{\partial p}{\partial T}\right)_v &= \left(\frac{\partial s}{\partial v}\right)_T \\
\left(\frac{\partial v}{\partial T}\right)_p &= -\left(\frac{\partial s}{\partial p}\right)_T
\end{aligned}}$$

These equations are called as Maxwell Equations.

TEXT / REFERENCE BOOKS

1. Nag P.K., “Engineering Thermodynamics”, Tata McGraw Hill Education, 2009.
2. Yunus A. Cengel, Michael A. Boles, “Thermodynamics: An Engineering Approach”, McGraw Hill Education, 2014.
3. Rajput R.K., “Engineering Thermodynamics”, Laxmi Publications, 2010.
4. Khurmi R.S., Gupta J.K, “Thermal Engineering”, S Chand, 2006.
5. P.L. Ballaney,” Thermal Engineering”, Khanna Publisher, 2005.

SATHYABAMA
INSTITUTE OF SCIENCE AND TECHNOLOGY
(DEEMED TO BE UNIVERSITY)
Accredited "A" Grade by NAAC | 12B Status by UGC | Approved by AICTE
www.sathyabama.ac.in

SCHOOL OF MECHANICAL ENGINEERING

DEPARTMENT OF AERONAUTICAL ENGINEERING

UNIT – I – Aero Engineering Thermodynamics – SAEA1301

I. INTRODUCTION TO AERO ENGINE THERMODYNAMICS

Thermodynamics may be defined as follows:

- Thermodynamics is an axiomatic science which deals with the relations among heat, work and properties of system which are in equilibrium. It describes state and changes in state of physical systems.

Or

- Thermodynamics is the science of the regularities governing processes of energy conversion.

Or

- Thermodynamics is the science that deals with the interaction between energy and material systems.

Thermodynamics, basically entails four laws or axioms known as Zeroth, First, Second and Third law of thermodynamics.

- The First law throws light on concept of internal energy.
- The Zeroth law deals with thermal equilibrium and establishes a concept of temperature.
- The Second law indicates the limit of converting heat into work and introduces the
- principle of increase of entropy.
- The Third law defines the absolute zero of entropy.

Thermodynamic System: A system is a finite quantity of matter or a prescribed region of space (Refer Fig. 1)

Boundary. The actual or hypothetical envelope enclosing the system is the boundary of the system. The boundary may be fixed or it may move, as and when a system containing a gas is compressed or expanded. The boundary may be real or imaginary.

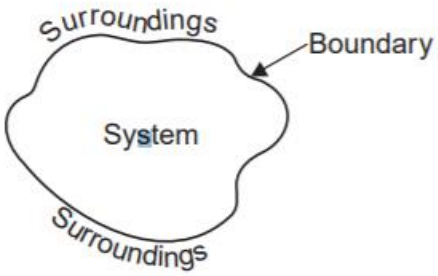


Fig. 1. Description of a system

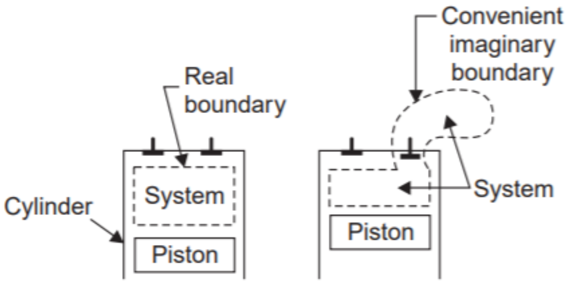


Fig. 2. Real and Imaginary Boundary

Closed System

If the boundary of the system is impervious to the flow of matter, it is called a closed system. An example of this system is mass of gas or vapour contained in an engine cylinder, the boundary of which is drawn by the cylinder walls, the cylinder head and piston crown. Here the boundary is continuous and no matter may enter or leave.

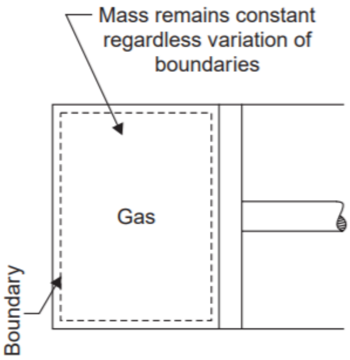


Fig. 3. Closed system

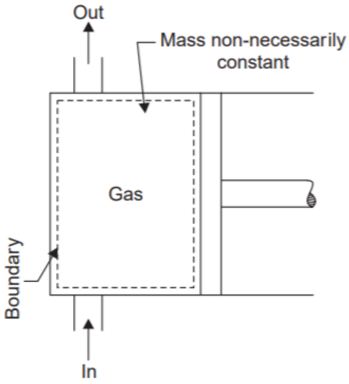


Fig. 4. Open system

Open System

An open system is one in which matter flows into or out of the system. Most of the engineering systems are open.

Isolated System

An isolated system is that system which exchanges neither energy nor matter with any other system or with environment.

Phase. A phase is a quantity of matter which is homogeneous throughout in chemical composition and physical structure.

Homogeneous System

A system which consists of a single phase is termed as homogeneous system. Examples: Mixture of air and water vapour, water plus nitric acid and octane plus heptane.

Heterogeneous System

A system which consists of two or more phases is called a heterogeneous system. Examples: Water plus steam, ice plus water and water plus oil.

Macroscopic and microscopic points of view

Table 1. Macroscopic and microscopic points of view

S.No	Macroscopic approach	Microscopic approach
1	In this approach a certain quantity of matter is considered without taking into account the events occurring at molecular level. In other words, this approach to thermodynamics is concerned with gross or overall	The approach considers that the system is made up of a very large number of discrete particles known as molecules. These molecules have different velocities and energies. The values of these energies are constantly changing with time. This

	behaviour. This is known as classical thermodynamics.	approach to thermodynamics which is concerned directly with the structure of the matter is known as statistical thermodynamics.
2	The analysis of macroscopic system requires simple mathematical formulae.	The behaviour of the system is found by using statistical methods as the number of molecules is very large. So advanced statistical and mathematical methods are needed to explain the changes in the system.
3	The values of the properties of the system are their average values. These properties like pressure and temperature can be measured very easily. The changes in properties can be felt by our senses	The properties like velocity, momentum, impulse, kinetic energy, force of impact etc. which describe the molecule cannot be easily measured by instruments. Our senses cannot feel them.
4	In order to describe a system only a few properties are needed.	Large number of variables are needed to describe a system. So, the approach is complicated.

Pure substance

A pure substance is one that has a homogeneous and invariable chemical composition even though there is a change of phase. In other words, it is a system which is (a) homogeneous in composition, (b) homogeneous in chemical aggregation. Examples: Liquid, water, mixture of liquid water and steam, mixture of ice and water. The mixture of liquid air and gaseous air is not a pure substance.

Thermodynamic equilibrium

A system is in thermodynamic equilibrium if the temperature and pressure at all points are same; there should be no velocity gradient; the chemical equilibrium is also necessary.

Thus, for attaining a state of thermodynamic equilibrium the following three types of equilibrium states must be achieved:

1**. Thermal equilibrium.** The temperature of the system does not change with time and has same value at all points of the system.

2. **Mechanical equilibrium**. There are no unbalanced forces within the system or between the surroundings. The pressure in the system is same at all points and does not change with respect to time.

3. **Chemical equilibrium.** No chemical reaction takes place in the system and the chemical composition which is same throughout the system does not vary with time.

Properties of systems

A property of a system is a characteristic of the system which depends upon its state, but not upon how the state is reached. There are two sorts of property:

1. **Intensive properties**. These properties do not depend on the mass of the system. Examples: Temperature and pressure.

2. **Extensive properties.** These properties depend on the mass of the system. Example: Volume. Extensive properties are often divided by mass associated with them to obtain the intensive properties. For example, if the volume of a system of mass m is V, then the specific volume of matter within the system is $\frac{V}{m} = v$ which is an intensive property.

State

State is the condition of the system at an instant of time as described or measured by its properties. Or each unique condition of a system is called a state.

Process

A process occurs when the system undergoes a change in a state or an energy transfer at a steady state.

Non-flow process

A process may be non-flow in which a fixed mass within the defined boundary is undergoing a change of state. Example: A substance which is being heated in a closed cylinder undergoes a non-flow process. Closed systems undergo non-flow processes.

Flow process

A process may be a flow process in which mass is entering and leaving through the boundary of an open system. In a steady flow process mass is crossing the boundary from surroundings at entry, and an equal mass is crossing the boundary at the exit so that the total mass of the system remains constant.

Quasi-static process. Quasi means ‘almost’. A quasi-static process is also called a reversible process. This process is a succession of equilibrium states and infinite slowness is its characteristic feature.

Cycle

Any process or series of processes whose end states are identical is termed a cycle.. Fig. 5 shows such a cycle in which a system commencing at condition ‘1’ changes in pressure and volume through a path 123 and returns to its initial condition ‘1’.

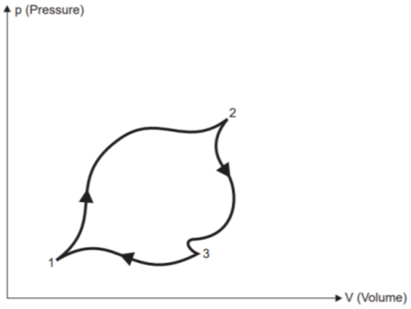


Fig.5. Cycle

Point function

When two properties locate a point on the graph (co-ordinate axes) then those properties are called as point function. Examples. Pressure, temperature, volume etc.

$$\int_1^2 dV = V_2 - V_1 \text{ (an } \textit{exact differential}\text{).}$$

Path function

There are certain quantities which cannot be located on a graph by a point but are given by the area or so, on that graph. In that case, the area on the graph, pertaining to the particular process, is a function of the path of the process. Such quantities are called path functions. Examples. Heat, work etc. Heat and work are inexact differentials. Their change cannot be written as difference between their end states.

$$\text{Thus } \int_1^2 \delta Q \neq Q_2 - Q_1 \text{ and is shown as } {}_1Q_2 \text{ or } Q_{1-2}$$

$$\text{Similarly } \int_1^2 \delta W \neq W_2 - W_1, \text{ and is shown as } {}_1W_2 \text{ or } W_{1-2}$$

Zeroth law of thermodynamics

Zeroth law of thermodynamics states that if two systems are each equal in temperature to a third, they are equal in temperature to each other.

Energy

Energy is a general term embracing energy in transition and stored energy. The stored energy of a substance may be in the forms of mechanical energy and internal energy (other forms of stored energy may be chemical energy and electrical energy). Part of the stored energy may take the form of either potential energy (which is the gravitational energy due to height above a chosen datum line) or kinetic energy due to velocity. The balance part of the energy is known as internal energy. In a non-flow process usually, there is no change of potential or kinetic energy and hence change of mechanical energy will not enter the calculations. In a flow process, however, there may be changes in both potential and kinetic energy and these must be taken into account while considering the changes of stored energy. Heat and work are the forms of energy in transition. These are the only forms in which energy can cross the boundaries of a system. Neither heat nor work can exist as stored energy.

Work

Work is said to be done when a force moves through a distance. If a part of the boundary of a system undergoes a displacement under the action of a pressure, the work done W is the product of the force (pressure × area), and the distance it moves in the direction of the force. Fig. 6 (a) illustrates this with the conventional piston and cylinder arrangement, the heavy line defining the boundary of the system. Fig. 6 (b) illustrates another way in which work might be applied to a system. A force is exerted by the paddle as it changes the momentum of the fluid, and since this force moves during rotation of the paddle work is done.

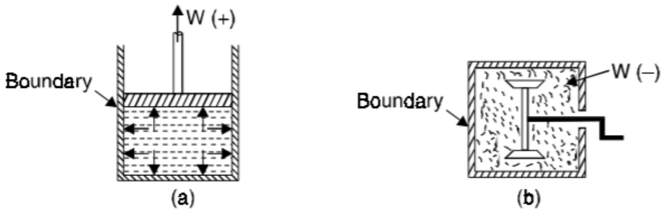


Fig. 6. Work Transfer

Internal energy (ΔU).

It is the heat energy stored in a gas. If a certain amount of heat is supplied to a gas the result is that temperature of gas may increase or volume of gas may increase thereby doing some external work or both temperature and volume may increase; but it will be decided by the conditions under which the gas is supplied heat. If during heating of the gas the temperature increases its internal energy will also increase.

Joule's law of internal energy states that internal energy of a perfect gas is a function of temperature only. In other words, internal energy of a gas is dependent on the temperature change only and is not affected by the change in pressure and volume.

First law of thermodynamics

When a system undergoes a thermodynamic cycle then the net heat supplied to the system from the surroundings is equal to net work done by the system on its surroundings.

$$\oint dQ = \oint dW$$

Application of first law to a process

When a process is executed by a system, the change in stored energy of the system is numerically equal to the net heat interactions minus the network interaction during the process.

$$\therefore \qquad E_2 - E_1 = Q - W$$

$$\therefore \qquad \Delta E = Q - W \qquad [\text{or } Q = \Delta E + W]$$

$$\int_1^2 d(Q - W) = \Delta E = E_2 - E_1$$

where E represents the total internal energy

Application of first law of thermodynamics to non-flow or Closed system

Reversible Constant Volume (or Isochoric) Process (v = constant) :

In a constant volume process the working substance is contained in a rigid vessel, hence the boundaries of the system are immovable and no work can be done on or by the system, other than paddle-wheel work input. It will be assumed that ‘constant volume’ implies zero work unless stated otherwise. Fig. 7 shows the system and states before and after the heat addition at constant volume.

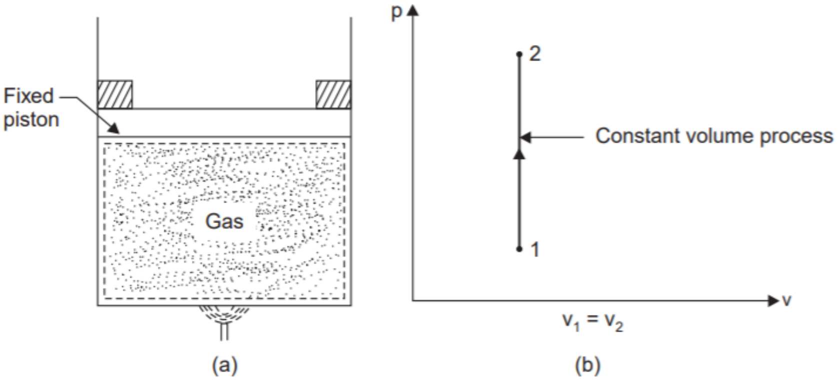


Fig. 7 Reversible constant volume process.

Considering mass of the working substance unity and applying first law of thermodynamics to the process,

$$Q = (u_2 - u_1) + W$$

The work done $W = \int_1^2 p\,dv = 0$ as $dv = 0$.

$$\therefore \quad Q = (u_2 - u_1) = c_v(T_2 - T_1)$$

where c_v = Specific heat at constant volume.

For mass, m, of working substance

$$Q = U_2 - U_1 = mc_v(T_2 - T_1)$$

Reversible Constant Pressure (or Isobaric) Process (p = constant).

It can be seen from Fig. 7 (b) that when the boundary of the system is inflexible as in a constant volume process, then the pressure rises when heat is supplied. Hence for a constant pressure process, the boundary must move against an external resistance as heat is supplied;

for instance, a gas [Fig. 8 (a)] in a cylinder behind a piston can be made to undergo a constant pressure process. Since the piston is pushed through a certain distance by the force exerted by the gas, then the work is done by the gas on its surroundings. Fig. 8 shows the system and states before and after the heat addition at constant pressure.

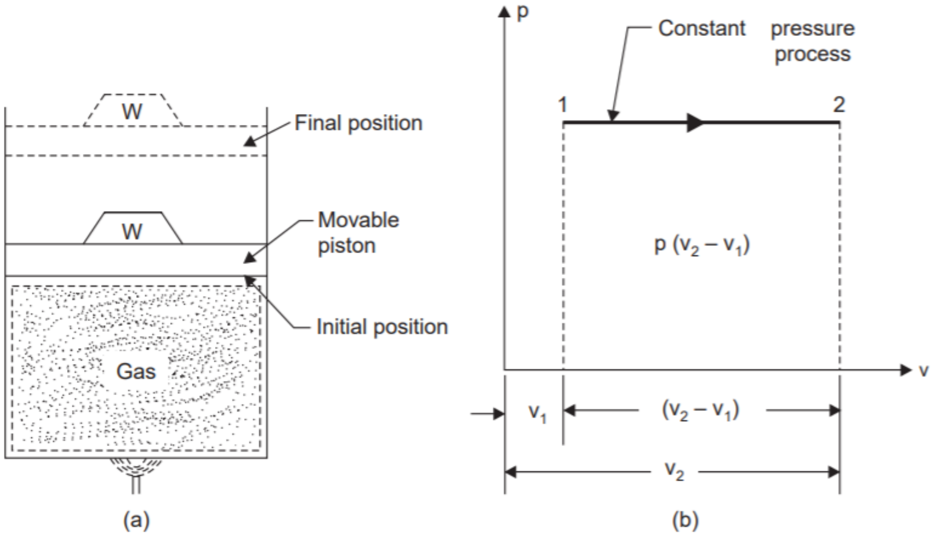


Fig. 8 Reversible constant pressure process.

Considering unit mass of working substance and applying first law of thermodynamics to the process,

$$Q = (u_2 - u_1) + W$$

The work done, $W = \int_1^2 p\,dv = p(v_2 - v_1)$

$\therefore$ $Q = (u_2 - u_1) + p(v_2 - v_1) = u_2 - u_1 + pv_2 - pv_1$

$$= (u_2 + pv_2) - (u_1 + pv_1) = h_2 - h_1$$

or $Q = h_2 - h_1 = c_p\,(T_2 - T_1)$

where h = Enthalpy (specific), and

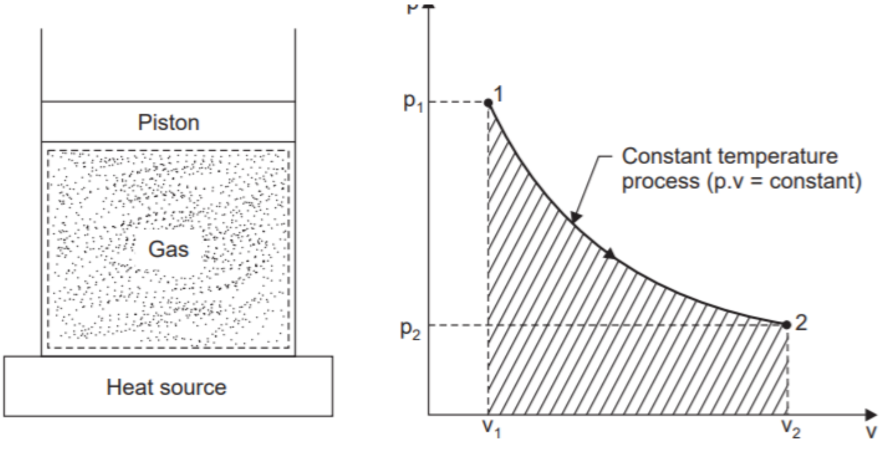
c_p = Specific heat at constant pressure.

For mass, m, of working substance

$$Q = H_2 - H_1 = mc_p\,(T_2 - T_1)$$

Reversible Temperature (or Isothermal) Process (pv = constant, T = constant) :

A process at a constant temperature is called an isothermal process. When a working substance in a cylinder behind a piston expands from a high pressure to a low pressure there is a tendency for the temperature to fall. In an isothermal expansion heat must be added continuously in order to keep the temperature at the initial value. Similarly, in an isothermal compression heat must be removed from the working substance continuously during the process. Fig. 9 shows the system and states before and after the heat addition at constant temperature.



(a) (b)

Fig. 9. Reversible isothermal process.

Considering unit mass of working substance and applying first law to the process,

$$\begin{aligned} Q &= (u_2 - u_1) + W \\ &= c_v (T_2 - T_1) + W \\ &= 0 + W \qquad [\because \quad T_2 = T_1] \end{aligned}$$

The work done, $W = \int_1^2 p dv$

In this case pv = constant or $p = \frac{C}{v}$ (where C = constant)

$$\therefore \qquad W = \int_{v_1}^{v_2} C \frac{dv}{v} = C[\log_e v]_{v_1}^{v_2} = C \log_e \frac{v_2}{v_1}$$

The constant C can either be written as p_1v_1 or as p_2v_2, since

$$p_1v_1 = p_2v_2 = \text{constant, } C$$

i.e., $W = p_1v_1 \log_e \frac{v_2}{v_1}$ per unit mass of working substance

or $W = p_2v_2 \log_e \frac{v_2}{v_1}$ per unit mass of working substance

$$\therefore \qquad Q = W = p_1v_1 \log_e \frac{v_2}{v_1}$$

For mass, m, of the working substance

$$Q = p_1V_1 \log_e \frac{V_2}{V_1}$$

or $Q = p_1V_1 \log_e \frac{p_1}{p_2} \qquad \left[\because \frac{V_2}{V_1} = \frac{p_1}{p_2}\right]$

Reversible Adiabatic Process (pv^γ = constant) :

An adiabatic process is one in which no heat is transferred to or from the fluid during the process. Such a process can be reversible or irreversible.

Considering unit mass of working substance and applying first law to the process,

$$Q = (u_2 - u_1) + W$$
$$O = (u_2 - u_1) + W$$
$$W = (u_1 - u_2) \text{ for any adiabatic process}$$

In an adiabatic expansion, the work done W by the fluid is at the expense of a reduction in the internal energy of the fluid. Similarly, in an adiabatic compression process all the work done on the fluid goes to increase the internal energy of the fluid. For an adiabatic process to take place, perfect thermal insulation for the system must be available.

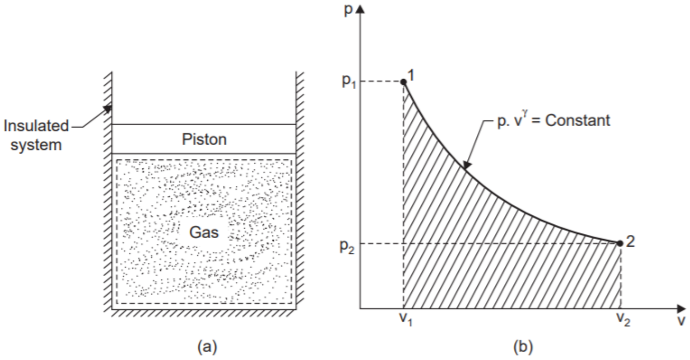


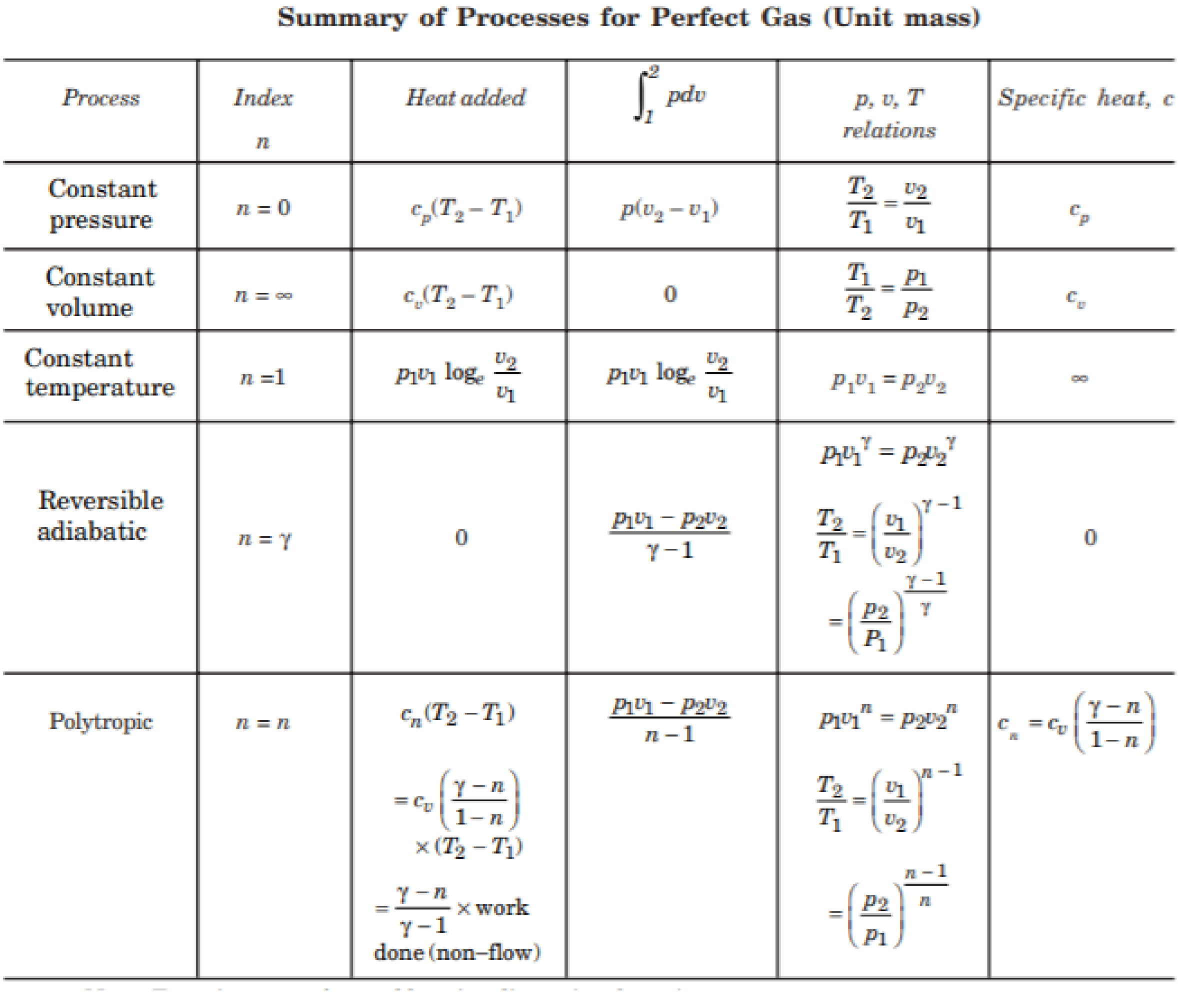
Fig. 10. Reversible adiabatic process

Polytropic Reversible Process (pv^n = constant) :

It is found that many processes in practice approximate to a reversible law of form pv^n = constant, where n is a constant. Both vapours and perfect gases obey this type of law closely in many non-flow processes. Such processes are internally reversible.

$$\text{Work done, } W = \frac{p_1v_1 - p_2v_2}{n-1}$$
$$W = \frac{R(T_1 - T_2)}{n-1}$$

Table 2. Expressions for all Process

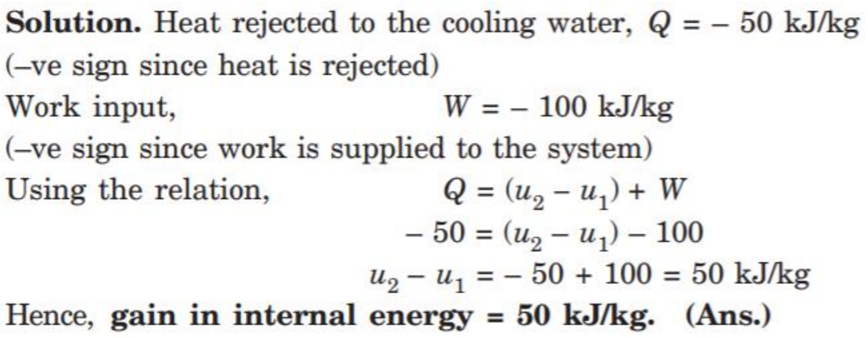


Summary of Processes for Perfect Gas (Unit mass)

Process	*Index* *n*	*Heat added*	$\int_1^2 pdv$	*p, v, T relations*	*Specific heat, c*
Constant pressure	$n = 0$	$c_p(T_2 - T_1)$	$p(v_2 - v_1)$	$\frac{T_2}{T_1} = \frac{v_2}{v_1}$	c_p
Constant volume	$n = \infty$	$c_v(T_2 - T_1)$	0	$\frac{T_1}{T_2} = \frac{p_1}{p_2}$	c_v
Constant temperature	$n = 1$	$p_1 v_1 \log_e \frac{v_2}{v_1}$	$p_1 v_1 \log_e \frac{v_2}{v_1}$	$p_1 v_1 = p_2 v_2$	∞
Reversible adiabatic	$n = \gamma$	0	$\frac{p_1 v_1 - p_2 v_2}{\gamma - 1}$	$p_1 v_1^{\gamma} = p_2 v_2^{\gamma}$ $\frac{T_2}{T_1} = \left(\frac{v_1}{v_2}\right)^{\gamma - 1}$ $= \left(\frac{p_2}{p_1}\right)^{\frac{\gamma - 1}{\gamma}}$	0
Polytropic	$n = n$	$c_n(T_2 - T_1)$ $= c_v\left(\frac{\gamma - n}{1 - n}\right) \times (T_2 - T_1)$ $= \frac{\gamma - n}{\gamma - 1} \times$ work done (non–flow)	$\frac{p_1 v_1 - p_2 v_2}{n - 1}$	$p_1 v_1^{n} = p_2 v_2^{n}$ $\frac{T_2}{T_1} = \left(\frac{v_1}{v_2}\right)^{n - 1}$ $= \left(\frac{p_2}{p_1}\right)^{\frac{n - 1}{n}}$	$c_n = c_v\left(\frac{\gamma - n}{1 - n}\right)$

Problems:

Problem 1: In an internal combustion engine, during the compression stroke the heat rejected to the cooling water is 50 kJ/kg and the work input is 100 kJ/kg. Calculate the change in internal energy of the working fluid stating whether it is a gain or loss.



Solution. Heat rejected to the cooling water, $Q = -50$ kJ/kg

(–ve sign since heat is rejected)

Work input, $W = -100$ kJ/kg

(–ve sign since work is supplied to the system)

Using the relation, $Q = (u_2 - u_1) + W$

$$-50 = (u_2 - u_1) - 100$$

$$u_2 - u_1 = -50 + 100 = 50 \text{ kJ/kg}$$

Hence, **gain in internal energy = 50 kJ/kg. (Ans.)**

Problem 2: Air enters a compressor at 105 Pa and 25°C having volume of 1.8 m3/kg and is compressed to 5 × 105 Pa isothermally. Determine: (i) Work done; (ii) Change in internal energy ; and (iii) Heat transferred.

Solution. Initial pressure of air, $p_1 = 10^5$ Pa

Initial temperature of air, $T_1 = 25 + 273 = 298$ K

Final pressure of air, $p_2 = 5 \times 10^5$ Pa

Final temperature of air, $T_2 = T_1 = 298$ K (isothermal process)

Since, it is a closed steady state process, we can write down the first law of thermodynamics

$$Q = (u_2 - u_1) + W \text{ ......per kg}$$

(*i*) *For isothermal process* :

$$W_{1-2} = \int_1^2 p \, . dv = p_1 v_1 \log_e \left(\frac{p_1}{p_2}\right)$$

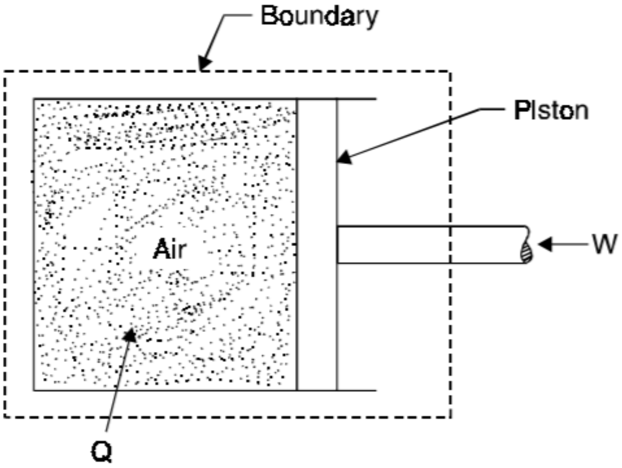


Fig. 11 Closed System

$$p_1 v_1 = p_2 v_2 \text{ for isothermal process}$$

$$\therefore \qquad W_{1-2} = -\,10^5 \times 1.8 \log_e \left(\frac{1 \times 10^5}{5 \times 10^5}\right)$$

$$= -\,2.897 \times 10^5 = -\,289.7 \text{ kJ/kg.}$$

(– ve sign indicates that the work is supplied to the air)

∴ **Work done on the air = 289.7 kJ/kg. (Ans.)**

(*ii*) Since temperature is constant,

$$\therefore \qquad u_2 - u_1 = 0$$

∴ **Change in internal energy = zero. (Ans.)**

(*iii*) Again, $Q_{1-2} = (u_2 - u_1) + W$

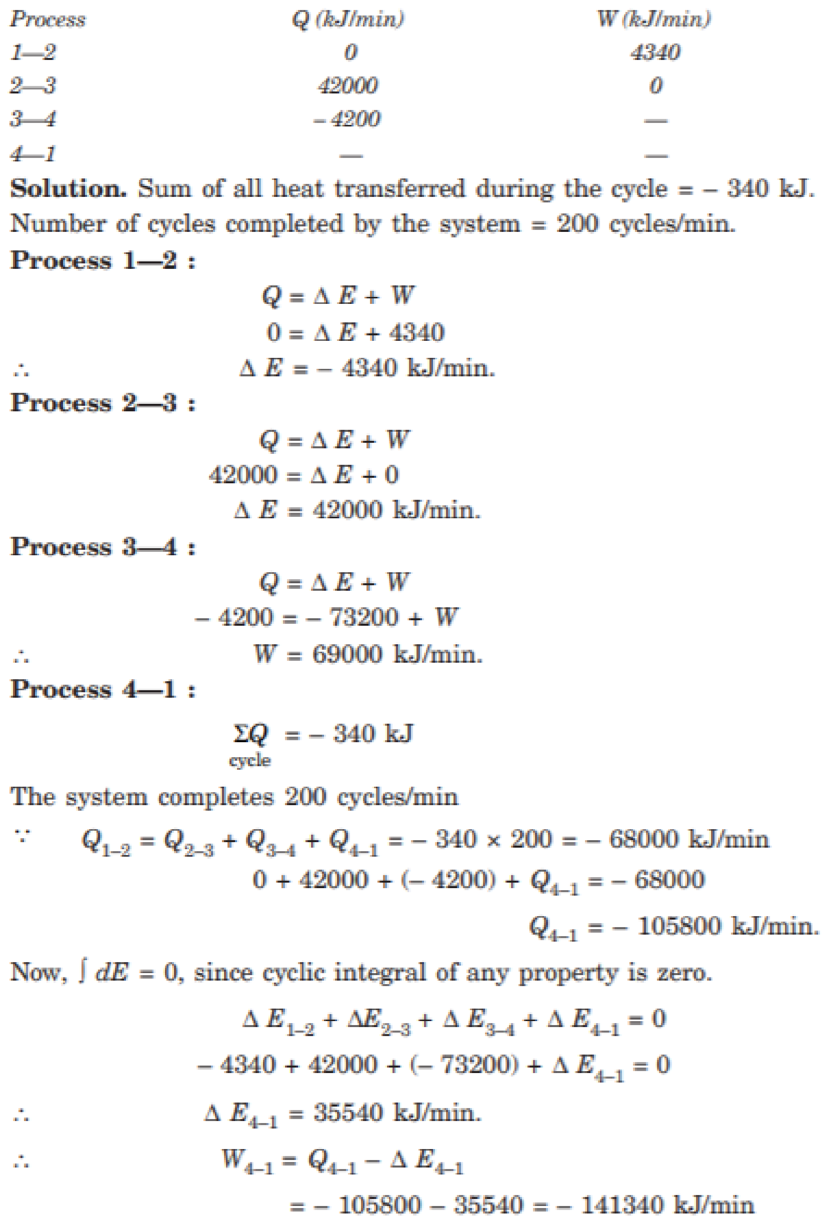
$$= 0 + (-\,289.7) = -\,289.7 \text{ kJ}$$

(– ve sign indicates that heat is lost from the system to the surroundings)

∴ **Heat rejected = 289.7 kJ/kg. (Ans.)**

Problem 3: A fluid system, contained in a piston and cylinder machine, passes through a complete cycle of four processes. The sum of all heat transferred during a cycle is – 340

kJ. The system completes 200 cycles per min. Complete the following table showing the method for each item, and compute the net rate of work output in kW



Process	*Q (kJ/min)*	*W (kJ/min)*	*ΔE (kJ/min)*
1—2	*0*	*4340*	—
2—3	*42000*	*0*	—
3—4	*– 4200*	—	*– 73200*
4—1	—	—	—

Solution. Sum of all heat transferred during the cycle = – 340 kJ.

Number of cycles completed by the system = 200 cycles/min.

Process 1—2 :

$$Q = \Delta E + W$$

$$0 = \Delta E + 4340$$

$$\therefore \quad \Delta E = -4340 \text{ kJ/min.}$$

Process 2—3 :

$$Q = \Delta E + W$$

$$42000 = \Delta E + 0$$

$$\Delta E = 42000 \text{ kJ/min.}$$

Process 3—4 :

$$Q = \Delta E + W$$

$$-4200 = -73200 + W$$

$$\therefore \quad W = 69000 \text{ kJ/min.}$$

Process 4—1 :

$$\sum_{\text{cycle}} Q = -340 \text{ kJ}$$

The system completes 200 cycles/min

$$\because \quad Q_{1-2} = Q_{2-3} + Q_{3-4} + Q_{4-1} = -340 \times 200 = -68000 \text{ kJ/min}$$

$$0 + 42000 + (-4200) + Q_{4-1} = -68000$$

$$Q_{4-1} = -105800 \text{ kJ/min.}$$

Now, $\int dE = 0$, since cyclic integral of any property is zero.

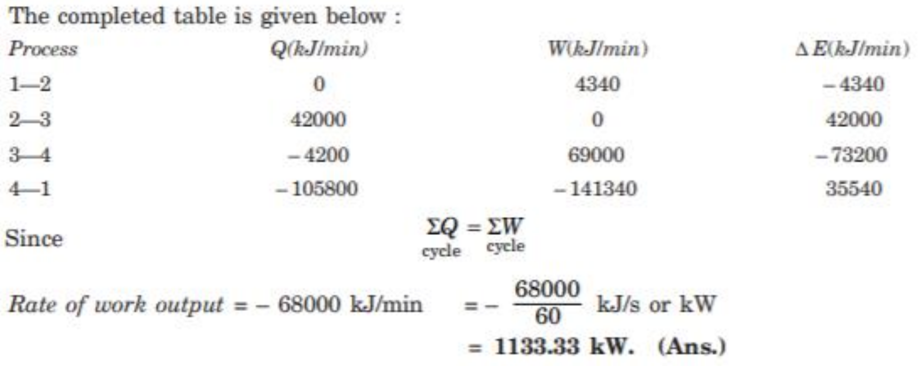
$$\Delta E_{1-2} + \Delta E_{2-3} + \Delta E_{3-4} + \Delta E_{4-1} = 0$$

$$-4340 + 42000 + (-73200) + \Delta E_{4-1} = 0$$

$$\therefore \quad \Delta E_{4-1} = 35540 \text{ kJ/min.}$$

$$\therefore \quad W_{4-1} = Q_{4-1} - \Delta E_{4-1}$$

$$= -105800 - 35540 = -141340 \text{ kJ/min}$$



The completed table is given below :

Process	*Q(kJ/min)*	*W(kJ/min)*	*ΔE(kJ/min)*
1—2	0	4340	– 4340
2—3	42000	0	42000
3—4	– 4200	69000	– 73200
4—1	– 105800	– 141340	35540

Since $\sum_{\text{cycle}} Q = \sum_{\text{cycle}} W$

Rate of work output = – 68000 kJ/min $= -\dfrac{68000}{60}$ kJ/s or kW

= **1133.33 kW. (Ans.)**

Problem 4: 0.44 kg of air at 180°C expands adiabatically to three times its original volume and during the process, there is a fall in temperature to 15°C. The work done during the process is 52.5 kJ. Calculate cp and cv

Solution:

Mass of air, $m = 0.44$ kg

Initial temperature, $T_1 = 180 + 273 = 453$ K

$$\text{Ratio} = \frac{V_2}{V_1} = 3$$

Final temperature, $T_2 = 15 + 273 = 288$ K

Work done during the process, $W_{1-2} = 52.5$ kJ

$$c_p = ?,\ c_v = ?$$

For adiabatic process, we have

$$\frac{T_2}{T_1} = \left(\frac{V_1}{V_2}\right)^{\gamma - 1}$$

$$\frac{288}{453} = \left(\frac{1}{3}\right)^{\gamma - 1} \quad \text{or} \quad 0.6357 = (0.333)^{\gamma-1}$$

or Taking log on both sides, we get

$$\log_e (0.6357) = (\gamma - 1) \log_e (0.333)$$

$$-0.453 = (\gamma - 1) \times (-1.0996)$$

$$\therefore \quad \gamma = \frac{0.453}{1.0996} + 1 = 1.41$$

Also, $$\frac{c_p}{c_v} = \gamma = 1.41$$

Work done during adiabatic process,

$$W_{1-2} = \frac{mR(T_1 - T_2)}{\gamma - 1}$$

$$\therefore \quad 52.5 = \frac{0.44\, R(453 - 288)}{(1.41 - 1)}$$

$$\therefore \quad R = \frac{52.5\,(1.41 - 1)}{0.44\,(453 - 288)} = 0.296$$

$$\therefore \quad c_p - c_v = 0.296 \qquad [\because \ R = c_p - c_v]$$

Also $$\frac{c_p}{c_v} = 1.41 \quad \text{or} \quad c_p = 1.41\, c_v$$

$$\therefore \quad 1.41\, c_v - c_p = 0.296$$

or c_v = **0.722 kJ/kg K. (Ans.)**

and c_p = **1.018 kJ/kg K. (Ans.)**

Problem 5: **0.1 m3 of an ideal gas at 300 K and 1 bar is compressed adiabatically to 8 bar. It is then cooled at constant volume and further expanded isothermally so as to reach the condition from where it started. Calculate: (i) Pressure at the end of constant volume cooling. (ii) Change in internal energy during constant volume process. (iii) Net work done and heat transferred during the cycle. Assume cp = 14.3 kJ/kg K and cv = 10.2 kJ/kg K**

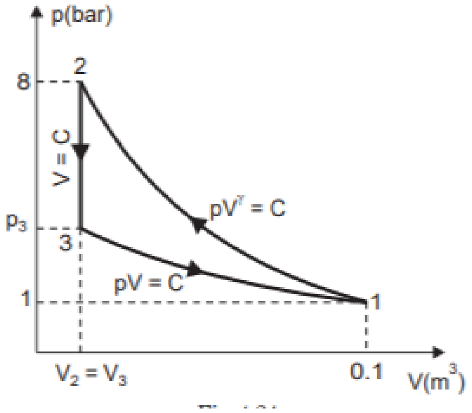


Fig. 11.1

Solution. *Given :* $V_1 = 0.1 \text{ m}^3$; $T_1 = 300$ K ; $p_1 = 1$ bar ; $c_p = 14.3$ kJ/kg K ; $c_v = 10.2$ kJ/kg K.

Refer to Fig. 4.24.

(*i*) **Pressure at the end of constant volume cooling, p_3 :**

$$\gamma = \frac{c_p}{c_v} = \frac{14.3}{10.2} = 1.402$$

Characteristic gas constant,

$$R = c_p - c_v = 14.3 - 10.2 = 4.1 \text{ kJ/kg K}$$

Considering *process 1-2*, we have :

$$p_1V_1^{\gamma} = p_2V_2^{\gamma}$$

$$V_2 = V_1 \times \left(\frac{p_1}{p_2}\right)^{\frac{1}{\gamma}} = 0.1 \times \left(\frac{1}{8}\right)^{\frac{1}{1.402}} = 0.0227 \text{ m}^3$$

Also, $$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{8}{1}\right)^{\frac{1.402-1}{1.402}} = 1.815$$

$$T_2 = T_1 \times 1.815 = 300 \times 1.815 = 544.5 \text{ K}$$

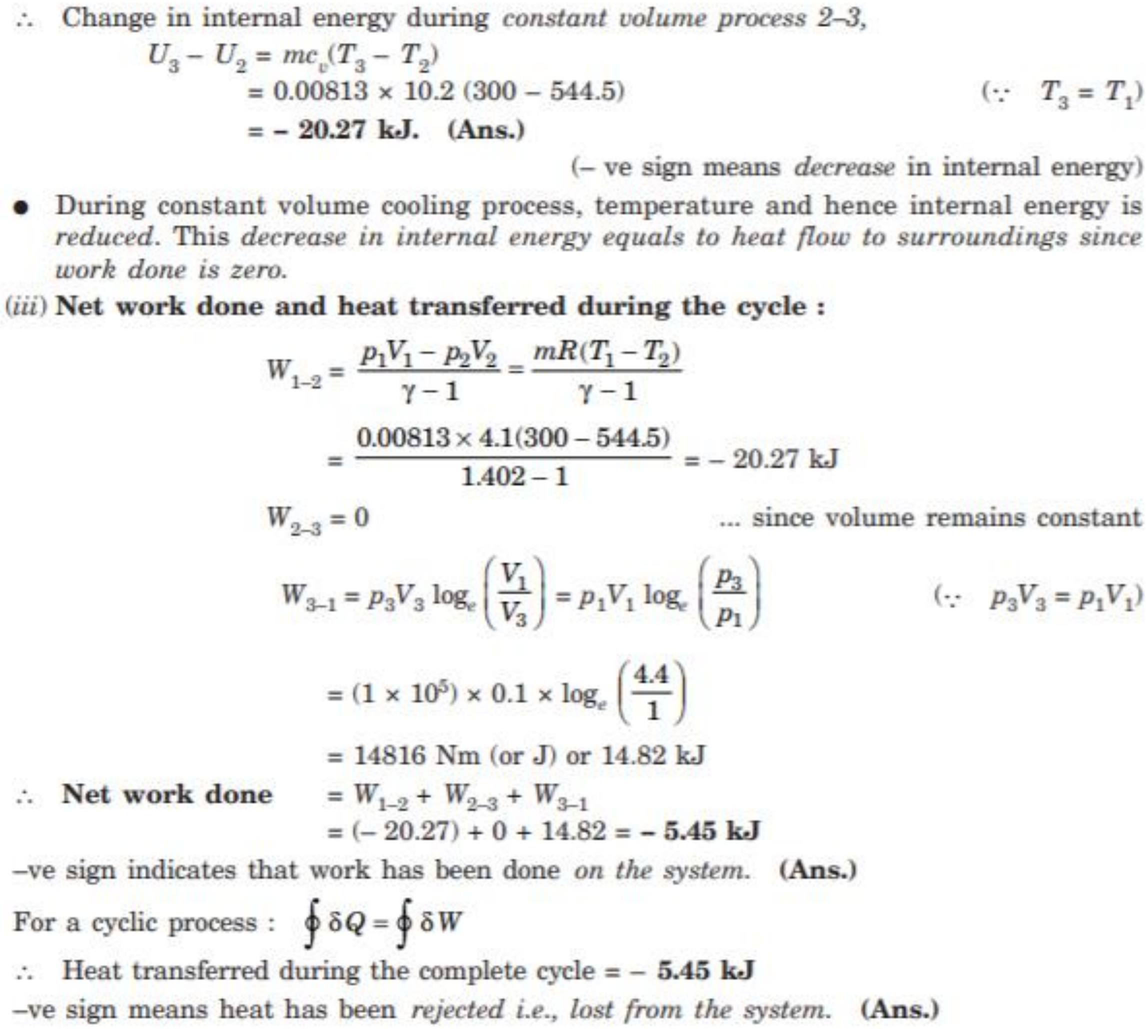
Considering *process 3–1*, we have

$$p_3V_3 = p_1V_1$$

$$\therefore \qquad p_3 = \frac{p_1V_1}{V_3} = \frac{1 \times 0.1}{0.0227} = \textbf{4.4 bar.} \quad \textbf{(Ans.)} \qquad (\because \ V_3 = V_2)$$

(*ii*) **Change in internal energy during constant volume process, $(U_3 - U_2)$:**

Mass of gas, $$m = \frac{p_1V_1}{RT_1} = \frac{(1 \times 10^5) \times 0.1}{(4.1 \times 1000) \times 300} = 0.00813 \text{ kg}$$



∴ Change in internal energy during *constant volume process 2–3,*

$$U_3 - U_2 = mc_v(T_3 - T_2)$$

$$= 0.00813 \times 10.2\,(300 - 544.5) \qquad (\because\ T_3 = T_1)$$

$$= -\,\mathbf{20.27\ kJ.}\quad \textbf{(Ans.)}$$

(– ve sign means *decrease* in internal energy)

- During constant volume cooling process, temperature and hence internal energy is *reduced*. This *decrease in internal energy equals to heat flow to surroundings since work done is zero.*

(iii) **Net work done and heat transferred during the cycle :**

$$W_{1-2} = \frac{p_1V_1 - p_2V_2}{\gamma - 1} = \frac{mR(T_1 - T_2)}{\gamma - 1}$$

$$= \frac{0.00813 \times 4.1(300 - 544.5)}{1.402 - 1} = -\,20.27\ \text{kJ}$$

$W_{2-3} = 0$ ... since volume remains constant

$$W_{3-1} = p_3V_3 \log_e\left(\frac{V_1}{V_3}\right) = p_1V_1 \log_e\left(\frac{p_3}{p_1}\right) \qquad (\because\ p_3V_3 = p_1V_1)$$

$$= (1 \times 10^5) \times 0.1 \times \log_e\left(\frac{4.4}{1}\right)$$

$$= 14816\ \text{Nm (or J) or } 14.82\ \text{kJ}$$

∴ **Net work done** $= W_{1-2} + W_{2-3} + W_{3-1}$

$= (-\,20.27) + 0 + 14.82 = \mathbf{-\,5.45\ kJ}$

–ve sign indicates that work has been done *on the system.* **(Ans.)**

For a cyclic process : $\oint \delta Q = \oint \delta W$

∴ Heat transferred during the complete cycle = – **5.45 kJ**

–ve sign means heat has been *rejected i.e., lost from the system.* **(Ans.)**

APPLICATION OF FIRST LAW TO STEADY FLOW PROCESS

Steady Flow Energy Equation (S.F.E.E.)

In many practical problems, the rate at which the fluid flows through a machine or piece of apparatus is constant. This type of flow is called steady flow.

Assumptions : The following assumptions are made in the system analysis : (i) The mass flow through the system remains constant. (ii) Fluid is uniform in composition. (iii) The only interaction between the system and surroundings are work and heat. (iv) The state of fluid at any point remains constant with time. (v) In the analysis only potential, kinetic and flow energies are considered.

Fig. 12. shows a schematic flow process for an open system. An open system is one in which both mass and energy may cross the boundaries. A wide interchange of energy may take place within an open system. Let the system be an automatic engine with the inlet manifold at

the first state point and exhaust pipe as the second point. There would be an interchange of chemical energy in the fuel, kinetic energy of moving particles, internal energy of gas and heat transferred and shaft work within the system. From Fig. 12 it is obvious that if there is no variation of flow of mass or energy with time across the boundaries of the system the steady flow will prevail. The conditions may pass through the cyclic or non-cyclic changes within the system. As a result, the mass entering the system equals the mass leaving, also energy entering the system equals energy leaving.

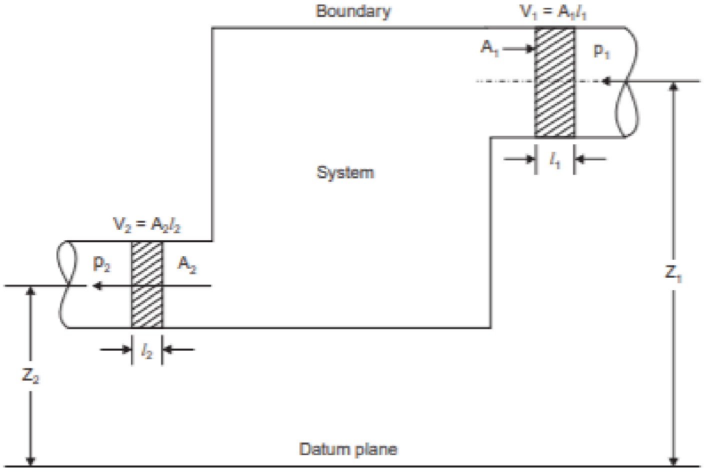


Fig.12. Open system

The steady flow equation can be expressed as follows :

$$u_1 + \frac{C_1^2}{2} + Z_1g + p_1v_1 + Q = u_2 + \frac{C_2^2}{2} + Z_2g + p_2v_2 + W$$

$$(u_1 + p_1v_1) + \frac{C_1^2}{2} + Z_1g + Q = (u_2 + p_2v_2) + \frac{C_2^2}{2} + Z_2g + W$$

$$h_1 + \frac{C_1^2}{2} + Z_1g + Q = h_2 + \frac{C_2^2}{2} + Z_2g + W \qquad [\because$$

If Z_1 and Z_2 are neglected, we get

$$h_1 + \frac{C_1^2}{2} + Q = h_2 + \frac{C_2^2}{2} + W$$

where, Q = Heat supplied (or entering the boundary) per kg of fluid,
W = Work done by (or work coming out of the boundary) 1 kg of fluid,

C = Velocity of fluid , Z = Height above datum, p = Pressure of the fluid, u = Internal energy per kg of fluid, and pv = Energy required for 1 kg of fluid.

This equation is applicable to any medium in any steady flow. It is applicable not only to rotary machines such as centrifugal fans, pumps and compressors but also to reciprocating machines such as steam engines. In a steady flow the rate of mass flow of fluid at any section is the same as at any other section. Consider any section of cross-sectional area A, where the fluid velocity is C, the rate of volume flow past the section is CA. Also, since mass flow is volume flow divided by specific volume,

$$\therefore \qquad \text{Mass flow rate, } \dot{m} = \frac{CA}{v}$$

(where v = *Specific volume* at the section)

This equation is known as the **continuity of mass equation**.

With reference to Fig. 4.30.

$$\therefore \qquad \dot{m} = \frac{C_1A_1}{v_1} = \frac{C_2A_2}{v_2}$$

Problem 6. In a gas turbine unit, the gases flow through the turbine is 15 kg/s and the power developed by the turbine is 12000 kW. The enthalpies of gases at the inlet and outlet are 1260 kJ/kg and 400 kJ/kg respectively, and the velocity of gases at the inlet and outlet are 50 m/s and 110 m/s respectively. Calculate : (i) The rate at which heat is rejected to the turbine, and (ii) The area of the inlet pipe given that the specific volume of the gases at the inlet is 0.45 m3/kg

Solution. Rate of flow of gases, $\dot{m} = 15$ kg/s
Volume of gases at the inlet, $v = 0.45$ m^3/kg
Power developed by the turbine, $P = 12000$ kW

$$\therefore \quad \text{Work done,} \qquad W = \frac{12000}{15} = 800 \text{ kJ/kg}$$

Enthalpy of gases at the inlet, $h_1 = 1260$ kJ/kg
Enthalpy of gases at the oulet, $h_2 = 400$ kJ/kg
Velocity of gases at the inlet, $C_1 = 50$ m/s
Velocity of gases at the outlet, $C_2 = 110$ m/s.

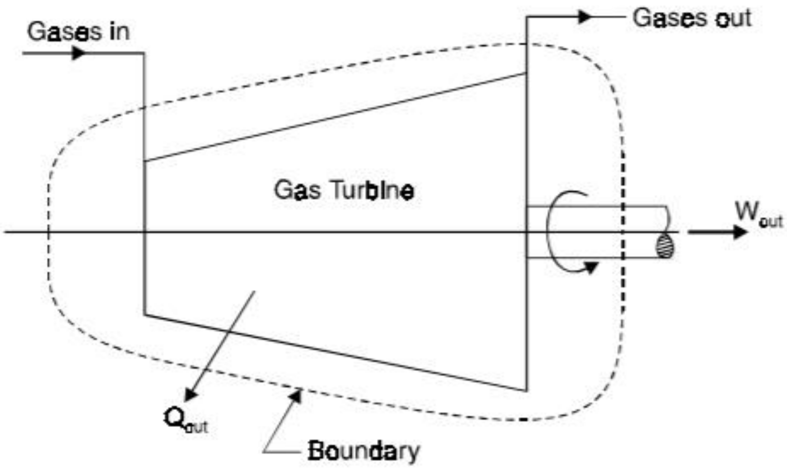


Fig 13. Turbine as a Open system

(*i*) **Heat rejected, Q :**

Using the flow equation,

$$h_1 + \frac{C_1^{\,2}}{2} + Q = h_2 + \frac{C_2^{\,2}}{2} + W \qquad \text{...}(i)\ [\because \quad Z_1 = Z_2]$$

Kinetic energy at inlet $= \frac{C_1^{\,2}}{2} = \frac{50^2}{2}\ \text{m}^2/\text{s}^2 = \frac{50^2\ \text{kg m}^3}{2\ \text{s}^2\text{kg}} = 1250\ \text{Nm/kg} = 1.25\ \text{kJ/kg}$

Kinetic energy at outlet $= \frac{C_1^{\,2}}{2} = \frac{110^2}{2 \times 1000} = 6.05\ \text{kJ/kg}$

Substituting these values in eqn. (*i*), we get

$$1260 + 1.25 + Q = 400 + 6.05 + 800$$

$$\therefore \qquad Q = -\ 55.2\ \text{kJ/kg}$$

Heat rejected $= +\ 55.2\ \text{kJ/kg} = 55.2 \times 15\ \text{kJ/s} =$ **828 kW. (Ans.)**

(*ii*) **Inlet area, A :**

Using the relation,

$$\dot{m} = \frac{CA}{v}$$

$$\therefore \qquad A = \frac{v\dot{m}}{C} = \frac{0.45 \times 15}{50} = \mathbf{0.135\ m^2.}\quad \textbf{(Ans.)}$$

Problem 7. In an air compressor air flows steadily at the rate of 0.5 kg/s through an air compressor. It enters the compressor at 6 m/s with a pressure of 1 bar and a specific volume of 0.85 m3/kg and leaves at 5 m/s with a pressure of 7 bar and a specific volume of 0.16 m3/kg. The internal energy of the air leaving is 90 kJ/kg greater than that of the air entering. Cooling water in a jacket surrounding the cylinder absorbs heat from the air at the rate of 60 kJ/s. Calculate : (i) The power required to drive the compressor ; (ii) The inlet and output pipe cross-sectional areas. Solution. Air flow rate through the compressor, m = 0.5 kg/s Velocity of air at the inlet to compressor, C1 = 6 m/s Velocity of air at the outlet of compressor, C2 = 5 m/s Pressure of air at the inlet to the compressor, p1 = 1 bar

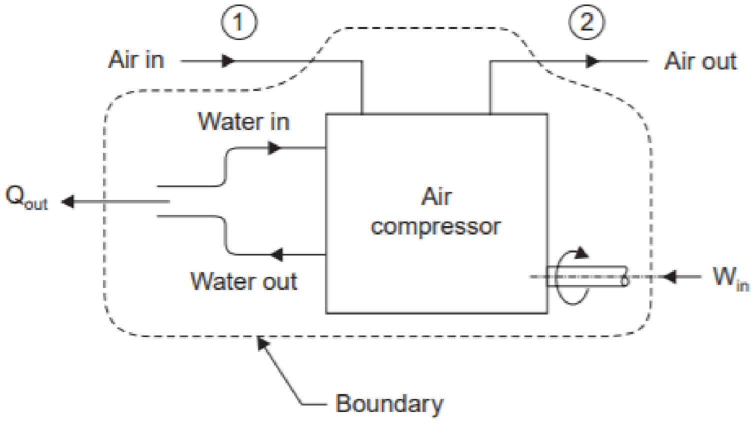
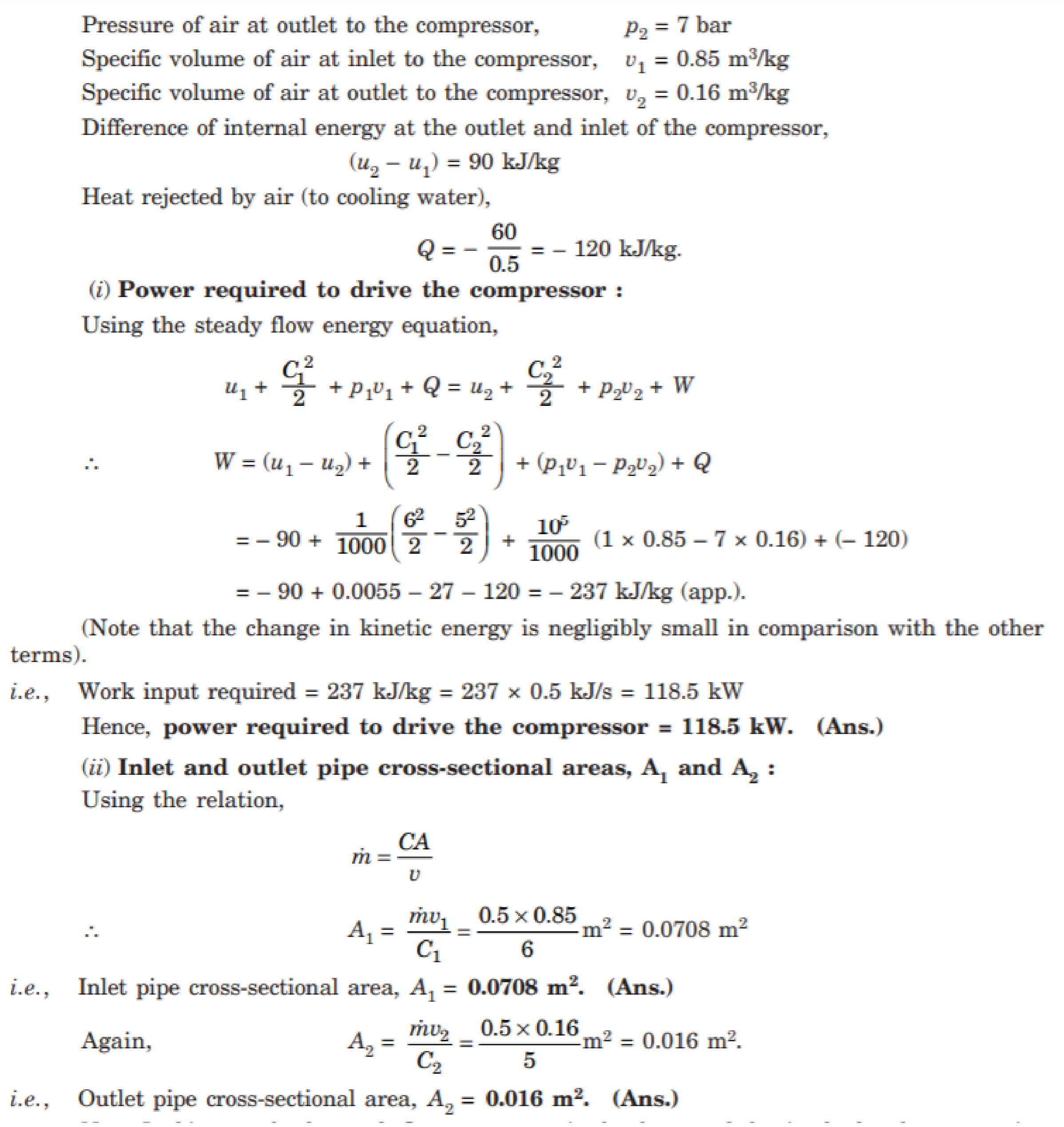


Fig 14. Compressor as a open system



Pressure of air at outlet to the compressor, $p_2 = 7$ bar

Specific volume of air at inlet to the compressor, $v_1 = 0.85$ m^3/kg

Specific volume of air at outlet to the compressor, $v_2 = 0.16$ m^3/kg

Difference of internal energy at the outlet and inlet of the compressor,

$$(u_2 - u_1) = 90 \text{ kJ/kg}$$

Heat rejected by air (to cooling water),

$$Q = -\frac{60}{0.5} = -120 \text{ kJ/kg.}$$

(*i*) **Power required to drive the compressor :**

Using the steady flow energy equation,

$$u_1 + \frac{C_1^2}{2} + p_1v_1 + Q = u_2 + \frac{C_2^2}{2} + p_2v_2 + W$$

$$\therefore \qquad W = (u_1 - u_2) + \left(\frac{C_1^2}{2} - \frac{C_2^2}{2}\right) + (p_1v_1 - p_2v_2) + Q$$

$$= -90 + \frac{1}{1000}\left(\frac{6^2}{2} - \frac{5^2}{2}\right) + \frac{10^5}{1000}(1 \times 0.85 - 7 \times 0.16) + (-120)$$

$$= -90 + 0.0055 - 27 - 120 = -237 \text{ kJ/kg (app.).}$$

(Note that the change in kinetic energy is negligibly small in comparison with the other terms).

i.e., Work input required = 237 kJ/kg = 237 × 0.5 kJ/s = 118.5 kW

Hence, **power required to drive the compressor = 118.5 kW. (Ans.)**

(*ii*) **Inlet and outlet pipe cross-sectional areas, A_1 and A_2 :**

Using the relation,

$$\dot{m} = \frac{CA}{v}$$

$$\therefore \qquad A_1 = \frac{\dot{m}v_1}{C_1} = \frac{0.5 \times 0.85}{6} \text{m}^2 = 0.0708 \text{ m}^2$$

i.e., Inlet pipe cross-sectional area, A_1 = **0.0708 m^2. (Ans.)**

Again, $$A_2 = \frac{\dot{m}v_2}{C_2} = \frac{0.5 \times 0.16}{5} \text{m}^2 = 0.016 \text{ m}^2.$$

i.e., Outlet pipe cross-sectional area, A_2 = **0.016 m^2. (Ans.)**

TEXT / REFERENCE BOOKS

1. Nag P.K., “Engineering Thermodynamics”, Tata McGraw Hill Education, 2009.
2. Yunus A. Cengel, Michael A. Boles, “Thermodynamics: An Engineering Approach”, McGraw Hill Education, 2014.
3. Rajput R.K., “Engineering Thermodynamics”, Laxmi Publications, 2010.
4. Khurmi R.S., Gupta J.K, “Thermal Engineering”, S Chand, 2006.
5. P.L. Ballaney,”Thermal Engineering”, Khanna Publisher, 2005.

SATHYABAMA
INSTITUTE OF SCIENCE AND TECHNOLOGY
(DEEMED TO BE UNIVERSITY)
Accredited "A" Grade by NAAC | 12B Status by UGC | Approved by AICTE
www.sathyabama.ac.in

SCHOOL OF MECHANICAL ENGINEERING

DEPARTMENT OF AERONAUTICAL

UNIT – IV – Aero Engineering Thermodynamics – SAEA1301

IV. AERO-ENGINE OPERATING CYCLES

Gas turbine cycle—Brayton cycle

Ideal Brayton Cycle

Brayton cycle is a constant pressure cycle for a perfect gas. It is also called Joule cycle. The heat transfers are achieved in reversible constant pressure heat exchangers. An ideal gas turbine plant would perform the processes that make up a Brayton cycle. The cycle is shown in the Fig. 1 and it is represented on p-v and T-s diagrams as shown in Figs. 1 (b) and (c).

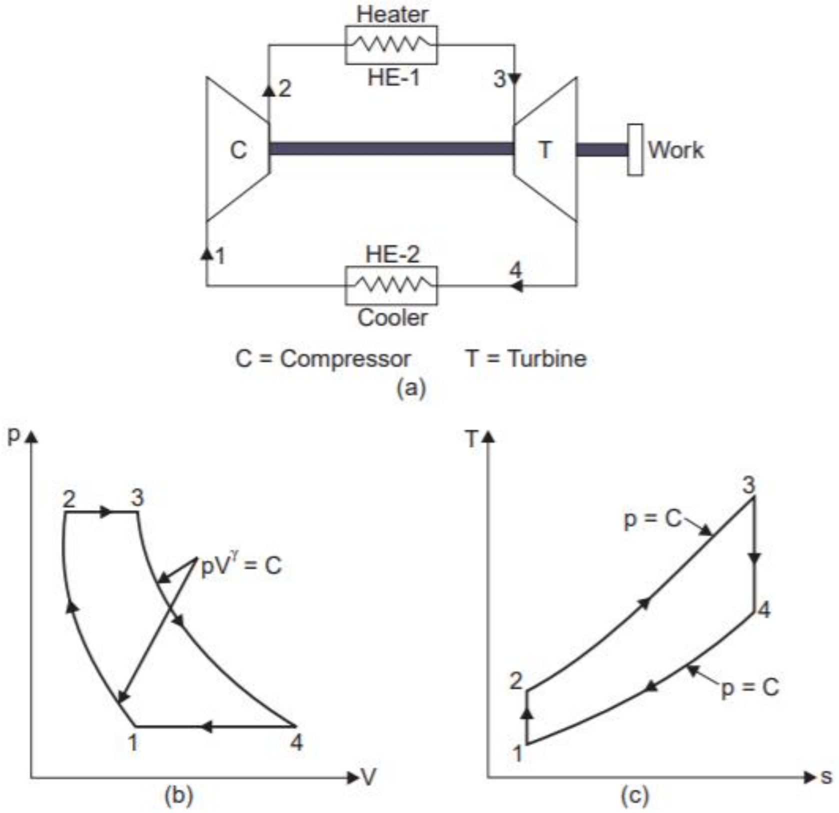


Fig. 1. Brayton cycle : (a) Basic components of a gas turbine power plant (b) p-V diagram (c) T-s diagram

The various operations are as follows :

Operation 1-2. The air is compressed isentropically from the lower pressure p_1 to the upper pressure p_2, the temperature rising from T_1 to T_2. No heat flow occurs.

Operation 2-3. Heat flows into the system increasing the volume from V_2 to V_3 and temperature from T_2 to T_3 whilst the pressure remains constant at p_2. Heat received = mcp (T_3 – T_2). Operation 3-4. The air is expanded isentropically from p_2 to p_1, the temperature falling from T_3 to T_4. No heat flow occurs.

Operation 4-1. Heat is rejected from the system as the volume decreases from V4 to V1 and the temperature from T_4 to T_1 whilst the pressure remains constant at p_1. Heat rejected = mcp (T_4 – T_1).

$$\eta_{\text{air-standard}} = \frac{\text{Work done}}{\text{Heat received}}$$

$$= \frac{\text{Heat received/cycle} - \text{Heat rejected/cycle}}{\text{Heat received/cycle}}$$

$$= \frac{mc_p\,(T_3 - T_2) - mc_p\,(T_4 - T_1)}{mc_p\,(T_3 - T_2)} = 1 - \frac{T_4 - T_1}{T_3 - T_2}$$

Now, from isentropic expansion,

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma - 1}{\gamma}}$$

$$T_2 = T_1\,(r_p)^{\frac{\gamma - 1}{\gamma}}\text{, where } r_p = \text{pressure ratio.}$$

Similarly $$\frac{T_3}{T_4} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma - 1}{\gamma}} \quad \text{or} \quad T_3 = T_4\,(r_p)^{\frac{\gamma - 1}{\gamma}}$$

$$\therefore \quad \eta_{\text{air-standard}} = 1 - \frac{T_4 - T_1}{T_4(r_p)^{\frac{\gamma - 1}{\gamma}} - T_1(r_p)^{\frac{\gamma - 1}{\gamma}}} = 1 - \frac{1}{(r_p)^{\frac{\gamma - 1}{\gamma}}}$$

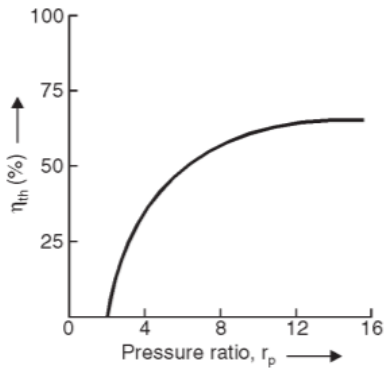


Fig. 2. Effect of pressure ratio on the efficiency of Brayton cycle.

Pressure Ratio for Maximum Work:

Now we shall prove that the pressure ratio for maximum work is a function of the limiting temperature ratio.

Work output during the cycle = Heat received/cycle – heat rejected/cycle

$$= mc_p\,(T_3 - T_2) - mc_p\,(T_4 - T_1)$$
$$= mc_p\,(T_3 - T_4) - mc_p\,(T_2 - T_1)$$
$$= mc_p\ T_3\left(1 - \frac{T_4}{T_3}\right) - T_1\left(\frac{T_2}{T_1} - 1\right)$$

In case of a given turbine the minimum temperature T1 and the maximum temperature T3 are prescribed, T1 being the temperature of the atmosphere and T3 the maximum temperature which the metals of turbine would withstand. Consider the specific heat at constant pressure cp to be constant. Then,

Since, $$\frac{T_3}{T_4} = (r_p)^{\frac{\gamma-1}{\gamma}} = \frac{T_2}{T_1}$$

Using the constant $‘z’ = \dfrac{\gamma-1}{\gamma}$,

we have, work output/cycle

$$W = K\left[T_3\left(1 - \frac{1}{r_p{}^{z}}\right) - T_1\,(r_p{}^{z} - 1)\right]$$

Differentiating with respect to r_p

$$\frac{dW}{dr_p} = K\left[T_3 \times \frac{z}{r_p(z+1)} - T_1\, z r_p{}^{(z-1)}\right] = 0 \text{ for a maximum}$$

$$\therefore \qquad \frac{zT_3}{r_p{}^{(z+1)}} = T_1 z\,(r_p)^{(z-1)}$$

$$\therefore \qquad r_p{}^{2z} = \frac{T_3}{T_1}$$

$$\therefore \qquad r_p = (T_3/T_1)^{1/2z} \quad i.e., \quad r_p = (T_3/T_1)^{\frac{\gamma}{2(\gamma-1)}}$$

Thus, the pressure ratio for maximum work is a function of the limiting temperature ratio.

Work Ratio:

Work ratio is defined as the ratio of net work output to the work done by the turbine

$$\therefore \qquad \text{Work ratio} = \frac{W_T - W_C}{W_T}$$

$$\begin{bmatrix} \text{where,} & W_T = \text{Work obtained from this turbine,} \\ \text{and} & W_C = \text{Work supplied to the compressor.} \end{bmatrix}$$

$$= \frac{mc_p(T_3 - T_4) - mc_p(T_2 - T_1)}{mc_p\,(T_3 - T_4)} = 1 - \frac{T_2 - T_1}{T_3 - T_4}$$

$$= 1 - \frac{T_1}{T_3}\left[\frac{(r_p)^{\frac{\gamma-1}{\gamma}} - 1}{1 - \dfrac{1}{(r_p)^{\frac{\gamma-1}{\gamma}}}}\right] = 1 - \frac{T_1}{T_3}(r_p)^{\frac{\gamma-1}{\gamma}}$$

Open Cycle Gas Turbine—Actual Brayton Cycle:

The fundamental gas turbine unit is one operating on the open cycle in which a rotary compressor and a turbine are mounted on a common shaft. Air is drawn into the compressor and after compression passes to a combustion chamber. Energy is supplied in the combustion chamber by spraying fuel into the air stream, and the resulting hot gases expand through the turbine to the atmosphere. In order to achieve net work output from the unit, the turbine must develop more gross work output than is required to drive the compressor and to overcome mechanical losses in the drive. The products of combustion coming out from the turbine are exhausted to the atmosphere as they cannot be used any more. The working fluids (air and fuel) must be replaced continuously as they are exhausted into the atmosphere.

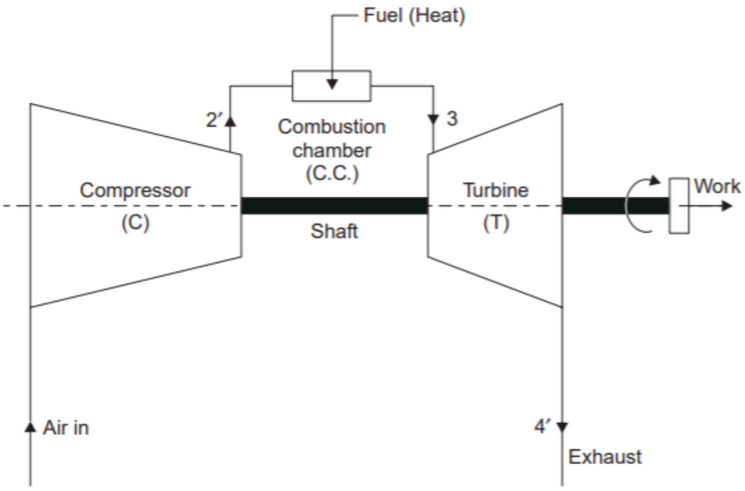


Fig. 3. Open cycle gas turbine.

If pressure loss in the combustion chamber is neglected, this cycle may be drawn on a T-s diagram as shown in Fig. 4.

- 1-2′ represents : irreversible adiabatic compression.
- 2′-3 represents : constant pressure heat supply in the combustion chamber.
- 3-4′ represents : irreversible adiabatic expansion.
- 1-2 represents : ideal isentropic compression.
- 3-4 represents : ideal isentropic expansion.

Assuming change in kinetic energy between the various points in the cycle to be negligibly small compared with enthalpy changes and then applying the flow equation to each part of cycle, for unit mass, we have

Work input (compressor) $= c_p\,(T_2' - T_1)$

Heat supplied (combustion chamber) $= c_p\,(T_3 - T_2')$

Work output (turbine) $= c_p\,(T_3 - T_4')$

$\therefore$ Net work output = Work output – Work input

$= c_p\,(T_3 - T_4') - c_p(T_2' - T_1)$

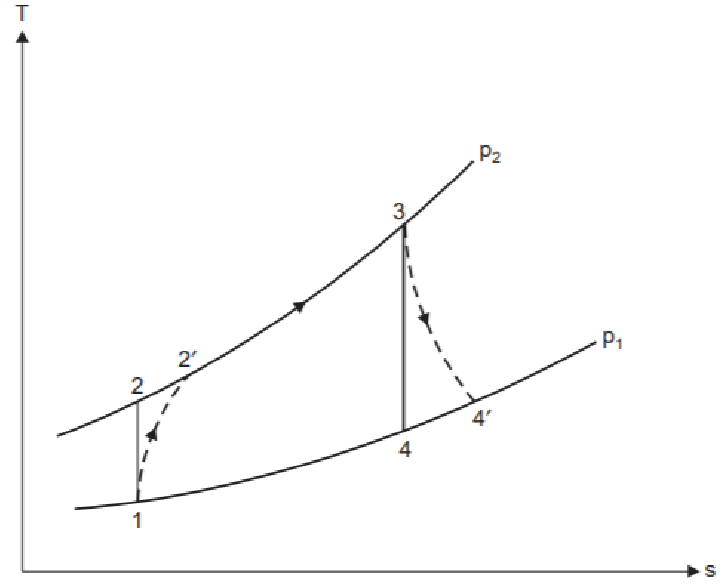


Fig. 4. T-s diagram

and $$\eta_{thermal} = \frac{\text{Net work output}}{\text{Heat supplied}}$$

$$= \frac{c_p(T_3 - T_4') - c_p(T_2' - T_1)}{c_p(T_3 - T_2')}$$

Compressor isentropic efficiency, η_{comp}

$$= \frac{\text{Work input required in isentropic compression}}{\text{Actual work required}}$$

$$= \frac{c_p(T_2 - T_1)}{c_p(T_2' - T_1)} = \frac{T_2 - T_1}{T_2' - T_1}$$

Turbine isentropic efficiency, $\eta_{turbine}$

$$= \frac{\text{Actual work output}}{\text{Isentropic work output}}$$

$$= \frac{c_p(T_3 - T_4')}{c_p(T_3 - T_4)} = \frac{T_3 - T_4'}{T_3 - T_4}$$

Methods for Improvement of Thermal Efficiency of Open Cycle Gas Turbine Plant:

The following methods are employed to increase the specific output and thermal efficiency of the plant : 1. Intercooling 2. Reheating 3. Regeneration.

1. Intercooling. A compressor in a gas turbine cycle utilises the major percentage of power developed by the gas turbine. The work required by the compressor can be reduced by compressing the air in two stages and incorporating an intercooler between the two as shown in Fig. 5.

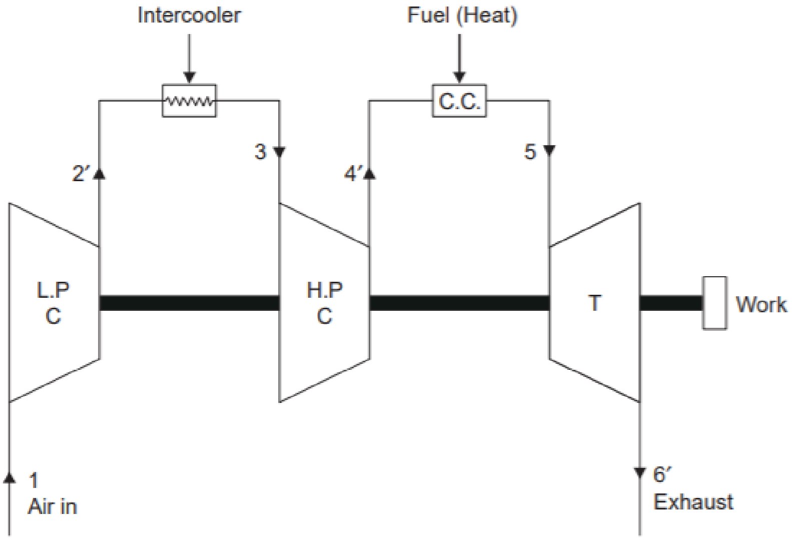


Fig. 5. Intercooling cycle

The corresponding T-s diagram for the unit is shown in Fig. 6. The actual processes take place as follows :

1-2′ ... L.P. (Low pressure) compression

2′-3 ... Intercooling

3-4′ ... H.P. (High pressure) compression

4′-5 ... C.C. (Combustion chamber)-heating

5-6′ ... T (Turbine)-expansion

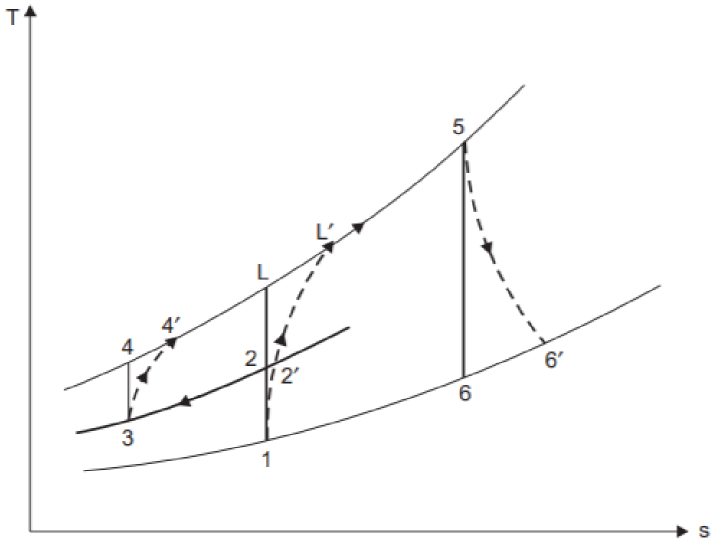


Fig. 6. T-S diagram

The ideal cycle for this arrangement is 1-2-3-4-5-6 ; the compression process without intercooling is shown as 1-L′ in the actual case, and 1-L in the ideal isentropic case.

Now, Work input (with intercooling)

$$= c_p(T_2' - T_1) + c_p(T_4' - T_3)$$

Work input (without intercooling)

$$= c_p(T_L' - T_1) = c_p(T_2' - T_1) + c_p(T_L' - T_2')$$

By comparing equations it can be observed that the work input with intercooling is less than the work input with no intercooling, when cp (T4′ – T3) is less than cp(TL′ – T2′). This is so if it is assumed that isentropic efficiencies of the two compressors, operating separately, are each equal to the isentropic efficiency of the single compressor which would be required if no intercooling were used. Then (T4′ – T3) < (TL′ – T2′) since the pressure lines diverge on the T-s diagram from left to the right.

$$\text{Again, work ratio} \quad = \frac{\text{Net work output}}{\text{Gross work output}}$$

$$= \frac{\text{Work of expansion} - \text{Work of compression}}{\text{Work of expansion}}$$

From this we may conclude that when the compressor work input is reduced then the work ratio is increased. However the heat supplied in the combustion chamber when intercooling is used in the cycle, is given by,

Heat supplied with intercooling = cp(T5 – T4′)

Also the heat supplied when intercooling is not used, with the same maximum cycle temperature T5, is given by

Heat supplied without intercooling = cp (T5 – TL′)

Thus, the heat supplied when intercooling is used is greater than with no intercooling. Although the net work output is increased by intercooling it is found in general that the increase in heat to be supplied causes the thermal efficiency to decrease. When intercooling is used a supply of cooling water must be readily available. The additional bulk of the unit may offset the advantage to be gained by increasing the work ratio.

Reheating. The output of a gas turbine can be amply improved by expanding the gases in two stages with a reheater between the two as shown in Fig. 7. The H.P. turbine drives the compressor and the L.P. turbine provides the useful power output. The corresponding T-s diagram is shown in Fig. 8. The line 4′-L′ represents the expansion in the L.P. turbine if reheating is not employed.

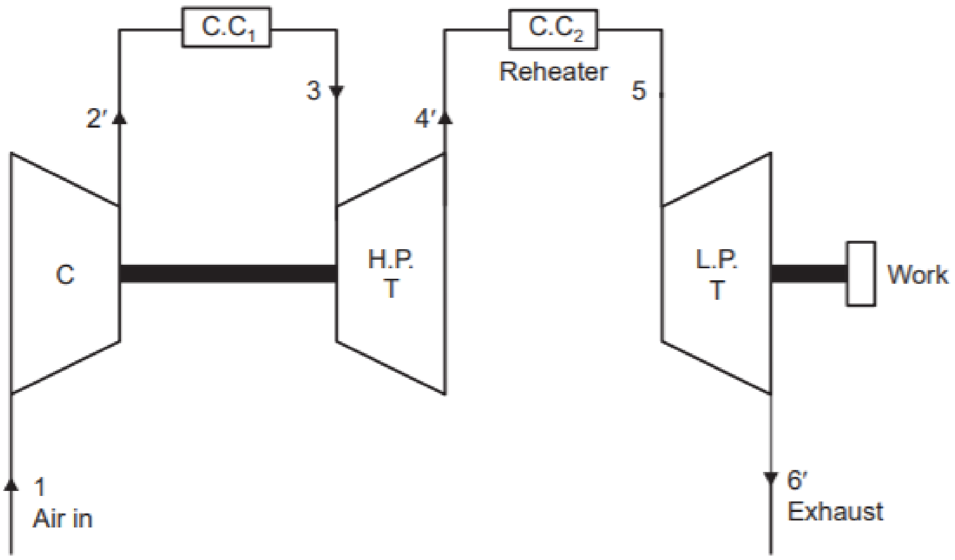


Fig. 7. Reheat cycle

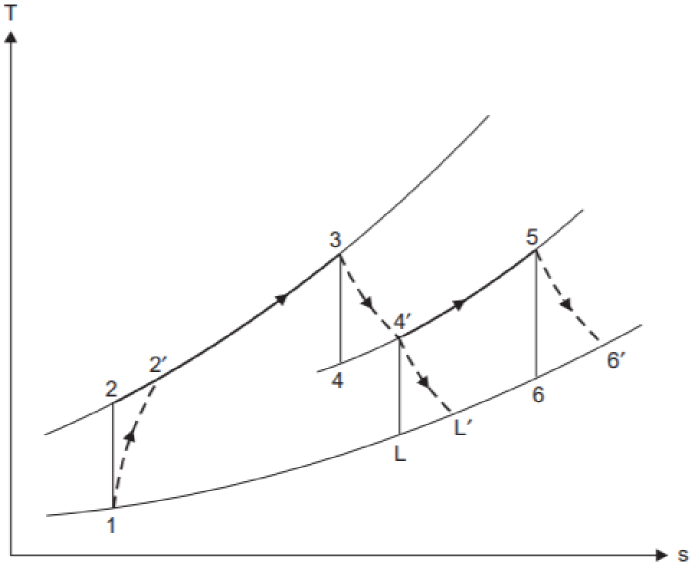


Fig. 8. T-S diagram

Neglecting mechanical losses the work output of the H.P. turbine must be exactly equal to the work input required for the compressor i.e., $c_{pa}(T_2' - T_1) = c_{pg}(T_3 - T_4')$

The work output (net output) of L.P. turbine is given by,

Net work output (with reheating) $= c_{pg}(T_5 - T_6')$

and Net work output (without reheating) $= c_{pg}(T_4' - T_L')$

Since the pressure lines diverge to the right on *T-s* diagram it can be seen that the temperature difference $(T_5 - T_6')$ is always *greater* than $(T_4' - T_L')$, so that *reheating increases the net work output.*

Although net work is increased by reheating the heat to be supplied is also increased, and the net effect can be to reduce the thermal efficiency

$$\text{Heat supplied} = c_{pg}(T_3 - T_2') + c_{pg}(T_5 - T_4').$$

Regeneration. The exhaust gases from a gas turbine carry a large quantity of heat with them since their temperature is far above the ambient temperature. They can be used to heat the air coming from the compressor thereby reducing the mass of fuel supplied in the combustion chamber. Fig. 9 shows a gas turbine plant with a regenerator. The corresponding T-s diagram is shown in Fig. 10. 2′-3 represents the heat flow into the compressed air during its passage through the heat exchanger and 3-4 represents the heat taken in from the combustion of fuel. Point 6 represents the temperature of exhaust gases at discharge from the heat exchanger. The maximum temperature to which the air could be heated in the heat exchanger is ideally that of exhaust gases, but less than this is obtained in practice because a temperature gradient must exist for an unassisted transfer of energy. The effectiveness of the heat exchanger is given by:

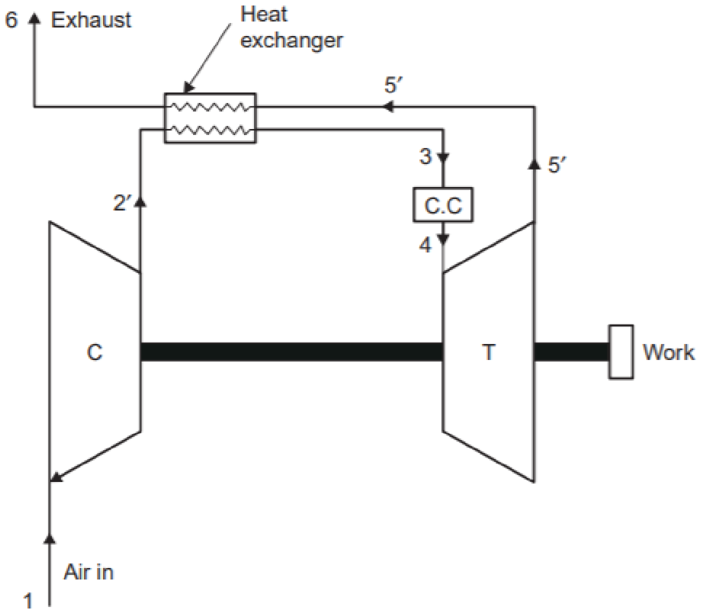


Fig. 9. Gas turbine with regenerator.

Effectiveness,
$$\varepsilon = \frac{\text{Increase in enthalpy per kg of air}}{\text{Available increase in enthalpy per kg of air}}$$
$$= \frac{(T_3 - T_2')}{(T_5' - T_2')}$$

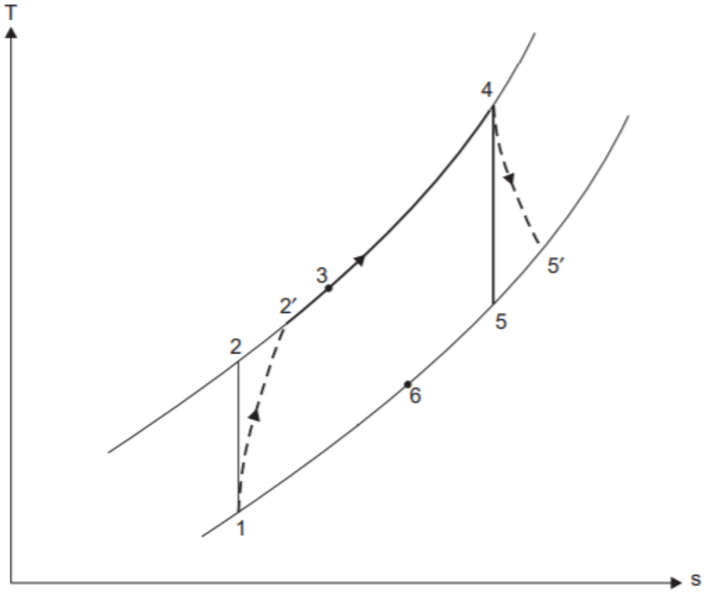


Fig. 10. T-s diagram for the unit

Problem 1: Air enters the compressor of a gas turbine plant operating on Brayton cycle at 101.325 kPa, 27°C. The pressure ratio in the cycle is 6. Calculate the maximum temperature in the cycle and the cycle efficiency. Assume WT = 2.5 WC , where WT and WC are the turbine and the compressor work respectively. Take $\gamma = 1.4$.

Solution.

Pressure of intake air, p1 = 101.325 kPa

Temperature of intake air, T1 = 27 + 273 = 300 K

The pressure ratio in the cycle, rp = 6

(i) Maximum temperature in the cycle, T_3 :

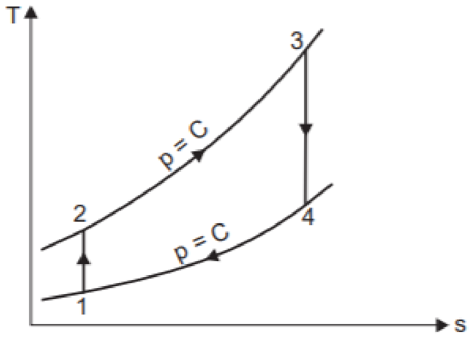


Fig. 10.1

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} = (r_p)^{\frac{\gamma-1}{\gamma}} = (6)^{\frac{1.4-1}{1.4}} = 1.668$$

$\therefore$ $$T_2 = 1.668\ T_1 = 1.668 \times 300 = 500.4 \text{ K}$$

Also, $$\frac{T_3}{T_4} = (r_p)^{\frac{\gamma-1}{\gamma}} = (6)^{\frac{1.4-1}{1.4}} = 1.668$$

$\therefore$ $$T_4 = \frac{T_3}{1.668}$$

But, $$W_T = 2.5\ W_C$$

$\therefore$ $$mc_p\ (T_3 - T_4) = 2.5\ mc_p\ (T_2 - T_1)$$

$$T_3 - \frac{T_3}{1.668} = 2.5\ (500.4 - 300) = 501 \quad \text{or} \quad T_3\left(1 - \frac{1}{1.668}\right) = 501$$

$\therefore$ $$T_3 = \frac{501}{\left(1 - \frac{1}{1.668}\right)} = \mathbf{1251\ K\ or\ 978°C.} \quad \textbf{(Ans.)}$$

(*ii*) **Cycle efficiency, η_{cycle} :**

Now, $$T_4 = \frac{T_3}{1.668} = \frac{1251}{1.668} = 750 \text{ K}$$

$$\eta_{\text{cycle}} = \frac{\text{Net work}}{\text{Heat added}} = \frac{mc_p\ (T_3 - T_4) - mc_p\ (T_2 - T_1)}{mc_p(T_3 - T_2)}$$

$$= \frac{(1251 - 750) - (500.4 - 300)}{(1251 - 500.4)} = \mathbf{0.4 \quad or \quad 40\%.} \quad \textbf{(Ans.)}$$

$$\left[\textit{Check}\ ; \eta_{\text{cycle}} = 1 - \frac{1}{(r_p)^{\left(\frac{\gamma-1}{\gamma}\right)}} = 1 - \frac{1}{(6)^{\frac{1.4-1}{1.4}}} = \mathbf{0.4 \quad or \quad 40\%.} \quad \textbf{(Ans.)}\right]$$

Problem 2: A gas turbine is supplied with gas at 5 bar and 1000 K and expands it adiabatically to 1 bar. The mean specific heat at constant pressure and constant volume are 1.0425 kJ/kg K and 0.7662 kJ/kg K respectively. (i) Draw the temperature-entropy diagram to represent the processes of the simple gas turbine system. (ii) Calculate the power developed in kW per kg of gas per second and the exhaust gas temperature.

Solution.

Given : p1 = 1 bar ; p2 = 5 bar ; T3 = 1000 K ; cp = 1.0425 kJ/kg K ; cv = 0.7662 kJ/kg K

$$\gamma = \frac{c_p}{c_v} = \frac{1.0425}{0.7662} = 1.36$$

(i) Temperature-entropy (T-s) diagram :

Temperature-entropy diagram representing the processes of the simple gas turbine system is shown in Fig.

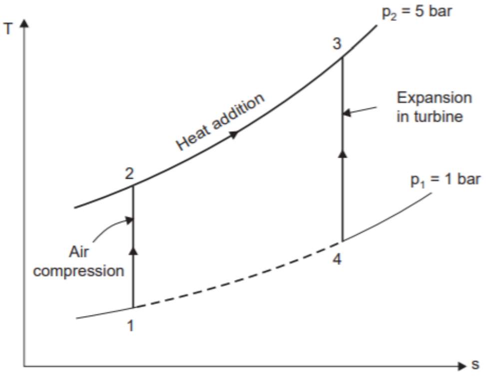


Fig. 10.2

(*ii*) **Power required :**

$$\frac{T_4}{T_3} = \left(\frac{p_1}{p_2}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{1}{5}\right)^{\frac{1.36-1}{1.36}} = 0.653$$

$\therefore$ $T_4 = 1000 \times 0.653 = 653$ K

Power developed per kg of gas per second

$= c_p\,(T_3 - T_4)$

$= 1.0425\,(1000 - 653) =$ **361.7 kW. (Ans.)**

Problem 3: An isentropic air turbine is used to supply 0.1 kg/s of air at 0.1 MN/m2 and at 285 K to a cabin. The pressure at inlet to the turbine is 0.4 MN/m2. Determine the temperature at turbine inlet and the power developed by the turbine. Assume cp= 1.0 kJ/kg K.

Solution. *Given* : $\dot{m}_a = 0.1$ kg/s ; $p_1 = 0.1$ MN/m^2 = 1 bar, $T_4 = 285$ K ; $p_2 = 0.4$ MN/m^2 = 4 bar ; $c_p = 1.0$ kJ/kg K.

Temperature at turbine inlet, T_3 :

$$\frac{T_3}{T_4} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{4}{1}\right)^{\frac{1.4-1}{1.4}} = 1.486$$

$\therefore$ $T_3 = 285 \times 1.486 =$ **423.5 K. (Ans.)**

Power developed, P :

$P = \dot{m}_a c_p (T_3 - T_4)$

$= 0.1 \times 1.0\,(423.5 - 285)$

= **13.85 kW. (Ans.)**

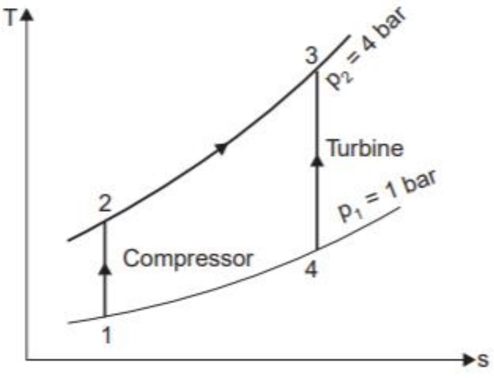
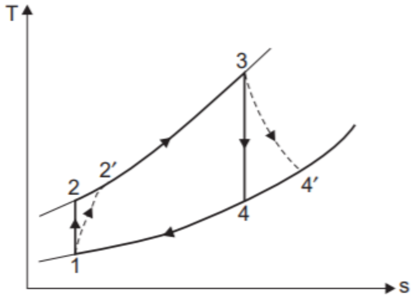


Fig. 10.3

Problem 4: In a gas turbine plant working on Brayton cycle, the air at inlet is 27°C, 0.1 MPa. The pressure ratio is 6.25 and the maximum temperature is 800°C. The turbine and compressor efficiencies are each 80%. Find compressor work, turbine work, heat supplied, cycle efficiency and turbine exhaust temperature. Mass of air may be considered as 1 kg. Draw T-s diagram.

Solution:



T-s diagram

Fig. 10.4

Given : T1 = 27 + 273 = 300 K ; p1 = 0.1 MPa ; rp = 6.25, T3 = 800 + 273 = 1073 K ; $\eta_{comp.} = \eta_{turbine} = 0.8$.

For the *compression process 1-2*, we have

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} = (r_p)^{\frac{\gamma-1}{\gamma}} = (6.25)^{\frac{1.4-1}{1.4}} = 1.688$$

or $$T_2 = 300 \times 1.688 = 506.4 \text{ K}$$

Also, $$\eta_{comp.} = \frac{T_2 - T_1}{T_2' - T_1} \quad \text{or} \quad 0.8 = \frac{506.4 - 300}{T_2' - 300}$$

or $$T_2' = \frac{506.4 - 300}{0.8} + 300 = 558 \text{ K}$$

∴ **Compressor work,** $W_{comp.} = 1 \times c_p \times (T_2' - T_1)$

$= 1 \times 1.005\ (558 - 300) =$ **259.29 kJ/kg. (Ans.)**

For *expansion process 3-4*, we have

$$\frac{T_3}{T_4} = \left(\frac{p_3}{p_4}\right)^{\frac{\gamma-1}{\gamma}} = (r_p)^{\frac{\gamma-1}{\gamma}} = (6.25)^{\frac{1.4-1}{1.4}} = 1.688$$

or $$T_4 = \frac{T_3}{1.688} = \frac{1073}{1.688} = 635.66 \text{ K}$$

Also, $$\eta_{turbine} = \frac{T_3 - T_4'}{T_3 - T_4} \quad \text{or} \quad 0.8 = \frac{1073 - T_4'}{1073 - 635.66}$$

or $$T_4' = 1073 - 0.8\ (1073 - 635.66) = 723.13 \text{ K}$$

∴ **Turbine work,** $W_{turbine} = 1 \times c_p \times (T_3 - T_4')$ (neglecting fuel mass)

$= 1 \times 1.005\ (1073 - 723.13) =$ **351.6 kJ/kg. (Ans.)**

Net work output, $W_{net} = W_{turbine} - W_{comp.} = 351.6 - 259.29 = 92.31$ kJ/kg

Heat supplied, $Q_s = 1 \times c_p \times (T_3 - T_2')$

$= 1 \times 1.005 \times (1073 - 558) =$ **517.57 kJ/kg. (Ans.)**

Cycle efficiency, $$\eta_{cycle} = \frac{W_{net}}{Q_s} = \frac{92.31}{517.57} = 0.1783 \text{ or } \mathbf{17.83\%}. \quad \textbf{(Ans.)}$$

Turbine exhaust temperature, $T_4' = 723.13$ K or **450.13°C. (Ans.)**

Problem 5: A gas turbine unit receives air at 1 bar and 300 K and compresses it adiabatically to 6.2 bar. The compressor efficiency is 88%. The fuel has a heating valve of 44186 kJ/kg and the fuel-air ratio is 0.017 kJ/kg of air. The turbine internal efficiency is 90%. Calculate the work of turbine and compressor per kg of air compressed and thermal efficiency
For products of combustion, cp = 1.147 kJ/kg K and γ = 1.333.

Solution.

Given : p1 (= p4) = 1 bar, T1 = 300 K ; p2(= p3) = 6.2 bar ; $\eta_{\text{compressor}}$ = 88% ; C = 44186 kJ/kg ; Fuel-air ratio = 0.017 kJ/kg of air, ηturbine = 90% ; cp = 1.147 kJ/kg K ; γ = 1.333.

For isentropic compression process 1-2 :

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{6.2}{1}\right)^{\frac{1.4-1}{1.4}} = 1.684$$

$\therefore \quad T_2 = 300 \times 1.684 = 505.2$ K

Now, $\eta_{\text{compressor}} = \dfrac{T_2 - T_1}{T_2' - T_1}$

$$0.88 = \frac{505.2 - 300}{T_2' - 300}$$

$$T_2' = \left(\frac{505.2 - 300}{0.88} + 300\right)$$

$= 533.2$ K

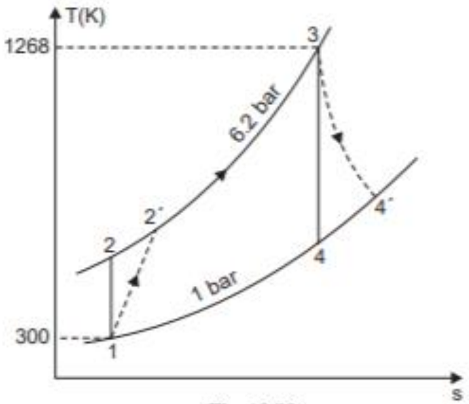


Fig. 13.58

Heat supplied $= (m_a + m_f) \times c_p(T_3 - T_2') = m_f \times C$

or $$\left(1 + \frac{m_f}{m_a}\right) \times c_p(T_3 - T_2') = \frac{m_f}{m_a} \times C$$

or $$(1 + 0.017) \times 1.005(T_3 - 533.2) = 0.017 \times 44186$$

$$\therefore \quad T_3 = \frac{0.017 \times 44186}{(1 + 0.017) \times 1.005} + 533.2 = 1268 \text{ K}$$

For isentropic expression process 3-4 :

$$\frac{T_4}{T_3} = \left(\frac{p_4}{p_3}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{1}{6.2}\right)^{\frac{1.333-1}{1.333}} = 0.634$$

$\therefore \quad T_4 = 1268 \times 0.634 = 803.9$ K $\quad$ ($\because$ $\gamma_g = 1.333$ ......*Given*)

Now, $$\eta_{\text{turbine}} = \frac{T_3 - T_4'}{T_3 - T_4}$$

$$0.9 = \frac{1268 - T_4'}{1268 - 803.9}$$

$\therefore \quad T_4' = 1268 - 0.9(1268 - 803.9) = 850.3$ K

$$W_{\text{compressor}} = c_p(T_2' - T_1) = 1.005(533.2 - 300) = 234.4 \text{ kJ/kg}$$

$$W_{\text{turbine}} = c_{pg}(T_3 - T_4') = 1.147(1268 - 850.3) = 479.1 \text{ kJ/kg}$$

$$\text{Net work} = W_{\text{turbine}} - W_{\text{compressor}}$$

$$= 479.1 - 234.4 = 244.7 \text{ kJ/kg}$$

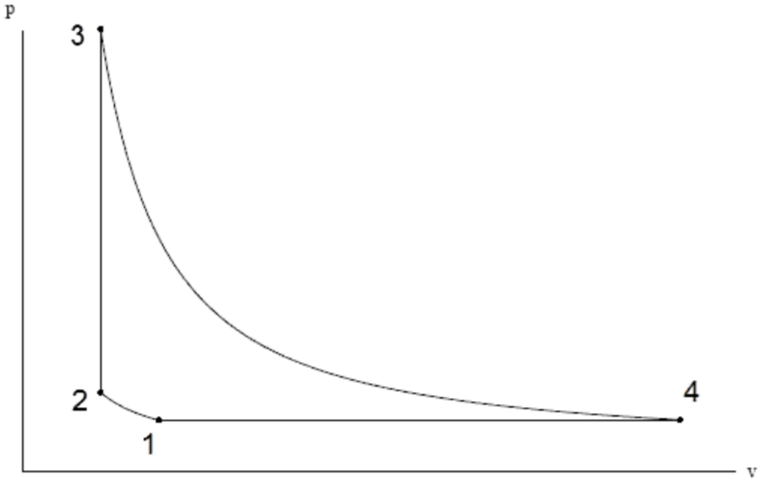
Heat supplied per kg of air $= 0.017 \times 44186 = 751.2$ kJ/kg

$\therefore$ Thermal efficiency, $\eta_{\text{th.}} = \dfrac{\text{Net work}}{\text{Heat supplied}}$

$$= \frac{244.7}{751.2} = 0.3257 \text{ or } \mathbf{32.57\%}. \quad \textbf{(Ans.)}$$

Humpery Cycle:

The Humphrey cycle is a thermodynamic cycle similar to the pulse detonation engine and pulse compression detonation system cycles. It may be considered to be a modification of the Brayton cycle in which the constant-pressure heat addition process of the Brayton cycle is replaced by a constant-volume heat addition process.



p–v diagram of a Humphrey cycle.

Fig.11. P-v Diagram of Humphery cycle

Hence, the ideal Humphrey cycle consists of 4 processes:

1. Reversible, adiabatic (isentropic) compression of the incoming gas. During this step incoming gas is compressed, usually by turbomachinery. Stagnation pressure and temperature increase because of the work done on the gas by the compressor. Entropy is unchanged. Static pressure and density of the gas increase.
2. Constant-volume heat addition. In this step, heat is added while the gas is kept at constant volume. In most cases, Humphrey-cycle engines are considered open cycles (meaning that air flows through continuously) which makes it difficult to have a "constant volume" during the addition of heat. Hence, instead of a deflagrative flame, which is commonly used in Brayton cycles (constant pressure heat addition), the combustion mode is detonative. In being detonative, the heat addition happens only for a small sector volume of premixture (in a combustor annulus) at a constant volume, while the remaining sections refill the chamber with fresh incoming premixture. This

method allows for continuous flow in the system while at the same time achieving the pseudo constant volume requirement for the heat addition process.

3. Reversible, adiabatic (isentropic) expansion of the gas. During this step incoming gas is expanded, usually by turbomachinery. Stagnation pressure and temperature decrease because of the work extracted from the gas by the turbine. Entropy is unchanged. Static pressure and density of the gas decrease.
4. Constant-pressure heat rejection. In this step, heat is removed from the working fluid while the fluid remains at constant pressure. In open-cycle engines this process usually represents expulsion of the gas from the engine, where it quickly equalizes to ambient pressure and slowly loses heat to the atmosphere, which is considered to be an infinitely large reservoir for heat storage, with constant pressure and temperature.

TEXT / REFERENCE BOOKS

1. Nag P.K., “Engineering Thermodynamics”, Tata McGraw Hill Education, 2009.
2. Yunus A. Cengel, Michael A. Boles, “Thermodynamics: An Engineering Approach”, McGraw Hill Education, 2014.
3. Rajput R.K., “Engineering Thermodynamics”, Laxmi Publications, 2010.
4. Khurmi R.S., Gupta J.K, “Thermal Engineering”, S Chand, 2006.
5. P.L. Ballaney,”Thermal Engineering”, Khanna Publisher, 2005.

SATHYABAMA
INSTITUTE OF SCIENCE AND TECHNOLOGY
(DEEMED TO BE UNIVERSITY)
Accredited "A" Grade by NAAC | 12B Status by UGC | Approved by AICTE
www.sathyabama.ac.in

SCHOOL OF MECHANICAL ENGINEERING

DEPARTMENT OF AERONAUTICAL

UNIT – V – Aero Engineering Thermodynamics – SAEA1301

V. AERO ENGINE COMBUSTION & HEAT TRANSFER

Introduction

Fuel may be chemical or nuclear. Here we shall consider briefly chemical fuels only. A chemical fuel is a substance which releases heat energy on combustion. The principal combustible elements of each fuel are carbon and hydrogen. Though sulphur is a combustible element too but its presence in the fuel is considered to be undesirable.

In chemical thermodynamics the study of systems involving chemical reactions is an important topic. A chemical reaction may be defined as the rearrangement of atoms due to redistribution of electrons. In a chemical reaction the terms, reactants and the products are frequently used. ‘Reactants’ comprise of initial constituents which start the reaction while ‘products’ comprise of final constituents which are formed by the chemical reaction. Although the basic principles which will be discussed in this chapter apply to any chemical reaction, here main attention will be focused on an important type of chemical reaction—combustion.

Classification of fuels:

Fuels can be classified according to whether : 1. They occur in nature called primary fuels or are prepared called secondary fuels ; 2. They are in solid, liquid or gaseous state.

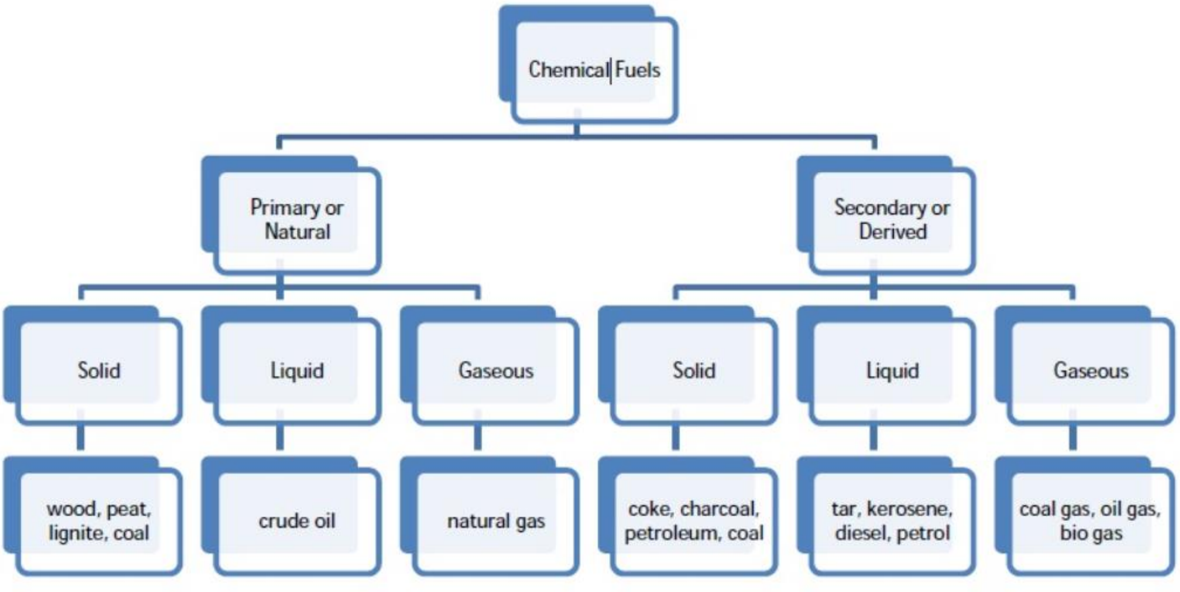


Fig.1. Classification of fuels

Solid fuels

Coal. Its main constituents are carbon, hydrogen, oxygen, nitrogen, sulphur, moisture and ash. Coal passes through different stages during its formation from vegetation. These stages are enumerated and discussed below :

Plant debris—Peat—Lignite—Brown coal—sub-bituminous coal—Bituminous coal—Semibituminous coal—Semi-anthracite coal—Anthracite coal—Graphite.

Peat. It is the first stage in the formation of coal from wood. It contains huge amount of moisture and therefore it is dried for about 1 to 2 months before it is put to use. It is used as a domestic fuel in Europe and for power generation in Russia. In India it does not come in the categories of good fuels.

Lignites and brown coals. These are intermediate stages between peat and coal. They have a woody or often a clay like appearance associated with high moisture, high ash and low heat contents. Lignites are usually amorphous in character and impose transport difficulties as they break easily. They burn with a smoky flame. Some of this type are suitable for local use only.

Bituminous coal. It burns with long yellow and smoky flames and has high percentages of volatile matter. The average calorific value of bituminous coal is about 31350 kJ/kg. It may be of two types, namely caking or noncaking.

Semi-bituminous coal. It is softer than the anthracite. It burns with a very small amount of smoke. It contains 15 to 20 per cent volatile matter and has a tendency to break into small sizes during storage or transportation.

Semi-anthracite. It has less fixed carbon and less lustre as compared to true anthracite and gives out longer and more luminous flames when burnt.

Anthracite. It is very hard coal and has a shining black lustre. It ignites slowly unless the furnace temperature is high. It is non-caking and has high percentage of fixed carbon. It burns either with very short blue flames or without flames. The calorific value of this fuel is high to the tune of 35500 kJ/kg and as such is very suitable for steam generation.

Wood charcoal. It is obtained by destructive distillation of wood. During the process the volatile matter and water are expelled. The physical properties of the residue (charcoal), however depends upon the rate of heating and temperature.

Coke. It consists of carbon, mineral matter with about 2% sulphur and small quantities of hydrogen, nitrogen and phosphorus. It is solid residue left after the destructive distillation of certain kinds of coals. It is smokeless and clear fuel and can be produced by several processes. It is mainly used in blast furnace to produce heat and at the same time to reduce the iron ore.

Briquettes. These are prepared from fine coal or coke by compressing the material under high pressure.

Liquid fuels:

The chief source of liquid fuels is petroleum which is obtained from wells under the earth's crust. These fuels have proved more advantageous in comparison to sold fuels in the following respects.

Advantages : 1. Require less space for storage. 2. Higher calorific value. 3. Easy control of consumption. 4. Staff economy. 5. Absence of danger from spontaneous combustion. 6. Easy handling and transportation. 7. Cleanliness. 8. No ash problem. 9. Non-deterioration of the oil in storage.

Petroleum. There are different opinions regarding the origin of petroleum. However, now it is accepted that petroleum has originated probably from organic matter like fish and plant life etc., by bacterial action or by their distillation under pressure and heat. It consists of a mixture of gases, liquids and solid hydrocarbons with small amounts of nitrogen and sulphur compounds. In India, the main sources of Petroleum are Assam and Gujarat. Heavy fuel oil or crude oil is imported and then refined at different refineries. The refining of crude oil supplies the most important product called petrol. Petrol can also be made by polymerization of refinery gases. Other liquid fuels are kerosene, fuels oils, colloidal fuels and alcohol.

Gaseous fuels:

Natural gas. The main constituents of natural gas are methane ($CH4$) and ethane ($C2H6$). It has calorific value nearly 21000 kJ/m3. Natural gas is used alternately or simultaneously with oil for internal combustion engines.

Coal gas. Mainly consists of hydrogen, carbon monoxide and hydrocarbons. It is prepared by carbonisation of coal. It finds its use in boilers and sometimes used for commercial purposes.

Coke-oven gas. It is obtained during the production of coke by heating the bituminous coal. The volatile content of coal is driven off by heating and major portion of this gas is utilised in heating the ovens. This gas must be thoroughly filtered before using in gas engines.

Blast furnance gas. It is obtained from smelting operation in which air is forced through layers of coke and iron ore, the example being that of pig iron manufacture where this gas is produced as by product and contains about 20% carbon monoxide (CO). After filtering it may be blended with richer gas or used in gas engines directly. The heating value of this gas is very low.

Producer gas. It results from the partial oxidation of coal, coke or peat when they are burnt with an insufficient quantity of air. It is produced in specially designed retorts. It has low heating value and in general is suitable for large installations. It is also used in steel industry for firing open hearth furnaces.

BASIC CHEMISTRY

Before considering combustion problems it is necessary to understand the construction and use of chemical formulae. This involves elementary concepts which are discussed below briefly.

Atoms. It is not possible to divide the chemical elements indefinitely, and the smallest particle which can take part in a chemical change is called an ‘atom’. If an atom is split as in nuclear reaction, the divided atom does not retain the original chemical properties.

Molecules. It is rare to find elements to exist naturally as single atom. Some elements have atoms which exist in pairs, each pair forming a molecule (e.g. oxygen), and the atoms of each molecule are held together by stronger inter-atomic forces. The isolation of a molecule of oxygen would be tedious, but possible ; the isolation of an atom of oxygen would be a different prospect. The molecules of some substances are formed by the mating up of atoms of different elements. For example, water has a molecule which consists of two atoms of hydrogen and one atom of oxygen. The atoms of different elements have different masses and these values are important when a quantitative analysis is required. The actual masses are infinitesimally small, and the ratios of the masses of atoms are used. These ratios are indicated by atomic weight quoted on a scale which defines the atomic weight of oxygen as 16.

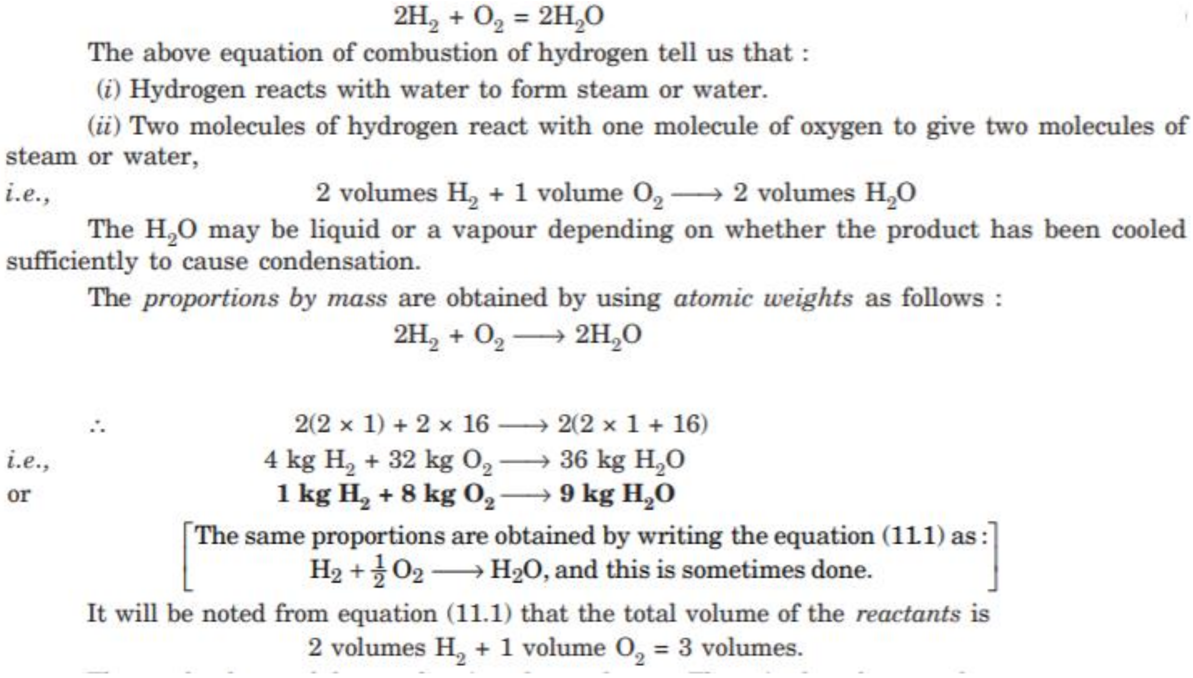
COMBUSTION EQUATIONS:

In a combustion chamber proportionate masses of air and fuel enter where the chemical reaction takes place, and then the combustion products pass to the exhaust. By the conservation of mass the mass flow remains constant (i.e., total mass of products = total mass of reactants), but the reactants are chemically different from the products, and the products leave at a higher temperature. The total number of atoms of each element concerned in the combustion remains constant, but the atoms are rearranged into groups having different chemical properties. This information is expressed in the chemical equation which shows (i) the reactants and the products of combustion, (ii) the relative quantities of the reactants and products. The two sides of the equation must be consistent, each having the same number of atoms of each element involved.

The oxygen supplied for combustion is usually provided by atmospheric air, and it is necessary to use accurate and consistent analysis of air by mass and by volume. It is usual in combustion calculations to take air as 23.3% O2, 76.7% N2 by mass, and 21% O2, 79% N2 by volume. The small traces of other gases in dry air are included in nitrogen, which is sometimes called ‘atmospheric nitrogen’.

Some important combustion equations are given below :

Combustion of hydrogen



$$2H_2 + O_2 = 2H_2O$$

The above equation of combustion of hydrogen tell us that :

(*i*) Hydrogen reacts with water to form steam or water.

(*ii*) Two molecules of hydrogen react with one molecule of oxygen to give two molecules of steam or water,

i.e., $$2 \text{ volumes } H_2 + 1 \text{ volume } O_2 \longrightarrow 2 \text{ volumes } H_2O$$

The H_2O may be liquid or a vapour depending on whether the product has been cooled sufficiently to cause condensation.

The *proportions by mass* are obtained by using *atomic weights* as follows :

$$2H_2 + O_2 \longrightarrow 2H_2O$$

$$\therefore \quad 2(2 \times 1) + 2 \times 16 \longrightarrow 2(2 \times 1 + 16)$$

i.e., $$4 \text{ kg } H_2 + 32 \text{ kg } O_2 \longrightarrow 36 \text{ kg } H_2O$$

or $$\mathbf{1 \text{ kg } H_2 + 8 \text{ kg } O_2 \longrightarrow 9 \text{ kg } H_2O}$$

[The same proportions are obtained by writing the equation (11.1) as : $H_2 + \frac{1}{2} O_2 \longrightarrow H_2O$, and this is sometimes done.]

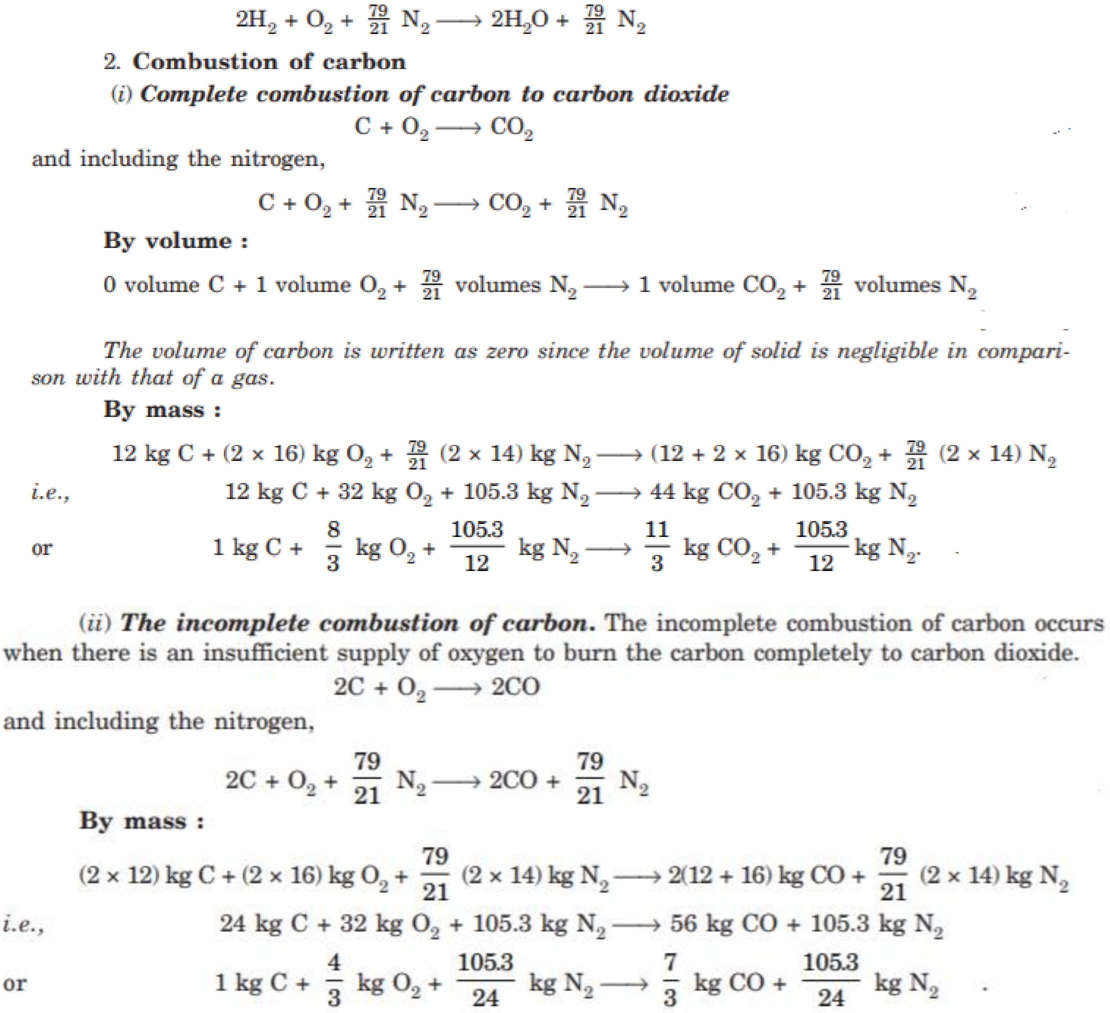
It will be noted from equation (11.1) that the total volume of the *reactants* is

$$2 \text{ volumes } H_2 + 1 \text{ volume } O_2 = 3 \text{ volumes.}$$

The total volume of the product is only 2 volumes. There is therefore a volumetric contraction on combustion.

Since the oxygen is accompanied by nitrogen if air is supplied for the combustion, then this nitrogen should be included in the equation. As nitrogen is inert as far as chemical reaction is concerned, it will appear on both sides of the equation.

With one mole of oxygen there are 79/21 moles of nitrogen, hence equation becomes,



$$2H_2 + O_2 + \tfrac{79}{21} N_2 \longrightarrow 2H_2O + \tfrac{79}{21} N_2$$

2. **Combustion of carbon**

(*i*) ***Complete combustion of carbon to carbon dioxide***

$$C + O_2 \longrightarrow CO_2$$

and including the nitrogen,

$$C + O_2 + \tfrac{79}{21} N_2 \longrightarrow CO_2 + \tfrac{79}{21} N_2$$

By volume :

$$0 \text{ volume C} + 1 \text{ volume } O_2 + \tfrac{79}{21} \text{ volumes } N_2 \longrightarrow 1 \text{ volume } CO_2 + \tfrac{79}{21} \text{ volumes } N_2$$

The volume of carbon is written as zero since the volume of solid is negligible in comparison with that of a gas.

By mass :

$$12 \text{ kg C} + (2 \times 16) \text{ kg } O_2 + \tfrac{79}{21} (2 \times 14) \text{ kg } N_2 \longrightarrow (12 + 2 \times 16) \text{ kg } CO_2 + \tfrac{79}{21} (2 \times 14) N_2$$

i.e., $$12 \text{ kg C} + 32 \text{ kg } O_2 + 105.3 \text{ kg } N_2 \longrightarrow 44 \text{ kg } CO_2 + 105.3 \text{ kg } N_2$$

or $$1 \text{ kg C} + \frac{8}{3} \text{ kg } O_2 + \frac{105.3}{12} \text{ kg } N_2 \longrightarrow \frac{11}{3} \text{ kg } CO_2 + \frac{105.3}{12} \text{ kg } N_2.$$

(*ii*) ***The incomplete combustion of carbon.*** The incomplete combustion of carbon occurs when there is an insufficient supply of oxygen to burn the carbon completely to carbon dioxide.

$$2C + O_2 \longrightarrow 2CO$$

and including the nitrogen,

$$2C + O_2 + \frac{79}{21} N_2 \longrightarrow 2CO + \frac{79}{21} N_2$$

By mass :

$$(2 \times 12) \text{ kg C} + (2 \times 16) \text{ kg } O_2 + \frac{79}{21} (2 \times 14) \text{ kg } N_2 \longrightarrow 2(12 + 16) \text{ kg CO} + \frac{79}{21} (2 \times 14) \text{ kg } N_2$$

i.e., $$24 \text{ kg C} + 32 \text{ kg } O_2 + 105.3 \text{ kg } N_2 \longrightarrow 56 \text{ kg CO} + 105.3 \text{ kg } N_2$$

or $$1 \text{ kg C} + \frac{4}{3} \text{ kg } O_2 + \frac{105.3}{24} \text{ kg } N_2 \longrightarrow \frac{7}{3} \text{ kg CO} + \frac{105.3}{24} \text{ kg } N_2$$

If a further supply of oxygen is available then the combustion can continue to completion,

$$2CO + O_2 + \tfrac{79}{21}\ N_2 \longrightarrow 2CO_2 + \tfrac{79}{21}\ N_2$$

By mass :

$$56\text{ kg } CO + 32\text{ kg } O_2 + \frac{79 \times 28}{21}\text{ kg } N_2 \longrightarrow 88\text{ kg } CO_2 + \frac{79 \times 28}{21}\text{ kg } N_2$$

or

$$1\text{ kg } CO + \frac{4}{7}\text{ kg } O_2 + \frac{105.3}{56}\text{ kg } N_2 \longrightarrow \frac{11}{7}\text{ kg } CO_2 + \frac{105.3}{56}\text{ kg } N_2.$$

THEORETICAL AIR AND EXCESS AIR

The minimum amount of air that supplies sufficient oxygen for the complete combustion of all the carbon, hydrogen, and any other elements in the fuel that may oxidise is called the “theoretical air”. When complete combustion is achieved with theoretical air, the products contain no oxygen. In practice, it is found that complete combustion is not likely to be achieved unless the amount of air supplied is somewhat greater than the theoretical amount. Thus 150 per cent theoretical air means that air actually supplied is 1.5 times the theoretical air. The complete combustion of methane with minimum amount of theoretical air and 150 per cent theoretical air respectively is written as :

$$CH_4 + 2O_2 + 2\left(\frac{79}{21}\right) N_2 \longrightarrow CO_2 + 2H_2O + 2\left(\frac{79}{21}\right) N_2$$

$$CH_4 + 2(1.5)\ O_2 + 2\left(\frac{79}{21}\right)(1.5)\ N_2 \longrightarrow CO_2 + 2H_2O + O_2 + 3\left(\frac{29}{21}\right) N_2$$

The amount of air actually supplied may also be expressed in terms of per cent excess air. The excess air is the amount of air supplied over and above the theoretical air. Thus 150 per cent theoretical air is equivalent to 50 per cent excess air.

STOICHIOMETRIC AIR FUEL (A/F) RATIO

Stoichiometric (or chemically correct) mixture of air and fuel is one that contains just sufficient oxygen for complete combustion of the fuel. A weak mixture is one which has an excess of air. A rich mixture is one which has a deficiency of air. The percentage of excess air is given as under :

$$\%\text{age excess air} = \frac{\text{Actural A/F ratio} - \text{Stoichiometric A/F ratio}}{\text{Stoichiometric A/F ratio}}$$

HOW TO CONVERT VOLUMETRIC ANALYSIS TO WEIGHT ANALYSIS ?

The conversion of volumetric analysis to weight analysis involves the following steps : 1. Multiply the volume of each constituent by its molecular weight. 2. Add all these weights and then divide each weight by the total of all and express it as percentage.

HOW TO CONVERT WEIGHT ANALYSIS TO VOLUMETRIC ANALYSIS ?

1. Divide the weight of each constituent by its molecular weight. 2. Add up these volumes and divide each volume by the total of all and express it as a percentage

WEIGHT OF CARBON IN FLUE GASES

The weight of carbon contained in one kg of flue or exhaust gas can be calculated from the amounts of CO2 and CO contained in it.

Therefore, weight of carbon per kg of fuel

$$= \left(\frac{3}{11}CO_2 + \frac{3}{7}CO\right)$$

WEIGHT OF FLUE GASES PER KG OF FUEL BURNT

Due to supply of air, the weight of flue gas or exhaust gas is always more than that of fuel burnt. The actual weight of dry flue gases can be obtained by comparing the weight of carbon present in the flue gases with the weight of carbon in the fuel, since there is no loss of carbon during the combustion process. As the analysis of the exhaust gases is volumetric, so this must first be reduced to weight analysis. Also, total weight of carbon in one kg of flue gas is

$$= \left(\frac{3}{11}CO_2 + \frac{3}{7}CO\right)$$

∴ The *weight of flue gas/kg of fuel burnt*

$$= \frac{\text{Weight of carbon in one kg of fuel}}{\text{Weight of carbon in one kg of flue gas}}.$$

ANALYSIS OF EXHAUST AND FLUE GAS

The combustion products are mainly gaseous. When a sample is taken for analysis it is usually cooled down to a temperature which is below the saturation temperature of the steam present. The steam content is therefore not included in the analysis, which is then quoted as the analysis of the dry products. Since the products are gaseous, it is usual to quote the analysis by volume. An analysis which includes the steam in the exhaust is called a wet analysis.

Practical analysis of combustion products : The most common means of analysis of the combustion products is the Orsat apparatus which is described below :

Construction. An Orsat's apparatus consists of the following : (i) A burette (ii) A gas cleaner (iii) Four absorption pipettes 1, 2, 3, 4. The pipettes are interconnected by means of a manifold fitted with cocks S1, S2, S3 and S4 and contain different chemicals to absorb carbon dioxide (CO2), carbonmonoxide (CO) and oxygen (O2). Each pipette is also fitted with a number of small glass tubes which provide a greater amount of surface. These tubes are wetted by the absorbing agents and are exposed to the gas under analysis. The measuring burrette is surrounded by a water jacket to prevent, changes in temperature and density of the gas. The pipettes 1, 2, 3, 4 contain the following chemicals :

Pipette 1 : Contains 'KOH' (caustic soda) to absorb CO2 (carbon dioxide)

Pipette 2 : Contains an alkaline solution of 'pyrogallic acid' to absorb O2 (oxygen)

Pipette 3, 4 : Contain an acid solution of 'cuprous chloride' to absorb CO (carbonmonoxide) Furthermore the apparatus has a levelling bottle and a three way cock to connect the apparatus either to gases or to atmosphere.

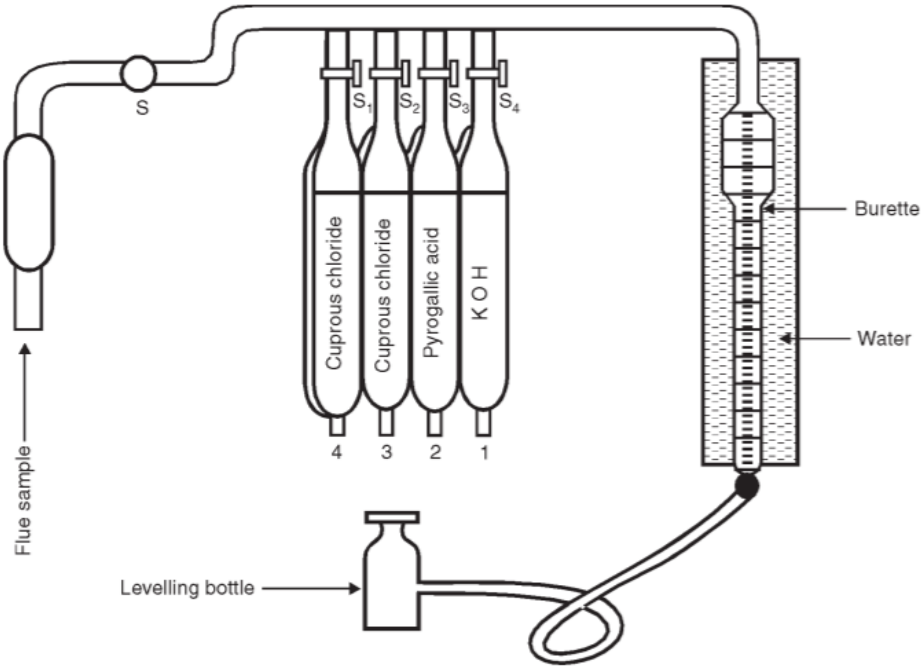


Fig.2. ANALYSIS OF EXHAUST AND FLUE GAS

Procedure. 100 cm^3 of gas whose analysis is to be made is drawn into the bottle by lowering the levelling bottle. The stop cock S4 is then opened and the whole flue gas is forced to pipette 1. The gas remains in this pipette for sometime and most of the carbondioxide is absorbed. The levelling bottle is then lowered to allow the chemical to come to its original level. The volume of gas thus absorbed is read on the scale of the measuring bottle. The flue gas is then forced through the pipette 1 for a number of times to ensure that the whole of the CO_2 is absorbed. Further, the remaining flue gas is then forced to the pipette 2 which contains pyrogallic acid to absorb whole of O_2. The reading on the measuring burette will be the sum of volume of CO_2 and O_2. The oxygen content can then be found out by subtraction. Finally, as before, the sample of gas is forced through the pipettes 3 and 4 to absorb carbonmonoxide completely. The amount of nitrogen in the sample can be determined by subtracting from total volume of gas the sum of CO_2, CO and O_2 contents. Orsat apparatus gives an analysis of the dry products of combustion. Steps may have been taken to remove the steam from the sample by condensing, but as the sample is collected over water it becomes saturated with water. The resulting analysis is nevertheless a true analysis of the dry products. This is because the volume readings are taken at a constant temperature and pressure, and the partial pressure of the vapour is constant. This means that the sum of the partial pressures of the remaining constituents is constant. The vapour then occupies the same proportion of the total volume at each measurement. Hence the vapour does not affect the result of the analysis. Note. Quantitatively the dry product analysis can be used to calculate A/F ratio. This method of obtaining the A/F ratio is not so reliable as direct measurement of air consumption and fuel consumption of the engine. More caution is required when analysing the products of consumption of a solid fuel since some of the products do not appear in the flue gases (e.g. ash and unburnt carbon). The residual solid must be analysed as well in order to determine the carbon content, if any. With an engine using petrol or diesel fuel the exhaust may include unburnt particles of carbon and this quantity will not appear in the analysis. The exhaust from internal combustion engines may contain also some CH_4 and H_2 due to incomplete combustion. Another piece of equipment called the Heldane apparatus measures the CH_4 content as well as CO_2, O_2 and CO.

CALORIFIC OR HEATING VALUES OF FUELS

The “calorific value or heating value” of the fuel is defined as the energy liberated by the complete oxidation of a unit mass or volume of a fuel. It is expressed in kJ/kg for solid and liquid fuels and kJ/m^3 for gases. If a fuel contains hydrogen water will be formed as one of the products of combustion. If this water is condensed, a large amount of heat will be released than if the water exists in the vapour phase. For this reason two heating values are defined ; the higher or gross heating value and the lower or net heating value. The higher heating value, HHV, is obtained when the water formed by combustion is completely condensed. The lower heating value, LHV, is obtained when the water formed by combustion exists completely in the vapour phase.

DETERMINATION OF CALORIFIC OR HEATING VALUES

Solid and Liquid Fuels

Dulong’s formula.

Dulong suggested a formula for the calculation of the calorific value of the solid or liquid fuels from their chemical composition which is as given below. Gross calorific value

$$\text{H.H.V.} = \frac{1}{100}\left[33800\,\text{C} + 144000\left(\text{H} - \frac{\text{O}}{8}\right) + 9270\,\text{S}\right] \text{ kJ/kg}$$

Laboratory method (Bomb calorimeter)

The calorific value of solid and liquid fuels is determined in the laboratory by ‘Bomb calorimeter’. It is so named because its shape resembles that of a bomb. Fig. 11.5 shows the schematic sketch of a bomb calorimeter

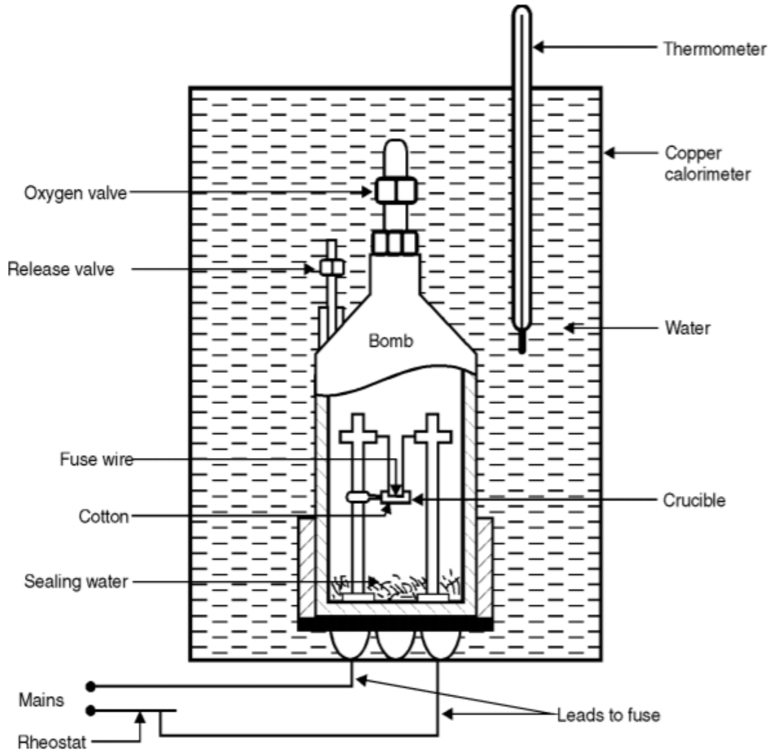


Fig. 3. Laboratory method (Bomb calorimeter)

The calorimeter is made of austenitic steel which provides considerable resistance to corrosion and enables it to withstand high pressure. In the calorimeter is a strong cylindrical bomb in which combustion occurs. The bomb has two valves at the top. One supplies oxygen to the bomb and other releases the exhaust gases. A crucible in which a weighted quantity of fuel sample is burnt is arranged between the two electrodes as shown in Fig. 11.5. The calorimeter is fitted with water jacket which surrounds the bomb. To reduce the losses due to radiation, calorimeter is further provided with a jacket of water and air. A stirrer for keeping the temperature of water uniform and a thermometer to measure the temperature up to an accuracy of 0.001°C are fitted through the lid of the calorimeter.

Procedure. To start with, about 1 gm of fuel sample is accurately weighed into the crucible and a fuse wire (whose weight is known) is stretched between the electrodes. It should be ensured that wire is in close contact with the fuel. To absorb the combustion products of sulphur and nitrogen 2 ml of water is poured in the bomb. Bomb is then supplied with pure oxygen through the valve to an amount of 25 atmosphere. The bomb is then placed in the weighed quantity of water, in the calorimeter. The stirring is started after making necessary electrical connections, and when the thermometer indicates a steady temperature fuel is fired and temperature readings are recorded after 1/2 minute intervals until maximum temperature is attained. The bomb is then removed ; the pressure slowly released through the exhaust valve and the contents of the bomb are carefully weighed for further analysis. The heat released by the fuel on combustion is absorbed by the surrounding water and the calorimeter.

From the above data the calorific value of the fuel can be found in the following way : Let wf = Weight of fuel sample (kg), w = Weight of water (kg), C = Calorific value (higher) of the fuel (kJ/kg), we = Water equivalent of calorimeter (kg), t1 = Initial temperature of water and calorimeter, t2 = Final temperature of water and calorimeter, t c = Radiation corrections, and c = Specific heat of water.

Heat released by the fuel sample = $w_f \times C$

Heat received by water and calorimeter

$$= (w_w + w_e) \times c \times [(t_2 - t_1) + t_c].$$

Heat lost = Heat gained

$$\therefore \qquad w_f \times C = (w + w_e) \times c \times [(t_2 - t_1) + t_c]$$

i.e.,

$$C = \frac{(w + w_e) \times c \times [(t_2 - t_1) + t_c]}{w_f}$$

Problem 1: The following particulars refer to an experimental determination of the calorific value of a sample of coal containing 88% C and 4.2% H2. Weight of coal = 0.848 gm, weight of fuse wire 0.027 gm, of calorific value 6700 J/gm, weight of water in the calorimeter = 1950 gm, water equivalent of calorimeter = 380 gm, observed temperature rise = 3.06°C, cooling correction = + 0.017°C. Find the higher and lower calorific values of the coal.

Solution. Percentage of *carbon* in coal = 88%

Percentage of *hydrogen* in coal = 4.2%

Weight of coal, $w_f = 0.848$ gm

Weight of fuse wire, $w_{fw} = 0.027$ gm

Weight of water in the calorimeter, $w = 1950$ gm

Water equivalent of calorimeter, $w_e = 380$ gm

Observed temperature rise $(t_2 - t_1) = 3.06$°C

Cooling correction, $t_c = + 0.017$°C

∴ Corrected temperature rise $= (t_2 - t_1) + t_c$

$= 3.06 + 0.017 = 3.077$°C

Calorific value of fuse wire = 6700 J/gm

Heat received by water $= (w + w_e) \times 4.18 \times [(t_2 - t_1) + t_c]$

$= (1950 + 380) \times 4.18 \times 3.077 = 29968$ J

Heat given out by fuse wire $= w_{fw} \times$ calorific value $= 0.027 \times 6700 = 180.9$ J

Heat produced due to combustion of fuel $= 29968 - 180.9 = 29787$ J

∴ *Higher calorific value of fuel, H.C.V.* $= \frac{29787}{0.848} = 35126$ J/gm = **35126 kJ/kg. (Ans.)**

Steam produced per kg of coal $= 9 \times 0.042 = 0.378$ kg

Lower calorific value of coal, L.C.V. = H.C.V. $- 2465 \times 0.378 = 35126 - 931.7$

= **34194.3 kJ/kg. (Ans.)**

Gaseous Fuels:

The calorific value of gaseous fuels can be determined by Junker's gas calorimeter. Fig. 11.6 illustrates Junker's gas calorimeter. Its principle is some what similar to Bomb calorimeter ; in respect that heat evolved by burning the gas is taken away by the water. In its simplest construction it consists of a combustion chamber in which the gas is burnt (in a gas burner). A water jacket through which a set of tubes called flues pass surrounds this chamber. Thermometers are incorporated at different places to measure the temperatures.

Procedure. A metered quantity of gas whose calorific value is to be determined is supplied to the gas burner via a gas meter which records its volume and a gas pressure regulator which measures the pressure of the gas by means of a manometer. When the gas burns the hot products of combustion travel upwards in the chamber and then downwards through the flues and finally escape to the atmosphere through the outlet. The temperature of the escaping gas is recorded by the thermometer fitted at the exit and this temperature should be as close to room temperature as possible so that entire heat of combustion is absorbed by water. The cold water enters the calorimeter near the bottom and leaves near the top. Water which is formed by condensation of steam is collected in a pot. The quantity of gas used during the experiment is accurately measured by the meter and temperature of ingoing and outgoing water are indicated by the thermometers. From the above data the calorific value of the gas can be calculated.

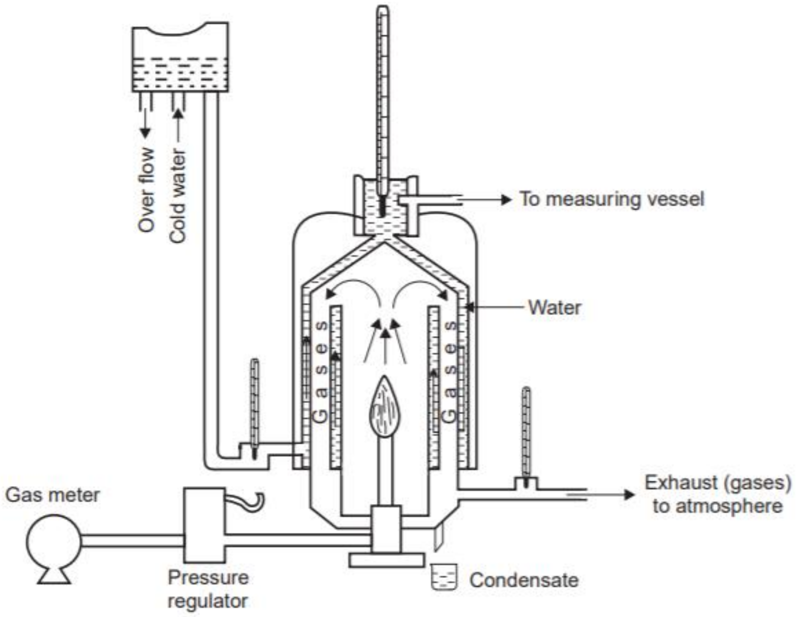


Fig.4. Junker's gas calorimeter

ADIABATIC FLAME TEMPERATURE

In a given combustion process, that takes place adiabatically and with no work or changes in kinetic or potential energy involved, the temperature of the products is referred to as the 'adiabatic flame temperature'. With the assumptions of no work and no changes in kinetic or potential energy, this is the maximum temperature that can be achieved for the given reactants because any heat transfer from the reacting substances and any incomplete combustion would tend to lower the temperature of the products. The following points are worthnoting : (i) The maximum temperature achieved through adiabatic complete combustion varies with the type of reaction and per cent of theoretical air supplied. An increase in the air-fuel ratio will effect a decrease in the maximum temperature. (ii) For a given fuel and given pressure and temperature of the reactants, the maximum adiabatic flame temperature that can be achieved is with a 'stoichiometric' mixture. (iii) The adiabatic flame temperature can be controlled by the amount of excess air that is used. This is important, for example, in gas turbines, where the maximum permissible temperature is determined by metallurgical considerations in the turbine, and close control of the temperature of the products is essential.

Problem 2: A coal sample gave the following analysis by weight, Carbon 85 per cent, Hydrogen 6 per cent, Oxygen 6 per cent, the remainder being incombustible. Determine minimum weight of air required per kg of coal for chemically correct composition.

Solution.

Element, wt. (kg)	*O_2 required (kg)*
$C = 0.85$	$0.85 \times \frac{8}{3} = 2.27$
$H_2 = 0.06$	$0.06 \times 8 = 0.48$
$O_2 = 0.06$	— — — — — —
	Total $O_2 = 2.75$

Weight of O_2 to be supplied = Wt. of O_2 needed – Wt. of O_2 already present in fuel

$$= 2.75 - 0.06 = 2.69 \text{ kg}$$

Weight of air needed $= 2.69 \times \frac{100}{23} = \mathbf{11.70\ kg.}$ **(Ans.)**

☞**Example 11.4.** *The percentage composition of sample of liquid fuel by weight is, $C = 84.8$ per cent, and $H_2 = 15.2$ per cent. Calculate (i) the weight of air needed for the combustion of 1 kg of fuel ; (ii) the volumetric composition of the products of combustion if 15 per cent excess air is supplied.*

Solution.

Element, wt. (kg)	*O_2 used (kg)*	*Dry products (kg)*
$C = 0.848$	$0.848 \times \frac{8}{3} = 2.261$	$\frac{0.848 \times 11}{3} = 3.109\ (CO_2)$
$H_2 = 0.152$	$0.152 \times 8 = 1.216$	
	Total $O_2 = 3.477$	

(*i*) **Minimum weight of air needed for combustion**

$$= \frac{3.477 \times 100}{23} = \mathbf{15.11\ kg.} \quad \textbf{(Ans.)}$$

Excess air supplied $= \frac{15.11 \times 15}{100} = 2.266$ kg

Wt. of oxygen in excess air $= \frac{2.266 \times 23}{100} = 0.521$ kg

Total air supplied for combustion = Minimum air + Excess air

$$= 15.11 + 2.266 = 17.376 \text{ kg}$$

∴ Wt. of nitrogen (N_2) in flue gases $= \frac{17.376 \times 77}{100} = 13.38$ kg.

(*ii*) To get **volumetric composition of the product of combustion** let us use tabular method.

Name of gas	*Weight (x)*	*Molecular weight (y)*	*Proportional volume (z)* = $\frac{(x)}{(y)}$	*Percentage volume* = $\frac{(z)}{\Sigma(z)} \times 100$
CO_2	3.109	44	0.0707	12.51 per cent. **(Ans.)**
O_2	0.521	32	0.0163	2.89 per cent. **(Ans.)**
N_2	13.38	28	0.4780	84.60 per cent. **(Ans.)**
			$\Sigma z = 0.5650$	

Problem 3: Percentage volumetric analysis of a sample of flue gases of a coal fired boiler gave CO2 = 10.4 ; CO = 0.2 ; O2 = 7.8 and N2 = 81.6 (by difference). Gravemetric percentage analysis of coal was C = 78, H2 = 6, O2 = 3 and incombustible = 13. Estimate : (i) Weight of dry flue gases per kg of fuel. (ii) Weight of excess air per kg of fuel.

Solution.

Element, wt. (kg)	*O_2 reqd. (kg)*	*Dry products (kg)*
C = 0.78	$0.78 \times \frac{8}{3} = 2.08$	$0.78 \times \frac{11}{3} = 2.86$ (CO_2)
H_2 = 0.06	$0.06 \times 8 = 0.48$	
O_2 = 0.03		
	Total O_2 = 2.56	

Minimum wt. of air needed for combustion = $(2.56 - .03) \times \frac{100}{23} = 11$ kg.

(*i*) **Weight of dry flue gases per kg of fuel :**

To determine the wt. of flue gases per kg of fuel let us use tabular method to convert volumetric analysis to analysis by weight.

Name of gas	*Volume per m^3 of flue gas* (*x*)	*Molecular weight* (*y*)	*Relative volume* $z = x \times y$	*Weight per kg of flue gas* $= \frac{z}{\Sigma z}$
CO_2	0.104	44	4.576	0.1525
CO	0.002	28	0.056	0.0019
N_2	0.816	28	22.848	0.7616
O_2	0.078	32	2.496	0.0832
			$\Sigma z = 29.976$ (say 30)	

Amount of carbon present per kg of gases

= Amount of carbon in 0.1525 kg of CO_2 + Amount of carbon present in 0.0019 kg of CO

$$= \frac{3}{11} \times 0.1525 + \frac{3}{7} \times 0.0019 = 0.0416 + 0.0008 = 0.0424 \text{ kg.}$$

Also carbon in the fuel = 0.78 kg.

∴ *Weight of dry flue gas per kg of fuel*

$$= \frac{\text{Weight of carbon in 1 kg of fuel}}{\text{Weight of carbon in 1 kg of flue gas}} = \frac{0.78}{0.0424} = \mathbf{18.4\ kg.} \quad \textbf{(Ans.)}$$

(*ii*) **Weight of excess air per kg of fuel :**

Weight of excess oxygen per kg of flue gas $= 0.0832 - \frac{4}{7} \times .0019$

$$\left[\begin{array}{l} 2CO + O_2 = 2CO_2 \\ \quad 56 \quad 32 \quad 88 \\ 1\ \text{kg} + \frac{4}{7}\ \text{kg} = \frac{11}{7}\ \text{kg} \end{array}\right]$$

$= 0.0832 - 0.0011$ (allowing for unburnt carbon monoxide)

$= 0.0821$ kg.

Weight of excess O_2 per kg of fuel $= 18.4 \times 0.0821 = 1.51$ kg

∴ *Weight of excess air per kg of fuel* $= \frac{1.51 \times 100}{23} = \mathbf{6.56\ kg.}$ **(Ans.)**

Problem 4: The following is the ultimate analysis of a sample of petrol by weight : Carbon = 85 per cent ; Hydrogen = 15 per cent. Calculate the ratio of air to petrol consumption by weight if the volumetric analysis of the dry exhaust gas is : CO2 = 11.5 per cent ; CO = 1.2 per cent ; O2 = 0.9 per cent ; N2 = 86 per cent. Also find percentage excess air.

Solution.

Name of gas	*Volume per m³ of flue gas* (*x*)	*Molecular weight* (*y*)	*Relative weight* $z = x \times y$	*Weight per kg of flue gas* $\frac{z}{\Sigma z}$
CO_2	0.115	44	5.06	0.1700
CO	0.012	28	0.336	0.0113
O_2	0.009	32	0.288	0.0096
N_2	0.86	28	24.08	0.8091
			$\Sigma z = 29.76$	

∴ Weight of carbon per kg of flue gas

= Weight of carbon in 0.17 kg of CO_2 + Weight of carbon in 0.0113 kg of CO

$$= \frac{3}{11} \times 0.17 + \frac{3}{7} \times 0.0113 = 0.0512 \text{ kg}$$

∴ Weight of dry flue gas per kg of fuel $= \frac{0.85}{0.0512} = 16.6$ kg

Vapour of combustion $= 9 \times 0.15 = 1.35$ kg

Total weight of gas $= 16.6 + 1.35 = 17.95$ kg per kg of fuel

∴ Air supplied $= (17.95 - 1) = 16.95$ kg/kg of fuel

∴ *Ratio of air to petrol* = **16.95 : 1. (Ans.)**

Stoichiometric air $= \left[\left(0.85 \times \frac{8}{3}\right) + (0.15 \times 8)\right] \times \frac{100}{23}$

$= 15.07$ kg per kg of fuel

∴ Excess air $= 16.95 - 15.07 = 1.88$ kg

∴ *Percentage excess air* $= \frac{1.88}{15.07} \times 100 = \mathbf{12.47\%.}$ **(Ans.)**

Heat transfer

MODES OF HEAT TRANSFER

‘‘Heat transfer’’ which is defined as the transmission of energy from one region to another as a result of temperature gradient takes place by the following three modes : (i) Conduction ; (ii) Convection ; (iii) Radiation. Heat transmission, in majority of real situations, occurs as a result of combinations of these modes of heat transfer. Example : The water in a boiler shell receives its heat from the firebed by conducted, convected and radiated heat from the fire to the shell, conducted heat through the shell and conducted and convected heat from the inner shell wall, to the water. Heat always flows in the direction of lower temperature. The above three modes are similar in that a temperature differential must exist and the heat exchange is in the direction of decreasing temperature ; each method, however, has different controlling laws.

(i) **Conduction**. ‘Conduction’ is the transfer of heat from one part of a substance to another part of the same substance, or from one substance to another in physical contact with it, without appreciable displacement of molecules forming the substance. In solids, the heat is conducted by the following two mechanisms : (i) By lattice vibration (The faster moving molecules or atoms in the hottest part of a body transfer heat by impacts some of their energy to adjacent molecules). (ii) By transport of free electrons (Free electrons provide an energy flux in the direction of decreasing temperature—For metals, especially good electrical conductors, the electronic mechanism is responsible for the major portion of the heat flux except at low temperature). In case of gases, the mechanisam of heat conduction is simple. The kinetic energy of a molecule is a function of temperature. These molecules are in a continuous random motion exchanging energy and momentum. When a molecule from the high temperature region collides with a molecule from the low temperature region, it loses energy by collisions.

In liquids, the mechanism of heat is nearer to that of gases. However, the molecules are more closely spaced and intermolecular forces come into play.

(ii) **Convection**. ‘Convection’ is the transfer of heat within a fluid by mixing of one portion of the fluid with another. Convection is possible only in a fluid medium and is directly linked with the transport of medium itself. Convection constitutes the macroform of the heat transfer since macroscopic particles of a fluid moving in space cause the heat exchange. The effectiveness of heat transfer by convection depends largely upon the mixing motion of the fluid. This mode of heat transfer is met with in situations where energy is transferred as heat to a flowing fluid at any surface over which flow occurs. This mode is basically conduction in a very thin fluid layer at the surface and then mixing caused by the flow. The heat flow depends on the properties of fluid and is independent of the properties of the material of the surface. However, the shape of the surface will influence the flow and hence the heat transfer.

Free or natural convection. Free or natural convection occurs where the fluid circulates by virtue of the natural differences in densities of hot and cold fluids ; the denser portions of the fluid move downward because of the greater force of gravity, as compared with the force on the less dense.

Forced convection. When the work is done to blow or pump the fluid, it is said to be forced convection.

(iii) **Radiation.** ‘Radiation’ is the transfer of heat through space or matter by means other than conduction or convection. Radiation heat is thought of as electromagnetic waves or quanta (as convenient) an emanation of the same nature as light and radio waves. All bodies radiate heat ; so a transfer of heat by radiation occurs because hot body emits more heat than it receives and a cold body receives more heat than it emits. Radiant energy (being electromagnetic radiation) requires no medium for propagation and will pass through a vacuum.

TEXT / REFERENCE BOOKS

1. Nag P.K., “Engineering Thermodynamics”, Tata McGraw Hill Education, 2009.
2. Yunus A. Cengel, Michael A. Boles, “Thermodynamics: An Engineering Approach”, McGraw Hill Education, 2014.
3. Rajput R.K., “Engineering Thermodynamics”, Laxmi Publications, 2010.
4. Khurmi R.S., Gupta J.K, “Thermal Engineering”, S Chand, 2006.
5. P.L. Ballaney,”Thermal Engineering”, Khanna Publisher, 2005.