MALLA REDDY COLLEGE OF ENGINEERING & TECHNOLOGY

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DEPARTMENT OF MECHANICAL ENGINEERING

DIGITAL NOTES of ADVANCED THERMAL ENGINEERING

For

B.Tech – III YEAR – I

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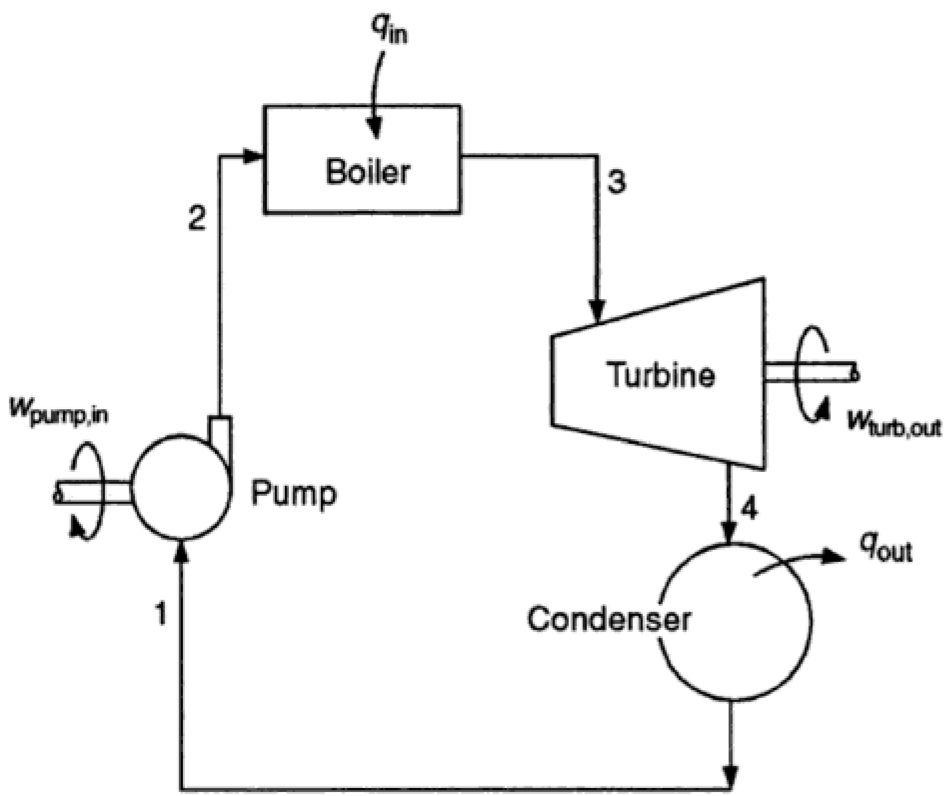
I UNIT

Rankine Cycle

The Rankine cycle is an ideal cycle for vapour power cycles.



Principle components of Rankine cycle:

The four basic components of Rankine cycle are shown in figure 1 each component in the cycle is regarded as control volume, operating at steady state.

Pump: The liquid condensate leaving the condenser at the state 1 is pumped to the operating pressure of the boiler. The pump operation is considered isentropic.

Boiler: The heat is supplied in the working fluid (feed water) in the boiler and thus vapor is generated. The vapor leaving the boiler is either at saturated at the state 3 or superheated at the state $3'$, depending upon the amount of heat supplied by the boiler.

Turbine: The vapor leaving the boiler enters the turbine, where it expands isentropically to the condenser pressure at the state 4. The work produced by the turbine is rotary (shaft) work and is used to drive an electric generator or machine.

Condenser: The condenser is attached at the exit of the turbine. The vapor leaving the turbine is wet vapor and it is condensed completely in the condenser to the state 1, by giving its latest heat to some other cooling fluid like water.

The **various processes in simple Rankine cycle** are:

1–2: Reversible adiabatic pumping process in the pump,

2–3: Constant-pressure transfer of heat in the boiler,

3–4: Reversible adiabatic expansion in the turbine (or other prime movers such as a steam engine),

4–1: Constant-pressure transfer of heat in the condenser.

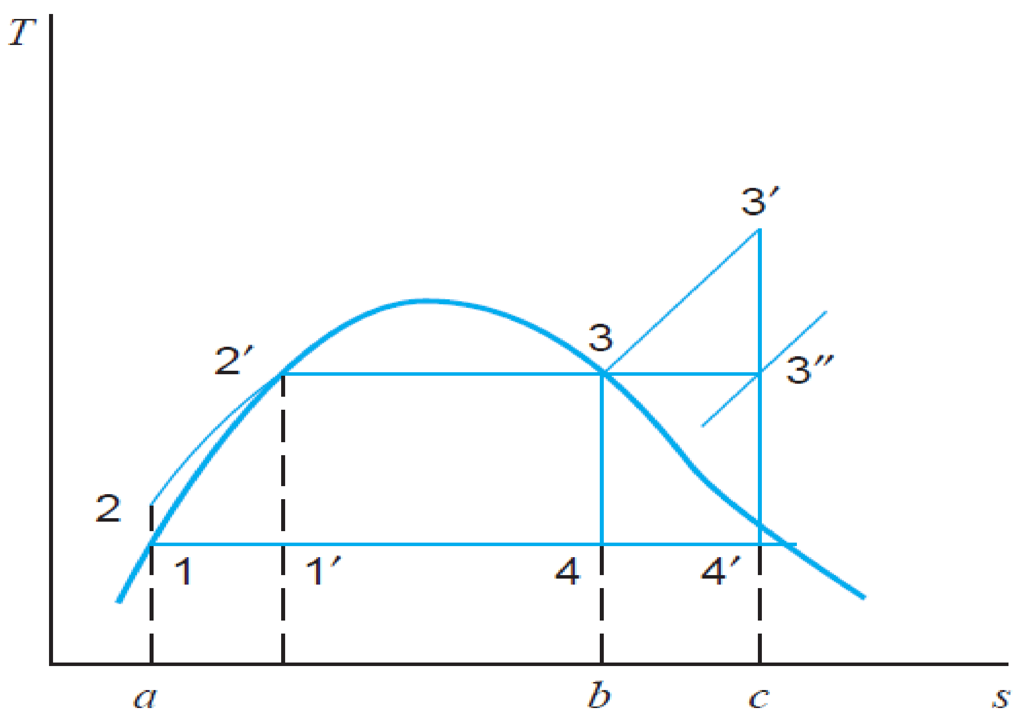


Figure 2: Temperature vs entropy diagram of Rankine cycle

The Rankine cycle also includes the possibility of superheating the vapor, as cycle 1–2–3′–4′–1.

If kinetic and potential energy changes are neglected, heat transfer and work may be represented by various areas on the T–s diagram. The heat transferred to the working fluid is represented by area a–2–2′–3–b–a and the heat transferred from the working fluid by area a–1–4–b–a. From the first law of thermodynamics we can conclude that the area representing the work is the difference between these two areas—area 1–2–2′–3–4–1. The thermal efficiency is defined by the relation

$$\eta_{th} = \frac{w_{net}}{q_H} = \frac{\text{area } 1-2-2'-3-4-1}{\text{area } a-2-2'-3-b-a}$$

Net heat transfer in the cycle

$$q_{net} = q_{boiler} - q_{condenser} = q_{in} - q_{out} = (h_1 - h_4) - (h_2 - h_3)$$

Again, for a cycle Net heat transfer = Net work transfer

$$\therefore \quad w_{net} = (h_1 - h_4) - (h_2 - h_3)$$

Also, it can be shown that

$$\begin{aligned} w_{net} &= w_{turbine} - w_{pump} \\ &= (h_1 - h_2) - (h_4 - h_3) \\ &= (h_1 - h_4) - (h_2 - h_3) \end{aligned}$$

The thermal efficiency of the Rankine cycle is

$$\eta_{th} = \frac{\text{Net work output}}{\text{Heat supplied in the boiler}}$$

$$= \frac{(h_1 - h_2) - (h_4 - h_3)}{h_1 - h_4}$$

$$\Rightarrow \quad \eta_{th} = \frac{(h_1 - h_2) - (h_4 - h_3)}{(h_1 - h_3) - (h_4 - h_3)}$$

If the feed-pump term, $h_4 - h_3$, is neglected, Eq. (6.8) becomes

$$\eta_{th} = \frac{h_1 - h_2}{h_1 - h_3}$$

Also, as the pump work is associated with a liquid which can be assumed to be incompressible, we can write

$$h_4 - h_3 = v\,(p_4 - p_3)$$

For an isentropic process

$$dq = dh - v\,dp = 0$$

$$\therefore \qquad dh = v\,dp$$

$$\text{or} \qquad \int_3^4 dh = \int_3^4 v\,dp$$

$$\therefore \qquad h_4 - h_3 = v\,(p_4 - p_3)$$

$$\therefore \qquad \text{Pump work input} = v\,(p_4 - p_3)$$

v is found from tables for water corresponding to p_3.

For analyzing the Rankine cycle, it is helpful to think of efficiency as depending on the average temperature at which heat is supplied and the average temperature at which heat is rejected. Any changes that increase the average temperature at which heat is supplied or decrease the average temperature heat is rejected will increase the Rankine-cycle efficiency.

The *efficiency ratio* of a cycle is the ratio of the actual efficiency to the ideal efficiency.

Here,
$$\text{Efficiency ratio} = \frac{\text{Actual cycle efficiency}}{\text{Rankine cycle efficiency}}$$

The actual expansion and compression processes are irreversible, as shown by broken lines in Fig. 6.4.

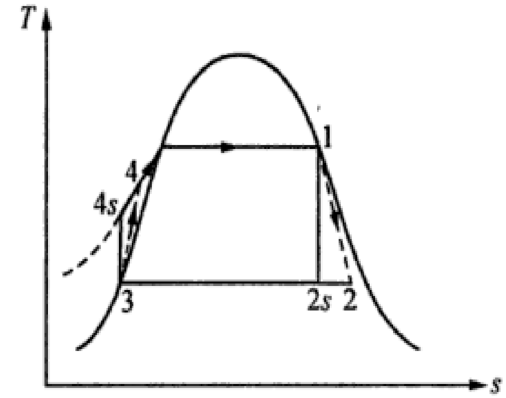


Fig. 6.4 Rankine cycle with real expansion and compression processes.

The *isentropic efficiency* of a process is defined as

$$\text{Isentopic efficiency} = \frac{\text{Actual work}}{\text{Isentropic work}}, \quad \text{for expansion.}$$

$$= \frac{\text{Isentropic work}}{\text{Actual work}}, \quad \text{for compression.}$$

For example,

$$\text{Turbine isentropic efficiency} = \frac{h_1 - h_2}{h_1 - h_{2s}}$$

There is another term called the ***specific steam consumption*** **(ssc).** It is the steam flow required to develop unit work output.

$$\Rightarrow \quad \text{ssc} = \frac{1}{h_1 - h_2}$$

Rankine Cycle with Superheat

The average temperature at which heat is supplied in the boiler can be increased by providing superheat to the steam. Generally, the dry saturated steam from the boiler drum is passed through a second bank of smaller tubes within the boiler. This bank is located so that it gets heat from the hot gases from the surface to enable the steam to reach the requisite temperature.

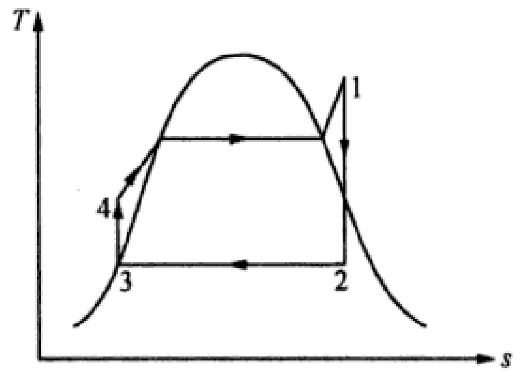


Fig. Rankine cycle with superheat.

Figure 6.5 shows the *T–s* diagram of a Rankine cycle with superheat. The effect of providing superheat is actually a two-fold one:

1. It increases the *average temperature of heat addition* and hence the cycle efficiency. This is clear from

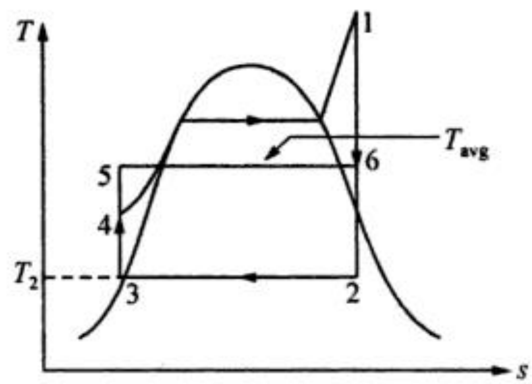


Fig. 6.6 Average temperature of heat addition.

T_{avg} is the average temperature of heat addition such that the area under 4–1 is equal to that under 5–6.

Heat added, $$q_{in} = h_1 - h_4 = T_{avg}(s_1 - s_4)$$

$\therefore$ $$T_{avg} = \frac{h_1 - h_4}{s_1 - s_4}$$

Heat rejected, $$q_{out} = h_2 - h_3 = T_2(s_1 - s_4)$$

$\therefore$ $$\eta_{th} = \frac{W_{net}}{q_{in}}$$

$$= 1 - \frac{q_{out}}{q_{in}}$$

$$= 1 - \frac{T_2(s_1 - s_4)}{T_{avg}(s_1 - s_4)}$$

$$= 1 - \frac{T_2}{T_{avg}}$$

So *increase in T_{avg} leads to increase in η_{th}* because T_2 may be fixed by the ambient condition of the plant.

2. Increase in superheat ensures a shift of point 2 to the right which results in less moisture in the turbine. The performance of the turbine improves.

In actual case, the fluid friction leads to a pressure drop in the boiler and the condenser and the actual cycle looks like Fig. 6.7.

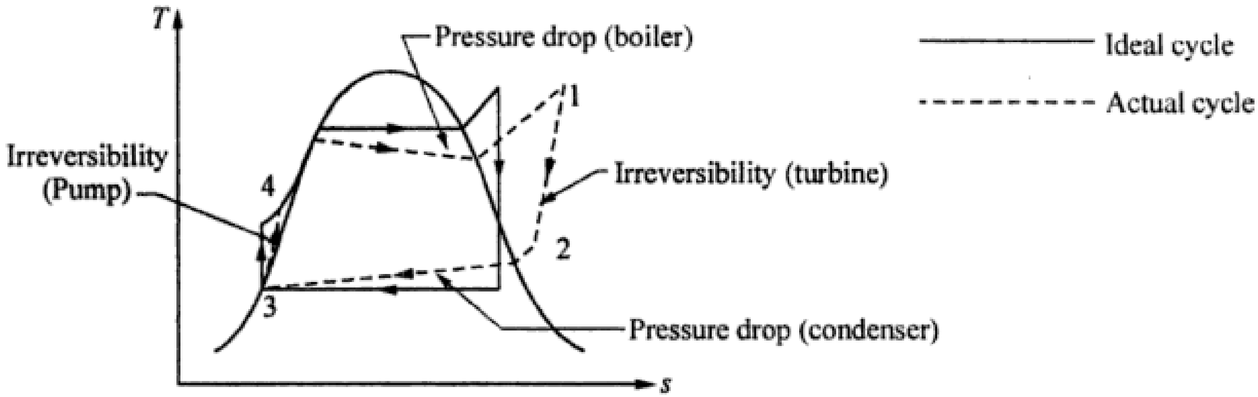
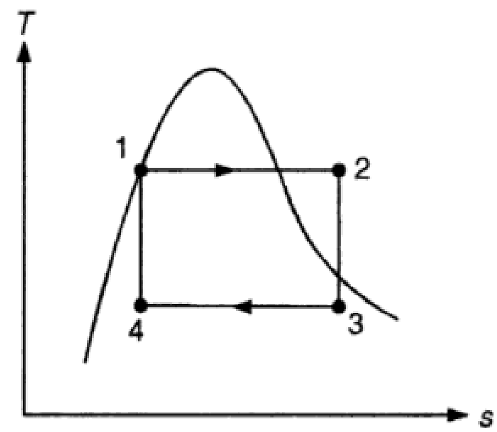


Fig. 6.7 Deviation of actual vapour power cycle from the ideal Rankine cycle.

EXAMPLE 9.1

A steam power plant operates between a boiler pressure of 4 MPa and 300°C and a condenser pressure of 50 kPa. Determine the thermal efficiency of the cycle, the work ratio, and the specific steam flow rate. assuming the cycle to be a Carnot cycle



Solution

(a) The T–s diagram of a Carnot cycle is shown in the adjacent figure.

Process 1–2 is reversible and isothermal heating of water in the boiler.

Process 2–3 is isentropic expansion of steam at state 2 in the turbine.

Process 3–4 is reversible and isothermal condensation of steam in the condenser.

Process 4–1 is isentropic compression of steam to initial state.

At state 1: P_1 = 4 MPa, T_1 = 300°C

At state 2: P_2 = 50 kPa, the steam is in a saturated state.

From the saturated water-pressure table (Table 4 of the Appendix), at 50 kPa, we get $T_2 = T_{min} = T_{sat} = 81.33°C$

Therefore, the thermal efficiency for the given Carnot cycle is

$$\eta_{th,carnot} = 1 - \frac{T_{min}}{T_{max}} = 1 - \frac{81.33 + 273.15}{300 + 273.15} = 0.3815$$

$$= \boxed{38.15 \text{ per cent}}$$

$$\text{The work ratio} = \frac{\text{net work output}}{\text{gross work output}} = \frac{w_{net,out}}{w_{gross,out}}$$

Heat supplied = $h_2 - h_1 = h_{fg\,@\,4MPa}$ = 1714.1 kJ/kg (From Table 4 of the Appendix)

$$\eta_{th,carnot} = \frac{w_{net,out} - w_{net,in}}{\text{gross heat supplied}} = 0.3815$$

Therefore,

$$w_{net,\,out} - w_{net,\,in} = 0.3815 \times 1714.1 = 653.9 \text{ kJ/kg}$$

That is, the net work output = 653.9 kJ/kg.

To find the expansion work for the process 2–3, h_3 is required.

From Table 4, h_2 = 2801.4 kJ/kg and $s_2 = s_3$ = 6.0701 kJ/(kg K)

But
$$s_3 = 6.0701 = s_{f3} + x_3 s_{fg3} = 1.0910 + x_3(7.5939 - 1.0910)$$
or
$$x_3 = 0.766$$

Now,
$$h_3 = h_{f3} + x_3 h_{fg3} = 340.49 + 0.766(2645.9 - 340.49) = 2106.4 \text{ kJ/kg}$$

Therefore,
$$w_{32} = h_2 - h_3 = 2801.4 - 2106.4 = 695 \text{ kJ/kg}$$

That is, the gross work output, $w_{gross,out}$ = 695 kJ/kg
Therefore,

$$\text{Work ratio} = \frac{w_{net,out}}{w_{gross,out}} = \frac{653.9}{695} = \boxed{0.94}$$

The specific steam flow rate (ssfr) is the steam flow required to develop unit power output. That is,

$$\text{ssfr} = \frac{\dot{m}_{\text{steam}}}{\dot{m}_s w_{\text{out}}} = \frac{1}{w_{\text{net, out}}}$$

$$= \frac{1}{653.9} = \boxed{0.00153 \text{ kg/kW}}$$

2. A steam power plant uses steam as working fluid and operates at a boiler pressure of 5 MPa, dry saturated and a condenser pressure of 5 kPa. Determine the cycle efficiency for (a) Carnot cycle (b) Rankine cycle. Also show the T-s representation for both the cycles.

Solution:

From steam tables:

At 5 MPa $\quad h_{f,\,5\text{MPa}} = 1154.23$ kJ/kg, $s_{f,\,5\text{ MPa}} = 2.92$ kJ/kg · K

$h_{g,\,5\text{MPa}} = 2794.3$ kJ/kg, $s_{g,\,5\text{ MPa}} = 5.97$ kJ/kg · K

At 5 kPa

$h_{f,\,5\text{kPa}} = 137.82$ kJ/kg, $s_{f,\,5\text{kPa}} = 0.4764$ kJ/kg · K

$h_{g,\,5\text{kPa}} = 2561.5$ kJ/kg, $s_{g,\,5\text{kPa}} = 8.3951$ kJ/kg · K

$v_{f,\,5\text{kPa}} = 0.001005$ m^3/kg

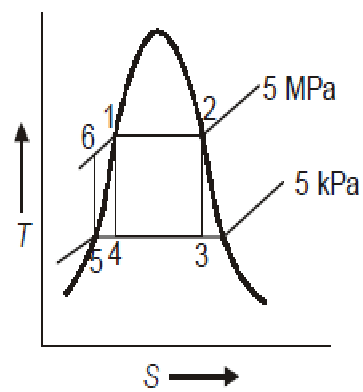


Fig. 8.30

Carnot cycle : 1–2–3–4–1

Rankine cycle : 1–2–3–5–6–1

As process 2-3 is isentropic, so $s_2 = s_3$

and

$$s_3 = s_{f,\,5\text{kPa}} + x_3 \cdot s_{fg,\,5\text{kPa}} = s_2 = s_{g,\,5\text{MPa}}$$

$$x_3 = 0.694$$

Hence enthalpy at 3,

$$h_3 = h_{f,\,5\text{kPa}} + x_3 \cdot h_{fg,\,5\text{kPa}}$$

$$h_3 = 1819.85 \text{ kJ/kg}$$

Enthalpy at 2, $\quad h_2 = h_{g,\,5\text{MPa}} = 2794.3$ kJ/kg

Process 1-4 is isentropic, so $s_1 = s_4$

$$s_1 = 2.92 = 0.4764 + x_4 \cdot (8.3951 - 0.4764)$$

$$x_4 = 0.308$$

Enthalpy at 4,

$$h_4 = 137.82 + (0.308 \times (2561.5 - 137.82))$$

$$h_4 = 884.3 \text{ kJ/kg}$$

Enthapy at 1,

$$h_1 = h_{f \text{ at 5 MPa}}$$

$$h_1 = 1154.23 \text{ kJ/kg}$$

Carnot cycle (1-2-3-4-1) efficiency:

$$\eta_{\text{carnot}} = \frac{\text{Net work}}{\text{Heat added}}$$

$$= \frac{(h_2 - h_3) - (h_1 - h_4)}{(h_2 - h_1)}$$

$$= \frac{\{(2794.3 - 1819.85) - (1154.23 - 884.3)\}}{(2794.3 - 1154.23)}$$

$$\eta_{\text{carnot}} = 0.4295$$

In Rankine cycle, 1-2-3-5-6-1

$$\text{Pump work, } h_6 - h_5 = v_{f,5}(p_6 - p_5)$$

$$= 0.001005\ (5000 - 5)$$

$$h_6 - h_5 = 5.02$$

$$h_5 = h_{f \text{ at 5kPa}} = 137.82 \text{ kJ/kg}$$

$$\text{Hence } h_6 = 137.82 + 5.02 = 142.84 \text{ kJ/kg}$$

$$h_6 = 142.84 \text{ kJ/kg}$$

$$\text{Net work in Rankine cycle} = (h_2 - h_3) - (h_6 - h_5)$$

$$= 974.45 - 5.02$$

$$= 969.43 \text{ kJ/kg}$$

$$\text{Heat added} = h_2 - h_6$$

$$= 2794.3 - 142.84$$

$$= 2651.46 \text{ kJ/kg}$$

$$\text{Rankine cycle efficiency} = \frac{969.43}{2651.46}$$

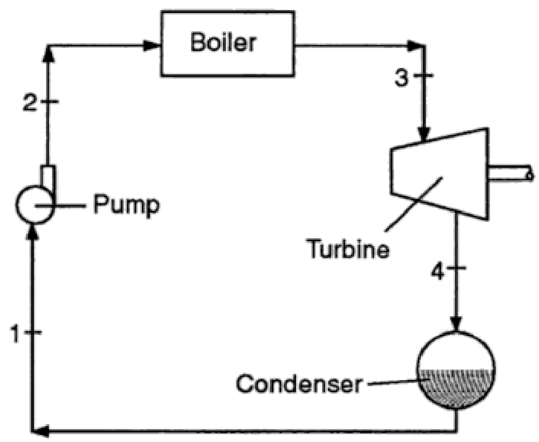
$$\eta_{\text{Rankine}} = 0.3656$$

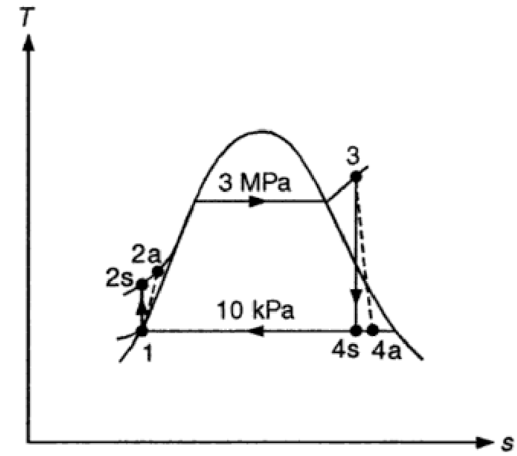
or

$$\eta_{\text{Rankine}} = \mathbf{36.56\%} \quad \textbf{Ans.}$$

EXAMPLE 2

A steam power plant operates on the cycle shown below with 3 MPa and 400°C at the turbine inlet and 10 kPa at the turbine exhaust. The adiabatic efficiency of the turbine is 85 per cent and that of the pump is 80 per cent. Determine (a) the thermal efficiency of the cycle, and (b) the mass flow rate of the steam if the power output is 20 MW.





Solution

All the components are treated as steady-flow devices. The changes, if any, in the kinetic and potential energies are assumed to be negligible. Losses other than those in the turbine and pump are neglected.

(a)
$$w_{\text{pump,in}} = \frac{v_1(P_2 - P_1)}{\eta_P} = \frac{0.001010\,(3000 - 10)}{0.80} = 3.77 \text{ kJ/kg}$$

Turbine work output is

$$w_{\text{turb,out}} = \eta_T w_{\text{turb,in}} = \eta_T (h_3 - h_{4s})$$
$$= 0.85(3230.90 - 2192.21) = 882.89 \text{ kJ/kg}$$

Boiler heat input is

$$q_{in} = h_3 - h_2 = 3230.9 - 195.59 = 3035.31 \text{ kJ/kg}$$

Thus,

$$w_{net,out} = w_{turb,out} - w_{pump,in} = 882.89 - 3.77 = 879.12 \text{ kJ/kg}$$

$$\eta_{th} = \frac{w_{net,out}}{q_{in}} = \frac{879.12}{3035.31} = 0.2896 = \boxed{28.96 \text{ per cent}}$$

If there are no losses in the turbine and the pump, the thermal efficiency would be 28.99 per cent.

(b) The power generated by the power plant is

$$\dot{W}_{net,out} = \dot{m} w_{net,out} = 20{,}000 \text{ kW}$$

Therefore, the mass flow rate, $\dot{m} = \dfrac{20{,}000}{879.12} = \boxed{22.75 \text{ kg/s}}$

3. A steam turbine plant operates on Rankine cycle with steam entering turbine at 40 bar, 350°C and leaving at 0.05 bar. Steam leaving turbine condenses to saturated liquid inside condenser. Feed pump pumps saturated liquid into boiler. Determine the net work per kg of steam and the cycle efficiency assuming all processes to be ideal. Also show cycle on T-s diagram. Also determine pump work per kg of steam considering linear variation of specific volume.

Solution:

From steam table

$$h_2 = h_{\text{at 40 bar, 350°C}} = 3092.5 \text{ kJ/kg}$$

$$s_2 = s_{\text{at 40 bar, 350°C}} = 6.5821 \text{ kJ/kg·K}$$

$$h_4 = h_{f \text{ at 0.05 bar}} = 137.82 \text{ kJ/kg}$$

$$s_4 = s_{f \text{ at 0.05 bar}} = 0.4764 \text{ kJ/kg}$$

$$v_4 = v_{f \text{ at 0.05 bar}} = 0.001005 \text{ m}^3\text{/kg}$$

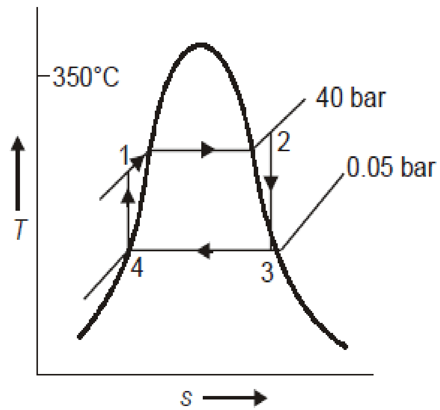


Fig. 8.31

Let dryness fraction at state 3 be x_3,

For ideal process, 2-3, $s_2 = s_3$

$$s_2 = s_3 = 6.5821 = s_{f \text{ at } 0.05 \text{ bar}} + x_3 \cdot s_{fg \text{ at } 0.05 \text{ bar}}$$

$$6.5821 = 0.4764 + x_3 \cdot 7.9187$$

$$x_3 = 0.7711$$

$$h_3 = h_{f \text{ at } 0.05 \text{ bar}} + x_3 \cdot h_{fg \text{ at } 0.05 \text{ bar}}$$

$$= 137.82 + (0.7711 \times 2423.7)$$

$$h_3 = 2006.74 \text{ kJ/kg}$$

For pumping process

$$h_1 - h_4 = v_4 \cdot \Delta p = v_4 \times (p_1 - p_4)$$

$$h_1 = h_4 + v_4 \times (p_1 - p_4)$$

$$= 137.82 + (0.001005 \times (40 - 0.05) \times 10^2)$$

$$h_1 = 141.84 \text{ kJ/kg}$$

Pump work per kg of steam $= (h_1 - h_4) = 4.02$ kJ/kg

Net work per kg of steam = (Expansion work – Pump work) per kg of steam

$$= (h_2 - h_3) - (h_1 - h_4)$$

$$= 1081.74 \text{ kJ/kg}$$

$$\text{Cycle efficiency} = \frac{\text{Net work}}{\text{Heat added}}$$

$$= \frac{1081.74}{(h_2 - h_1)}$$

$$= \frac{1081.74}{(3092.5 - 141.84)}$$

$$= 0.3667 \text{ or } 36.67\%$$

$$\boxed{\begin{aligned} \text{Net work per kg of steam} &= \mathbf{1081.74 \ kJ/kg} \\ \text{Cycle efficiency} &= \mathbf{36.67\%} \\ \text{Pump work per kg of steam} &= \mathbf{4.02 \ kJ/kg} \end{aligned}} \quad \textbf{Ans.}$$

Methods to Increase the Rankine Cycle Efficiency

The efficiency of the Rankine cycle can be increased, as we have already discussed in Eq. (6.14), by increasing the average temperature of the heat addition (T_{avg}), or by reducing the average temperature at which heat is rejected in the condenser (T_2). Let us discuss the several ways to increase this efficiency.

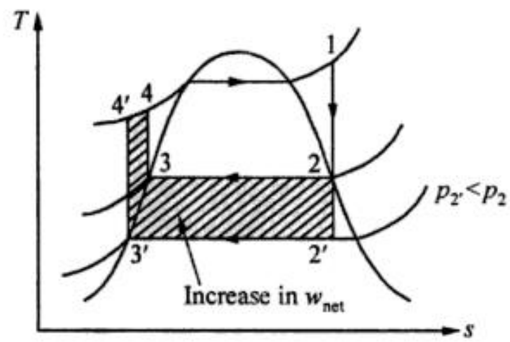


Fig. 6.8 Effect of reducing condenser pressure.

1. Lowering the condenser pressure: By lowering the temperature of heat rejection the efficiency increases (Eq. 6.14). Reduction in the condenser pressure also leads to decrease in temperature of heat rejection. In Fig. 6.8, the shaded area shows the increase in net work output as we move to pressure $p_{2'}$ from p_2 ($p_{2'} < p_2$). The input heat requirement increases a bit as evident from the area under the curve $4' - 4$. But this area is quite small.

Usually condensers in steam power plants are operated below atmospheric pressure just to increase the efficiency by lowering the temperature of heat rejection. This incurs the problem of air leakage. Also, the moisture content in the turbine increases. The second problem can be taken care of by suitable methods to be discussed later on.

2. Superheating the steam to high temperatures: This method increases the average temperature of heat addition and reduces moisture content in the turbine (Fig. 6.9). But the maximum temperature of superheating is restricted by metallurgical constraints.

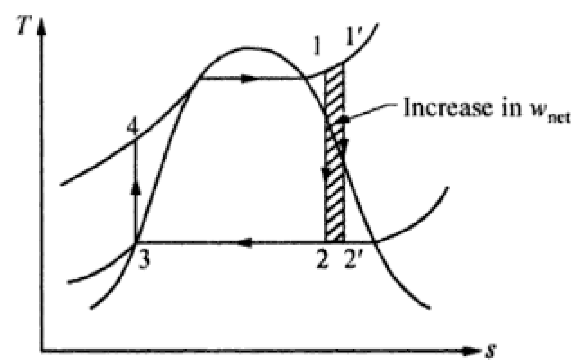


Fig. 6.9 Effect of superheating the steam.

3. Increasing the boiler pressure: This is another way to increase T_{avg}. Increase in boiler pressure automatically increase the temperature of boiling. But the shift of the process 1–2 to 1′–2′ towards left increase the moisture contents of the turbine (Fig. 6.10). This problem has a remedy in the form of superheating, to be discussed in the next article.

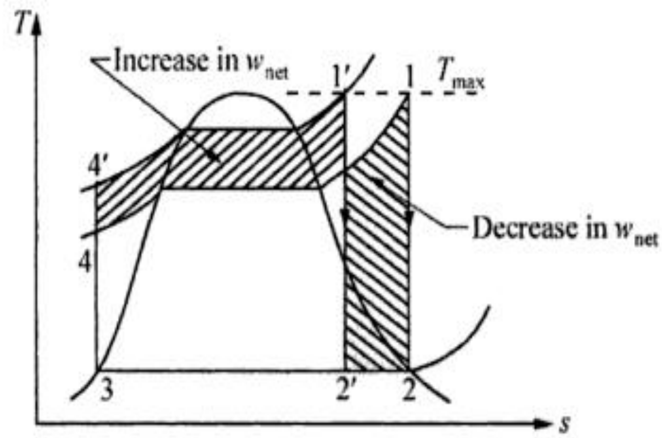


Fig. 6.10 Effect of increase in boiler pressure.

REHEAT CYCLE

Schematic of reheat cycle is as shown in Fig. 8.10. Reheat cycle is based on the simple fact of realizing high efficiency with higher boiler pressure and yet avoid low quality of steam at turbine exhaust.

Here steam generated in boiler is supplied to high pressure steam turbine at state 2 and is expanded upto state 3. This steam is sent to boiler for being reheated so that its temperature gets increased, normally this temperature after reheating may be equal to temperature at inlet of high pressure steam turbine. Steam after reheating is supplied to subsequent turbine at state 4, say to low pressure steam turbine. Steam is now expanded upto the exhaust pressure say state '5'. Expanded steam is subsequently sent to condenser and condensate at state '6' is pumped back to the boiler employing feed pump at state '1'. Thus, it is possible to take advantage of high steam pressure at inlet to steam turbine as the problem of steam becoming excessively wet with increasing steam pressure could be regulated by reheating during the expansion. Expansion occurs in two stages one begining at high pressure and other occurring at low pressure with reheating in between. The principal advantage of reheat is to increase the quality of steam at turbine exhaust.

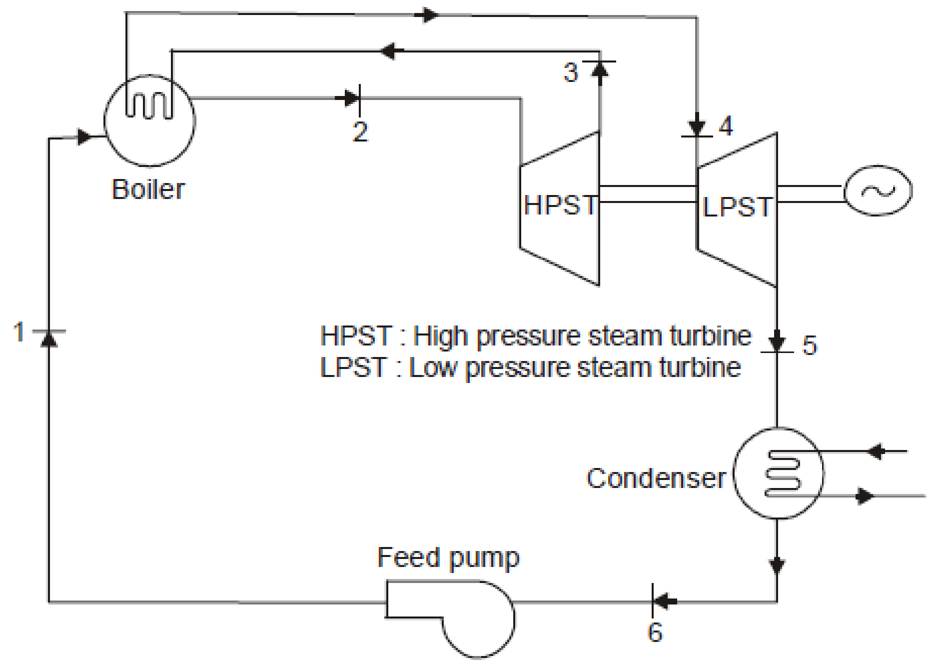


Fig 8.10 Reheat cycle

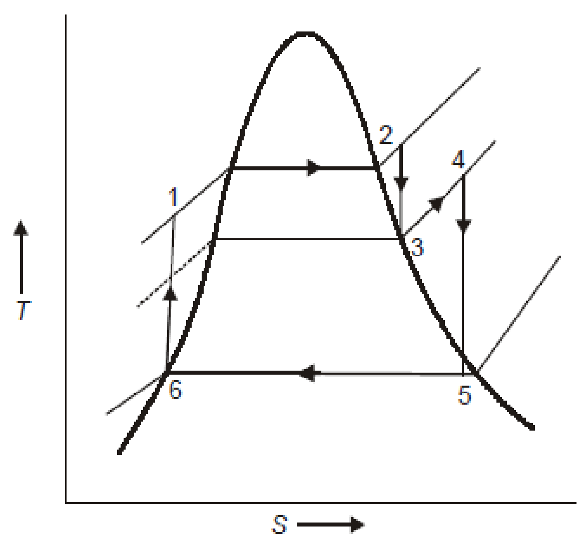


Fig 8.11 T–S representation for reheat cycle

Secondary advantage of reheating is marginal improvement in thermal efficiency when steam pressure is above 100 bar. At low steam pressure reheating does not show gain in cycle thermal efficiency and even the efficiency may be less than that of Rankine cycle due to mean temperature of heat addition being lower. Generally, with modern high pressure boilers and supercritical boilers reheating is essentially employed. Reheating is disadvantageous from economy of plant perspective as the cost of plant increases due to arrangement for reheating and increased condensation requirements due to increased dryness fraction of steam after expansion.

Thermodynamic analysis of reheat cycle as shown on *T–S* diagram may be carried out for estimation of different parameters as below,

$$\text{Total turbine work output} = W_{\text{HPST}} + W_{\text{LPST}}$$

$$\text{Net work, } W_{\text{net}} = (\text{Total turbine work output}) - (\text{Pump work})$$

$$W_{\text{net}} = W_{\text{HPST}} + W_{\text{LPST}} - W_p$$

where different works for m_s mass of steam are,

$$\text{HP steam turbine, } W_{\text{HPST}} = m_s \cdot (h_2 - h_3)$$

$$\text{LP steam turbine, } W_{\text{LPST}} = m_s \cdot (h_4 - h_5)$$

$$\text{Feed Pump, } W_p = (h_1 - h_6) \cdot m_s$$

$$W_{\text{net}} = \{(h_2 - h_3) + (h_4 - h_5) - (h_1 - h_6)\} \cdot m_s$$

Heat supplied for m_s mass of steam; $Q_{\text{add}} = (h_2 - h_1) \cdot m_s + m_s \cdot (h_4 - h_3)$

$$\text{Cycle thermal efficiency, } \eta_{\text{Reheat}} = \frac{W_{\text{net}}}{Q_{\text{add}}}$$

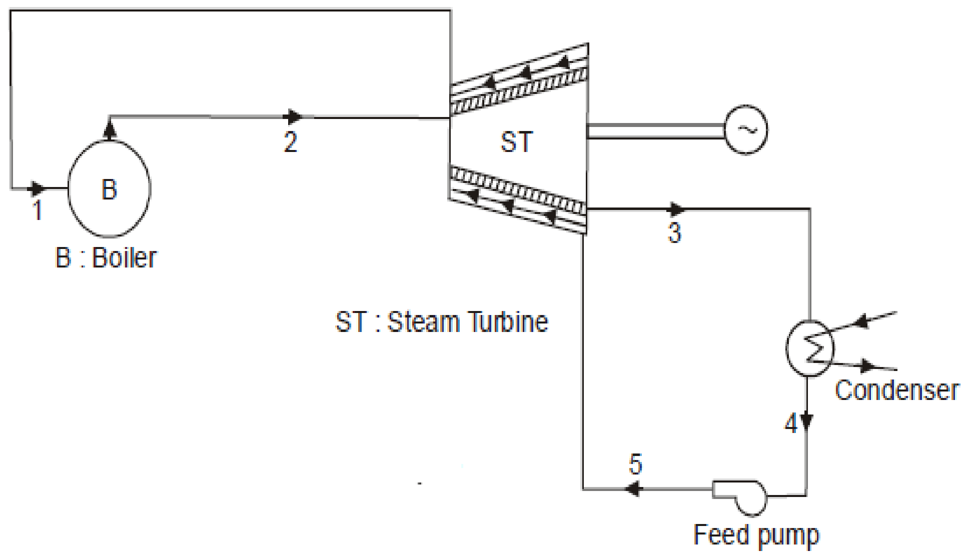
$$\boxed{\eta_{\text{Reheat}} = \frac{\{(h_2 - h_3) + (h_4 - h_5) - (h_1 - h_6)\}}{\{(h_2 - h_1) + (h_4 - h_3)\}}}$$

Specific work output, $\boxed{W_{\text{reheat}} = \{(h_2 - h_3) + (h_4 - h_5) - (h_1 - h_6)\}}$

Generally not more than two stages of reheat are practically employed. Theoretically, the improvement in efficiency due to second reheat is about half of that which results from single reheat. Also more number of reheat stages shall result into superheated steam at turbine exhaust. Thus, mean temperature of heat rejection gets raised and efficiency drops.

REGENERATIVE CYCLE

Regenerative cycle is a modified form of Rankine cycle in which it is devised to increase mean temperature of heat addition so that cycle gets close to Carnot cycle in which all heat addition occurs at highest possible temperature. In regenerative cycle the feed water is heated up so as to reduce the heat addition in boiler and heat addition occur at hotter feed water temperature. Theoretically regenerative cycle arrangement is as shown in Fig. 8.12.



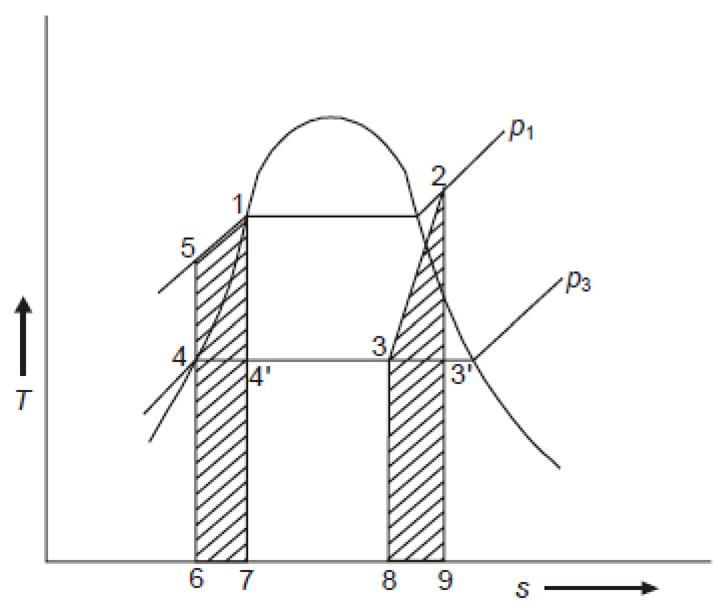


Fig 8.12 Schematic for theoretical regenerative cycle and T-s representation.

Theoretical arrangement shows that the steam enters the turbine at state 2 (temperature T_2) and expands to (temperature T_3) state 3. Condensate at state 5 enters the turbine casing which has annular space around turbine. Feed water enters turbine casing at state 5 and gets infinitesimally heated upto state 1 while flowing opposite to that of expanding steam. This hot feed water enters into boiler where steam generation occurs at desired state, say 2. Feed water heating in steam turbine casing is assumed to occur reversibly as the heating of feed water occurs by expanding steam with infinitesimal temperature difference and is called "regenerative heating". This cycle is called regenerative cycle due to regenerative heating employed in it. Regenerative heating refers to the arrangement in which working fluid at one state is used for heating itself and no external heat source is used for this purpose. Here feed water picks up heat from steam expanding in steam turbine, thus the expansion process in steam turbine shall get modified from 2-3′ ideally to 2-3. Heat picked up by feed water for getting heated up from state 5 to 1 is shown by hatched area A_{17651} on *T-S* diagram. Under ideal conditions for cent per cent heat exchange effectiveness the two areas i.e. A_{29832} indicating heat extraction from steam turbine and A_{17651} indicating heat recovered by feed water shall be same. Thus, *T-S* representation of regenerative cycle indicates that the cycle efficiency shall be more than that of Rankine cycle due to higher average temperature of heat addition.

But there exists serious limitation regarding realization of the arrangement described above. Limitations are due to impossibility of having a steam turbine which shall work as both expander for getting work output and heat exchanger for feed water heating. Also with the heat extraction from steam turbine the state of expanded steam at exhaust pressure shall be extremely wet, hence not desired. Due to these limitations the regenerative cycle is realized employing the concept of bleeding out steam from turbine and using it for feed water heating in feed water heaters.

Feed Water Heaters: The feed water heater refers to the device in which heat exchange occurs between two fluids i.e. steam and feed water either in direct contact or indirect contact. Direct contact feed water heater is the one in which bled steam and feed water come in direct contact. These are also called open feed water heater.

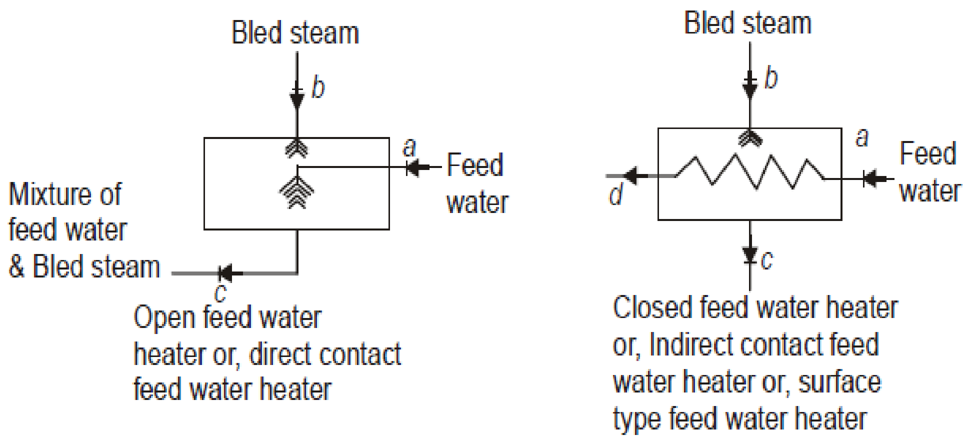


Fig. 8.13 Feed water heaters

In open feed water heater two fluids i.e., bled steam and feed water are at same pressure and adiabatic mixing is assumed to take place. Normally, it is considered that the mixture leaves open feed water heater as saturated liquid. Energy balance upon it shall be as follows,

$$m_a \cdot h_a + m_b \cdot h_b = (m_a + m_b) \cdot h_c$$

where subscripts a,b and c are for feed water, bled steam and mixture of the two as shown in Figure 8.13.

A steam power plant running on Rankine cycle has steam entering HP turbine at 20 MPa, 500°C and leaving LP turbine at 90% dryness. Considering condenser pressure of 0.005 MPa and reheating occurring upto the temperature of 500°C determine,

(a) the pressure at wich steam leaves HP turbine

(b) the thermal efficiency

Solution:

Let us assume that the condensate leaves condenser as saturated liquid and the expansion in turbine and pumping processes are isentropic.

From steam tables,

$$h_2 = h_{\text{at 20 MPa, 500°C}} = 3238.2 \text{ kJ/kg}$$

$$s_2 = 6.1401 \text{ kJ/kg} \cdot \text{K}$$

$$h_5 = h_{\text{at 0.005 MPa, 0.90 dry}}$$

$$h_5 = h_{f \text{ at 0.005 MPa,}} + 0.9 \times h_{fg \text{ at 0.005 MPa}}$$

$$= 137.82 + (0.9 \times 2423.7)$$

$$h_5 = 2319.15 \text{ kJ/kg}$$

$$s_5 = s_{f \text{ at 0.005 MPa,}} + 0.9 \times s_{fg \text{ at 0.005 MPa}}$$

$$= 0.4764 + (0.9 \times 7.9187)$$

$$s_5 = 7.6032 \text{ kJ/kg} \cdot \text{K}$$

$$h_6 = h_{f \text{ at 0.005 MPa}} = 137.82 \text{ kJ/kg}$$

It is given that temperature at state 4 is 500°C and due to isentropic process $s_4 = s_5 = 7.6032$ kJ/kg ·K. The state 4 can be conveniently located on Mollier chart by the intersection of 500°C constant temperature line and entropy value of 7.6032 kJ/kg · K and the pressure and enthalpy obtained. But these shall be approximate.

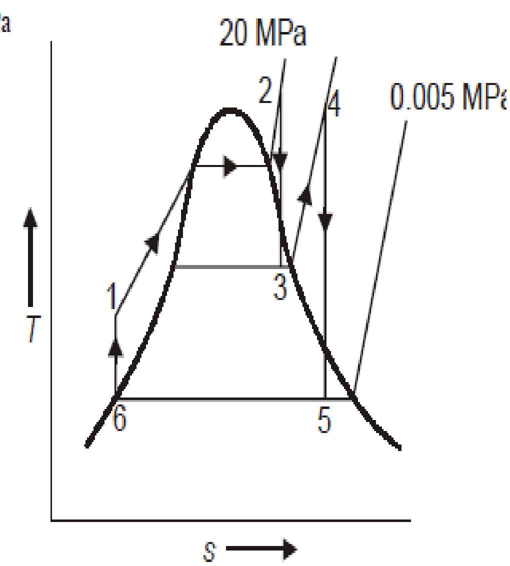


Fig. 8.32

The state 4 can also be located by interpolation using steam table. The entropy value of 7.6032 kJ kg · K lies between the superheated steam states given under, p = 1.20 MPa, $s_{\text{at 1.20 MPa and 500°C}}$ = 7.6759 kJ/kg · K

p = 1.40 MPa, $s_{\text{at 1.40 MPa and 500°C}}$ = 7.6027 kJ/kg · K

By interpolation state 4 lies at pressure

$$= 1.20 + \frac{(1.40-1.20)}{(7.6027-7.6759)}\,(7.6032 - 7.6759)$$

$$= 1.399 \text{ MPa} \approx 1.40 \text{ MPa}$$

Thus, steam leaves HP turbine at 1.4 MPa

$$\text{Enthalpy at state 4, } h_4 = 3474.1 \text{ kJ/kg}$$

For process 2-3, $s_2 = s_3$ = 6.1401 kJ/kg · K. The state 3 thus lies in wet region as $s_3 < s_{g \text{ at 1.40 MPa}}$. Let dryness fraction at state 3 be x_3.

$$s_3 = s_{f \text{ at 1.4 MPa}} + x_3 \cdot s_{fg \text{ at 1.4 MPa}}$$

$$6.1401 = 2.2842 + x_3 \cdot 4.1850$$

$$x_3 = 0.9214$$

$$h_3 = h_{f \text{ at 1.4 MPa}} + x_3 \cdot h_{fg \text{ at 1.4 MPa}}$$

$$= 830.3 + (0.9214 \times 1959.7) = 2635.97 \text{ kJ/kg}$$

$$\text{Enthalpy at 1, } h_1 = h_6 + v_6(p_1 - p_6)$$

$$= h_{f \text{ at 0.005 MPa}} + v_{f \text{ at 0.005 MPa}}\,(20 - 0.005) \times 10^3$$

$$= 137.82 + (0.001005 \times 19.995 \times 10^3)$$

$$h_1 = 157.91 \text{ kJ/kg}$$

$$\text{Net work per kg of steam} = (h_2 - h_3) + (h_4 - h_5) - (h_1 - h_6)$$

$$= 1737.09 \text{ kJ/kg}$$

$$\text{Heat added per kg of steam} = (h_2 - h_1) = 3080.29 \text{ kJ/kg}$$

$$\text{Thermal efficiency} = \frac{\text{Net work}}{\text{Heat added}} = \frac{1737.09}{3080.29} = 0.5639 \text{ or } 56.39\%$$

Pressure of steam leaving HP turbine = **1.40 MPa**
Thermal efficiency = **56.39%**

Ans.

II - UNIT

Steam Boilers:

INTRODUCTION

In simple a boiler may be defined as *a closed vessel in which steam is produced from water by combustion of fuel*.

According to American Society of Mechanical Engineers (A.S.M.E.) a *'steam generating unit'* is defined as :

"A combination of apparatus for producing, furnishing or recovering heat together with the apparatus for transferring the heat so made available to the fluid being heated and vapourised".

The steam generated is employed for the following *purposes* :

(*i*) For generating power in steam engines or steam turbines.

(*ii*) In the textile industries for sizing and bleaching etc., and many other industries like sugar mills ; chemical industries.

(*iii*) For heating the buildings in cold weather and for producing hot water for hot water supply.

The *primary requirements* of steam generators or boilers are :

(*i*) The water must be contained safely.

(*ii*) The steam must be safely delivered in desired condition (as regards its pressure, temperature, quality and required rate).

Boilers Classification:

There are large numbers of boiler designs, but boilers can be classified according to the following criteria:

1. According to Relative Passage of water and hot gases:

Water Tube Boiler: A boiler in which the water flows through some small tubes which are surrounded by hot combustion gases, e.g., Babcock and Wilcox, Stirling, Benson boilers, etc.

Fire-tube Boiler: The hot combustion gases pass through the boiler tubes, which are surrounded by water, e.g., Lancashire, Cochran, locomotive boilers, etc.

2. According to Water Circulation Arrangement:

Natural Circulation: Water circulates in the boiler due to density difference of hot and water, e.g., Babcock and Wilcox boilers, Lancashire boilers, Cochran, locomotive boilers, etc.

Forced Circulation: A water pump forces the water along its path, therefore, the steam generation rate increases, Eg: Benson, La Mont, Velox boilers, etc.

3. According to the Use:

Stationary Boiler: These boilers are used for power plants or processes steam in plants.

Portable Boiler: These are small units of mobile and are used for temporary uses at the sites.

Locomotive: These are specially designed boilers. They produce steam to drive railway engines.

Marine Boiler: These are used on ships.

4. According to Position of the Boilers:

Horizontal, inclined or vertical boilers

5. According to the Position of Furnace

Internally fired: The furnace is located inside the shell, e.g., Cochran, Lancashire boilers, etc.

Externally fired: The furnace is located outside the boiler shell, e.g., Babcock and Wilcox, Stirling boilers, etc.

6. According to Pressure of steam generated

Low-pressure boiler: a boiler which produces steam at a pressure of 15-20 bar is called a low-pressure boiler. This steam is used for process heating.

Medium-pressure boiler: It has a working pressure of steam from 20 bars to 80 bars and is used for power generation or combined use of power generation and process heating.

High-pressure boiler: It produces steam at a pressure of more than 80 bars.

Sub-critical boiler: If a boiler produces steam at a pressure which is less than the critical pressure, it is called as a subcritical boiler.

Supercritical boiler: These boilers provide steam at a pressure greater than the critical pressure. These boilers do not have an evaporator and the water directly flashes into steam, and thus they are called once through boilers.

7. According to charge in the furnace.

Pulverized fuel,

Supercharged fuel and

Fluidized bed combustion boilers.

SELECTION OF A BOILER

While selecting a boiler the following *factors* should be considered :

1. The working pressure and quality of steam required (*i.e.*, whether wet or dry or superheated).
2. Steam generation rate.
3. Floor area available.
4. Accessibility for repair and inspection.
5. Comparative initial cost.
6. Erection facilities.
7. The portable load factor.
8. The fuel and water available.
9. Operating and maintenance costs.

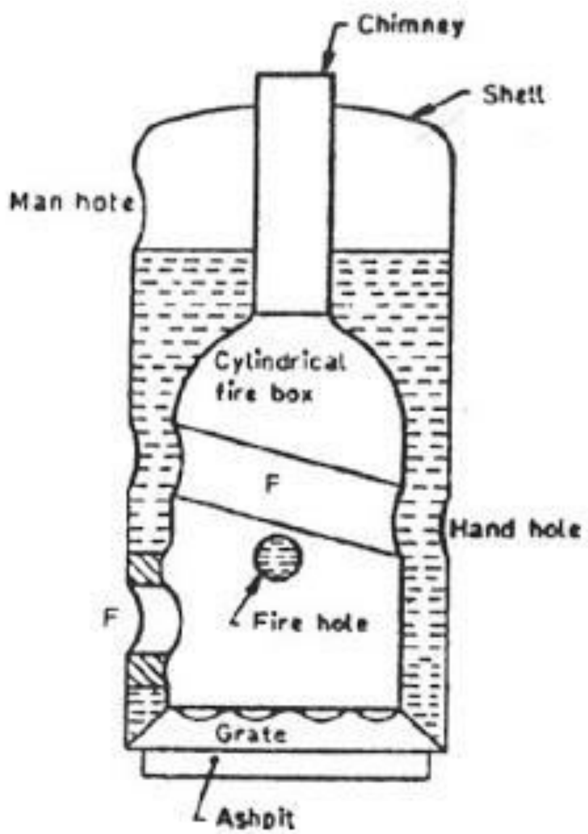
ESSENTIALS OF A GOOD STEAM BOILER

A good boiler should possess the following *features* :

1. The boiler should produce the maximum weight of steam of the required quality at minimum expenses.
2. Steam production rate should be as per requirements.
3. It should be absolutely reliable.
4. It should occupy minimum space.
5. It should be light in weight.
6. It should be capable of quick starting.
7. There should be an easy access to the various parts of the boiler for repairs and inspection.
8. The boiler components should be transportable without difficulty.
9. The installation of the boiler should be simple.
10. The tubes of the boiler should not accumulate soot or water deposits and should be sufficiently strong to allow for wear and corrosion.
11. The water and gas circuits should be such as to allow minimum fluid velocity (for low frictional losses).

Simple vertical boiler:

The image shows the simplest form of an internally fired vertical fire-tube boiler. It does not require heavy foundation and requires very small floor area.



Simple Vertical Boiler

Cylindrical shell:
The shell is vertical and it attached to the bottom of the furnace. Greater portion of the shell is full of water which surrounds the furnace also. Remaining portion is steam space. The shell may be of about 1.25 metres diameter and 2.0 meters height.

Cross-tubes:
One or more cross tubes are either riveted or flanged to the furnace to increase the heating surface and to improve the water circulation.

Furnace (or fire box):
Combustion of coal takes place in the furnace (fire box).

Grate:
It is placed at the bottom of fire box and coal is fed on it for burning.

Fire door:
Coal is fed to the grate through the fire door.

Chimney (or stack):
The chimney (stack) passes from the top of the firebox through the top of the shell.

Manhole:

It is provided on the top of the shell to enable a man to enter into it and inspect and repair the boiler from inside it. It is also, meant for cleaning the interior of the boiler shell and exterior of the compbustion chamber and stack (chimney).

Hand holes:
These are provided in the shell opposite to the ends of each cross tube for cleaning the cross tube.

Ashipt:
It is provide for collecting the ash deposit, which can be removed away at intervals.

Working:
The fuel (coal) is fed into the grate through the fire hole and is burnt. The ashpit placed below the grate collect the ashes of the burning fuel.

The combustion gas flows from the furnace, passes around the cross tubes and escapes to the atmosphere through the chimney.

Water goes by natural circulation due to convection currents, from the lower end of the cross tube and comes out from the higher end.

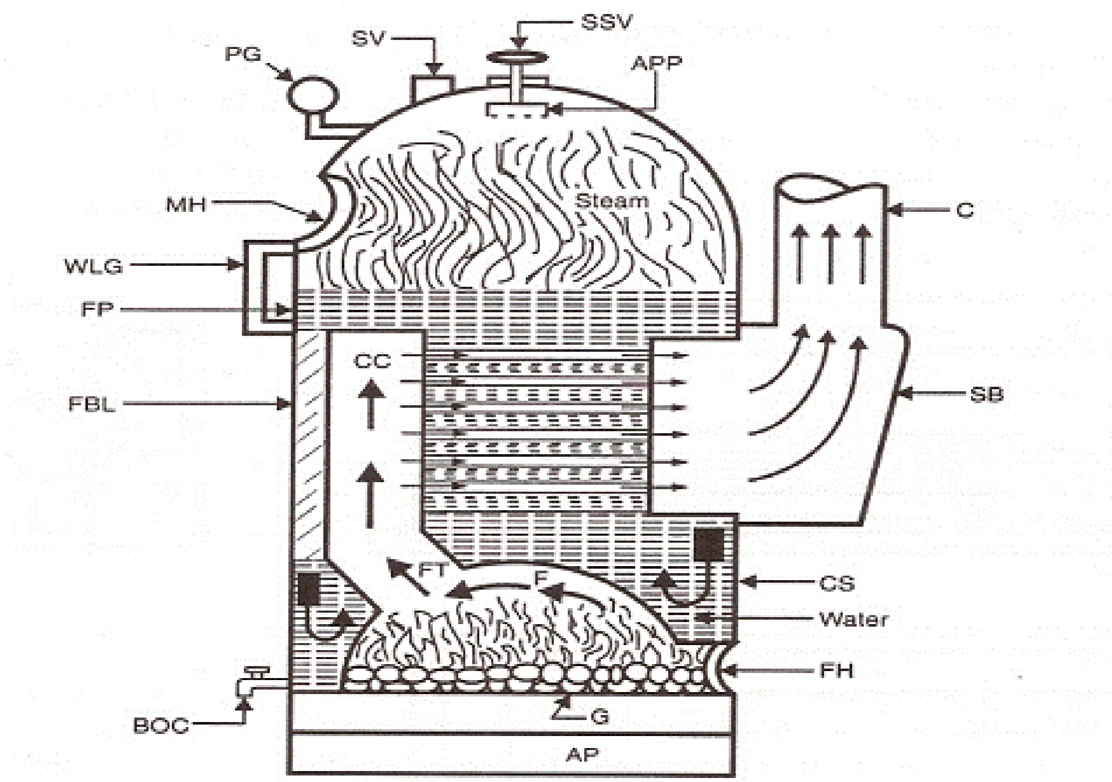
The working pressure of the simple vertical boiler does not exceed 70 N/cm^2.

Cochran boiler

It is a multi-tubular vertical fire tube boiler having a number of horizontal fire tubes. T is the modification of a simple vertical boiler where the heating surface has been increased by means of a number of fire tubes.

It consists of

1. Shell
2. Crate
3. Fire box
4. Flue pipe
5. Fire tubes
6. Combustion chamber
7. Chimney
8. Man-hole



CS = Cylindrical shell
CC = Combustion chamber
FBL = Fir brick lining
F = Furnace (dome shaped)
BOC = Blow off cock
SSV = Steam stop valve
APP = Antipriming pipe
PG = Pressure gauge

FT = Flue tube
SB = Smoke box
C = Chimney
FH = Fire hole
G = Grate
AP = Ash pit
SV = Safety valve
MH = Man hole
WLG = Water level gauge

Cochran boiler

Shell
It is hemispherical on the top, where space is provided for steam.

Grate
It is placed at the bottom of the furnace where coal is burnt.

Fire box (furnace)
It is also dome-shaped like the shell so that the gases can be deflected back till they are passed out through the flue pipe to the combustion chamber.

Flue pipe:
It is a short passage connecting the fire box with the combustion chamber.

Fire tubes:
A number of horizontal fire tubes are provided, thereby the heating surface is increased.

Combustion chamber:
It is lined with fire bricks on the side of the shell to prevent overheating of the boiler. Hot gases enter the fire tubes from the flue pipe through the combustion chamber.

Chimney:
It is provided for the exit of the flue gases to the atmosphere from the smoke box.

Manhole:
It is provided for inspection and repair of the interior of the boiler shell.

Normal size of a Cochran boiler:
Shell diameter – 2.75 meters:
Height of the shell – 6 meters.

Working of the Cochran boiler:
Coal is fed into the grate through the fire hole and burnt. Ash formed during burning is collected in the ashpit provided just below the grate and then it is removed manually.

The host gases from the grate pass through the flue pipe to the combustion chamber. The hot gases from the combustion chamber flow through the horizontal fire tubes and transfer the heat to the water by convection.

The flue gases coming out of fire tubes pass through the smoke box and are exhausted to the atmosphere through the chimney.
Smoke box is provided with a door for cleaning the fire tubes and smoke box.

The following mountings are fitted to the boiler:

Pressure gauge: this indicates the pressure of the steam inside the boiler.

Water gauge: this indicates the water level in the boiler. The water level in the boiler should not fall below a particular level, otherwise the boiler will be over heated and the tubes may burn out.

Safety valve: the function of the safety valve is to prevent an increase of steam pressure in the boiler above its normal working pressure.

Steam stop valve: it regulates the flow of steam supply to requirements.

Blow-off cock: it is located at the bottom of the boiler. When the blow-off cock is opened during the running of the boiler, the high pressure steam pushes (drains) out the impurities like mud, sand, etc., in the water collected at the bottom.

Fusible plug: it protects the fire tubes from burning when the water level in the boiler falls abnormally low.

Salient features of Cochran boiler:

1. The dome shape of the furnace causes the hot gases to deflect back and pass through the flue. The un-burnt fuel if any will also be deflected back.
2. Spherical shape of the top of the shell and the fire box gives higher area by volume ratio.
3. It occupies comparatively less floor area and is very compact.
4. It is well suited for small capacity requirements.

Locomotive Boiler

It is mainly employed in locomotives though it may also be used as a stationary boiler. It is compact and its capacity for steam production is quite high for its size as it can raise large quantity of steam rapidly.

Dimensions and the specifications of the locomotive boilers (made at Chitranjan works in India) are given below :

Barrel diameter	......	2.095 m
Length of the barrel	......	5.206 m
Size of the tubes (superheater)	......	14 cm
No. of ordinary tubes	......	116
Steam capacity	......	9000 kg/h
Working pressure	......	14 bar
Grate Area	......	4.27 m^2
Coal burnt/hr	......	1600 kg
Heating surface	......	271 m^2
Efficiency	......	70%

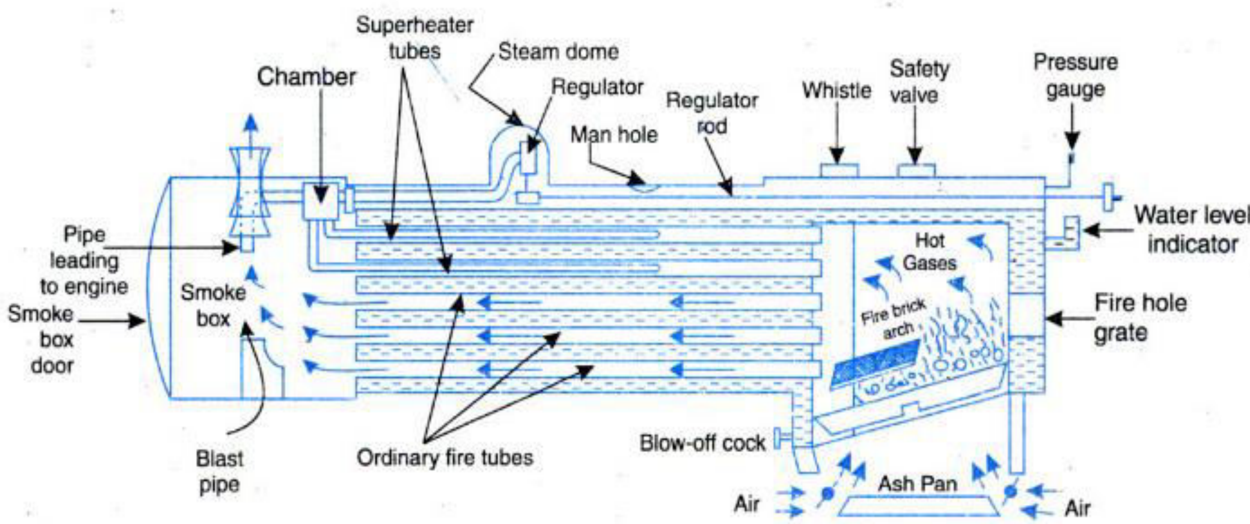


Fig. 11.5. Locomotive boiler.

Refer Fig. 11.5. The locomotive boiler consists of a cylindrical barrel with a rectangular fire box at one end and a smoke box at the other end. The coal is introduced through the fire hole into the grate which is placed at the bottom of the fire box. The hot gases which are generated due to burning of the coal are deflected by an arch of fire bricks, so that walls of the fire box may be heated properly. The fire box is entirely surrounded by water except for the fire hole and the ash pit which is situated below the fire box which is fitted with dampers at its front and back ends. The dampers control the flow of air to the grate. The hot gases pass from the fire box to the smoke box through a series of fire tubes and then they are discharged into the atmosphere through the chimney. The fire tubes are placed inside the barrel. Some of these tube are of larger diameter and the others of smaller diameter. *The superheater tubes are placed inside the fire tubes of larger diameter*. The heat of the hot gases is transmitted into the water through the heating surface of the fire tubes. The steam generated is collected over the water surface.

A dome shaped chamber known as *steam dome* is fitted on the upper part of the barrel, from where the steam flows through a steam pipe into the chamber. The flow of steam is regulated by means of a regulator. From the chamber it passes through the superheater tubes and returns to the superheated steam chamber (not shown) from which it is led to the cylinders through the pipes, one to each cylinder.

In this boiler natural draught cannot be obtained because it requires a very high chimney which cannot be provided on a locomotive boiler since it has to run on rails. Thus some artificial arrangement has to be used to produce a correct draught. As such the draught here is *produced by exhaust steam* from the cylinder which is discharged through the blast pipe to the chimney. When the *locomotive is standing and no exhaust steam is available from the engine fresh steam from the boiler is used for the purpose.*

The various boiler mountings include :

Safety valves, pressure gauge, water level indicator, fusible plug, man hole, blow-off cock and feed check valve.

A locomotive boiler entails the following merits and demerits :

Merits :

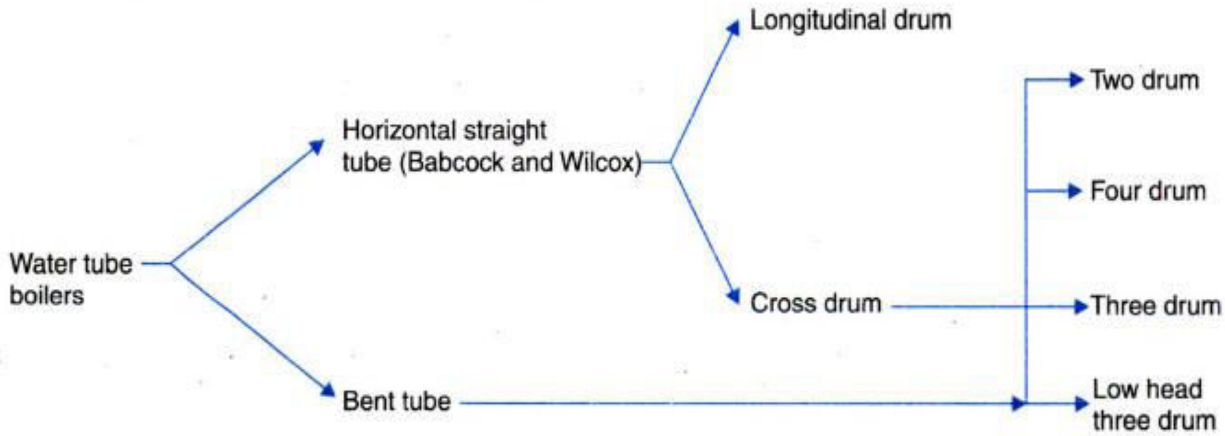
1. High steam capacity.
2. Low cost of construction.
3. Portability.
4. Low installation cost.
5. Compact.

Demerits :

1. There are chances to corrosion and scale formation in the water legs due to the accumulation of sediments and the mud particles.
2. It is difficult to clean some water spaces.
3. Large flat surfaces need bracing.
4. It cannot carry high overloads without being damaged by overheating.
5. There are practical constructional limits for pressure and capacity which do not meet requirements.

WATER TUBE BOILERS

The types of water tube boilers are given below :



Babcock and Wilcox Water-tube Boiler

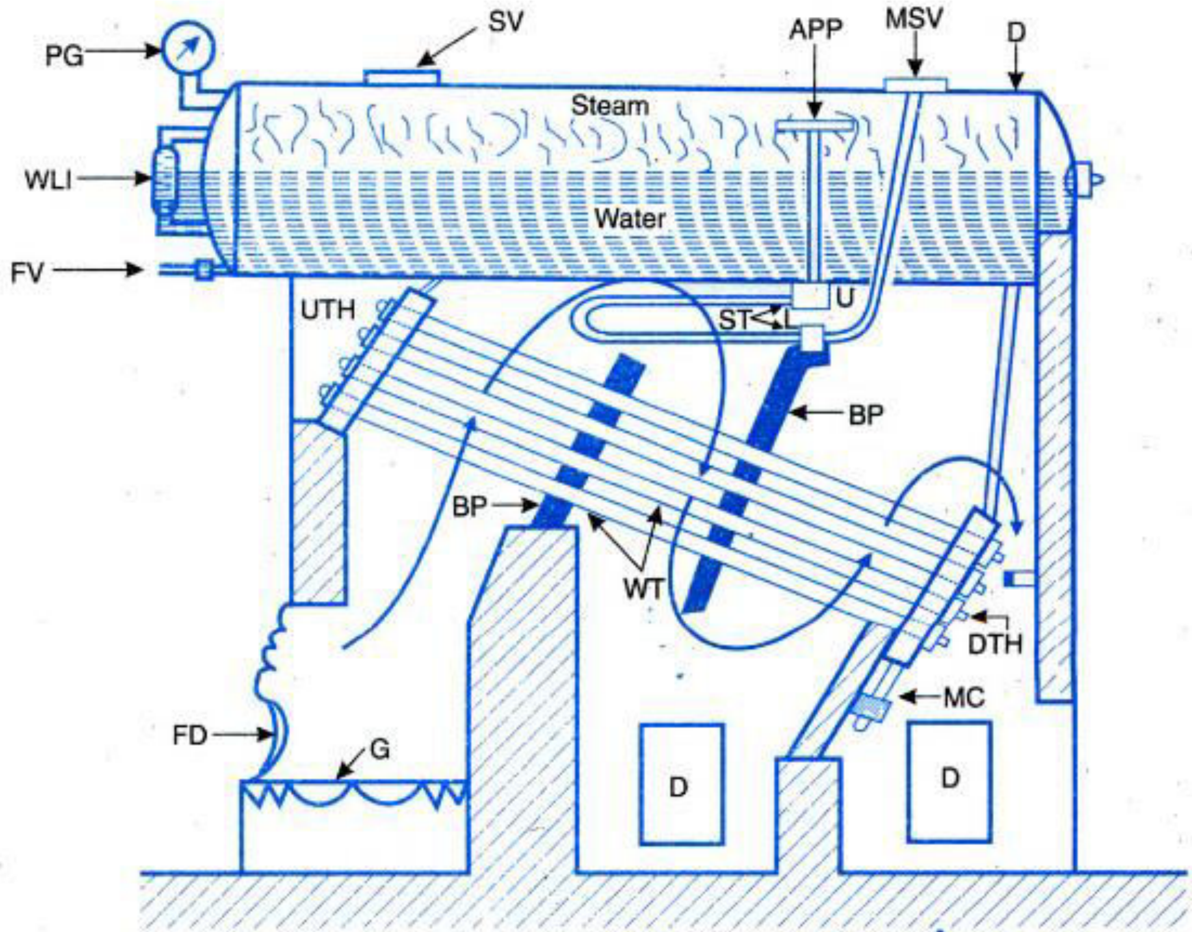
The water tube boilers are used exclusively, when pressure above 10 bar and capacity in excess of 7000 kg of steam per hour is required. Babcock and Wilcox water-tube boiler is an example of horizontal straight tube boiler and may be designed for stationary or marine purposes.

The *particulars* (dimensions, capacity etc.) relating to this boiler are given below :

Diameter of the drum	......	1.22 to 1.83 m
Length	......	6.096 to 9.144 m
Size of the water tubes	......	7.62 to 10.16 cm
Size of superheater tubes	......	3.84 to 5.71 cm
Working pressure	......	40 bar (max.)
Steaming capacity	......	40000 kg/h (max.)
Efficiency	......	60 to 80%

Fig. 11.7 shows a Babcock and Wilcox boiler with longitudinal drum. It consists of a drum connected to a series of front end and rear end header by short riser tubes. To these headers are connected a series of inclined water tubes of solid drawn mild steel.

The angle of inclination of the water tubes to the horizontal is about 15° or more. A hand hole is provided in the header in front of each tube for cleaning and inspection of tubes. A feed valve is provided to fill the drum and inclined tubes with water the level of which is indicated by the water level indicator. Through the fire door the fuel is supplied to grate where it is burnt. The hot gases are forced to move upwards between the tubes by baffle plates provided. The water from the drum flows through the inclined tubes *via* downtake header and goes back into the shell in the form of water and steam *via* uptake header. The steam gets collected in the steam space of the drum. The steam then enters through the antipriming pipe and flows in the superheater tubes where it is further heated and is finally taken out through the main stop valve and supplied to the engine when needed.



D = Drum	PG = Pressure gauge
DTH = Down take header	ST = Superheater tubes
WT = Water tubes	SV = Safety valve
BP = Baffle plates	MSV = Main stop valve
D = Doors	APP = Antipriming pipe
G = Grate	L = Lower junction box
FD = Fire door	U = Upper junction box
MC = Mud collector	FV = Feed valve
WLI = Water level indicator	

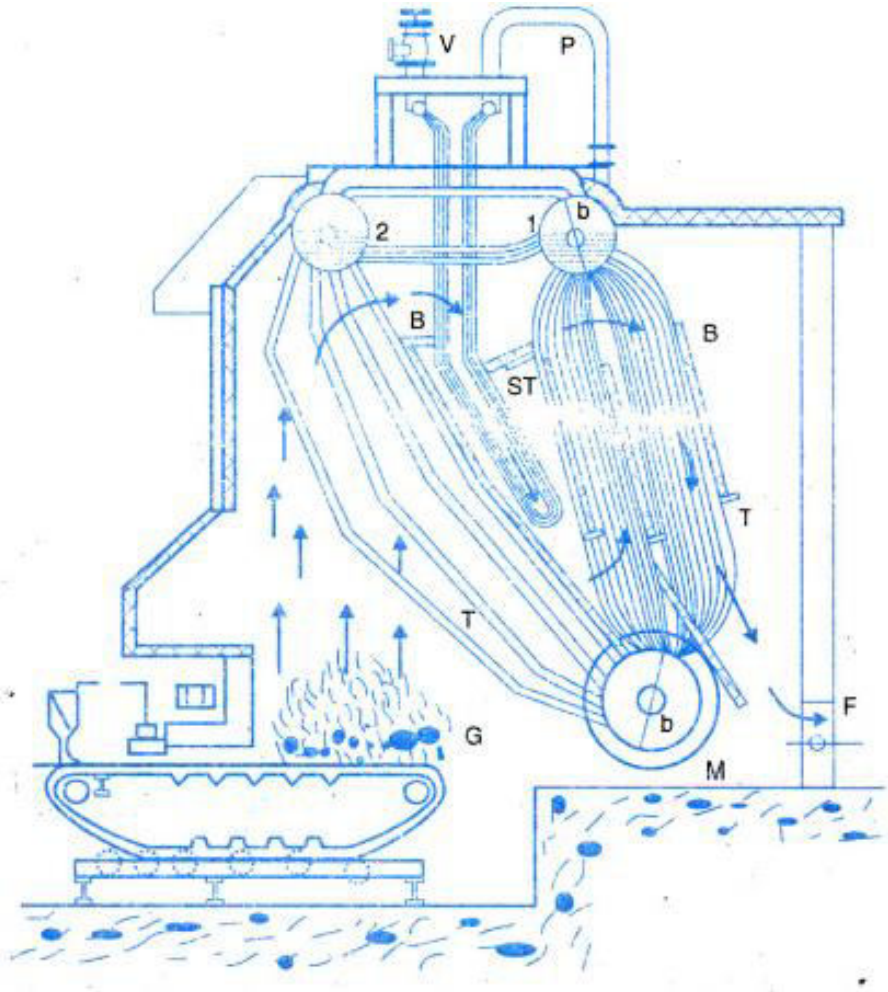
Fig. 11.7. Babcock and Wilcox boiler.

At the lowest point of the boiler is provided a mud collector to remove the mud particles through a blow-down-cock.

The entire boiler except the furnace are hung by means of metallic slings or straps or wrought iron girders supported on pillars. This arrangement enables the drum and the tubes to expand or contract freely. The brickwork around the boiler encloses the furnace and the hot gases.

Stirling Boiler

Stirling water tube boiler is an example of *bent tube* boiler. The main elements of a bent type water tube boiler are essentially drum or drums and headers connected by bent tubes. For large central power stations these boilers are very popular. They have steaming capacities as high as 50000 kg/h and pressure as high as 60 bar.



V = Stop valve
P = Steam pipe
b = Water baffle
B = Baffle wall
ST = Superheater tubes
T = Water tubes
G = Grate
M = Mud drum

Fig. 11.8. Stirling boiler

Fig. 11.8 shows a small-sized stirling water tube boiler. It consists of two upper drums known as *steam drums* and a lower drum known as *mud or water drum*. The steam drums are connected to mud drum by banks of bent tubes. The steam and water space of the steam drums are interconnected with each other, so that balance of water and steam may be obtained. For carrying out cleaning operation a man hole at one end of each drum is provided. The feed water from the economiser (not shown) is delivered to the steam drum-1 which is fitted with a baffle. The baffle deflects the water to move downwards into the drum. The water flows from the drum 1 to the mud drum through the rearmost water tubes at the backside. So the mud particles and other impurities will move to the mud drum, where these particles may be deposited. As this drum is not subjected to high temperature, so the impurities may not cause harm to the drum. The blow-off cock blows off the impurities. The baffle provided at the mud drum deflects the pure water to move upwards to the drum 1 through the remaining half of the water tubes at the back. The water also flows from it to the drum 2 through the water tubes which are just over the furnace. So they attain a higher temperature than the remaining portion of the boiler and a major portion of evaporation takes place in these tubes. The steam is taken from the drum 1 through a steam pipe and then it passes through the superheater tubes where the steam is superheated. Finally the steam moves to the stop valve from where it can be supplied for further use.

The combustion products ensuing from the grate move in the upward and downward directions due to the brickwall baffles and are finally discharged through the chimney into the atmosphere. Fire brick arch gets incandescent hot and helps in combustion and preventing the chilling of the furnace when fire door is opened and cold air rushes in.

The steam drums and mud drum are supported on steel beams independent of the brickwork.

HIGH PRESSURE BOILERS

11.9.1. Introduction

In applications where steam is needed at pressure, 30 bar, and individual boilers are required to raise less than about 30000 kg of steam per hour, *shell boilers are considerably cheaper than the water tube boilers*. Above these limits, shell boilers (generally factory built) are difficult to transport if not impossible. There are no such limits to water tube boilers. These can be site erected from easily transportable parts, and moreover the pressure parts are of smaller diameter and therefore can be thinner. The geometry can be varied to suit a wide range of situations and furnace is not limited to cylindrical form. Therefore, *water tube boilers are generally preferred for high pressure and high output whereas shell boilers for low pressure and low output.*

The modern high pressure boilers employed for power generation are for steam capacities 30 to 650 tonnes/h and above with a pressure upto 160 bar and maximum steam temperature of about 540°C.

11.9.2. Unique Features of the High Pressure Boilers

Following are the unique features of high pressure boilers :

1. Method of water circulation
2. Type of tubing
3. Improved method of heating.

Advantages of High Pressure Boilers

The following are the advantages of high pressure boilers.

1. In high pressure boilers pumps are used to maintain forced circulation of water through the tubes of the boiler. This *ensures positive circulation of water and increases evaporative capacity of the boiler and less number of steam drums will be required.*
2. The *heat of combustion is utilised more efficiently* by the use of small diameter tube in large number and in multiple circuits.
3. *Pressurised combustion* is used which *increases rate of firing of fuel* thus increasing the rate of heat release.
4. Due to compactness less floor space is required.
5. The tendency of scale formation is eliminated due to high velocity of water through the tubes.
6. All the parts are uniformly heated, therefore the danger of overheating is reduced and thermal stress problem is simplified.
7. The differential expansion is reduced due to uniform temperature and this reduces the possibility of gas and air leakages.
8. The components can be arranged horizontally as high head required for natural circulation is eliminated using forced circulation. There is a greater flexibility in the components arrangement.
9. The steam can be raised quickly to meet the variable load requirements without the use of complicated control devices.
10. The *efficiency of plant is increased upto 40 to 42 per cent* by using high pressure and high temperature steam.

LaMont Boiler

This boiler works on a forced circulation and the circulation is maintained by a centrifugal pump, driven by a steam turbine using steam from the boiler. For emergency an electrically-driven pump is also fitted.

Fig. 11.9 shows a LaMont steam boiler. The feed water passes through the economiser to the drum from which it is drawn to the circulation pump. The pump delivers the feed water to the tube evaporating section which in turn sends a mixture of steam and water to the drum. The steam in the drum is then drawn through the superheater. The superheated steam so obtained is then supplied to the prime mover.

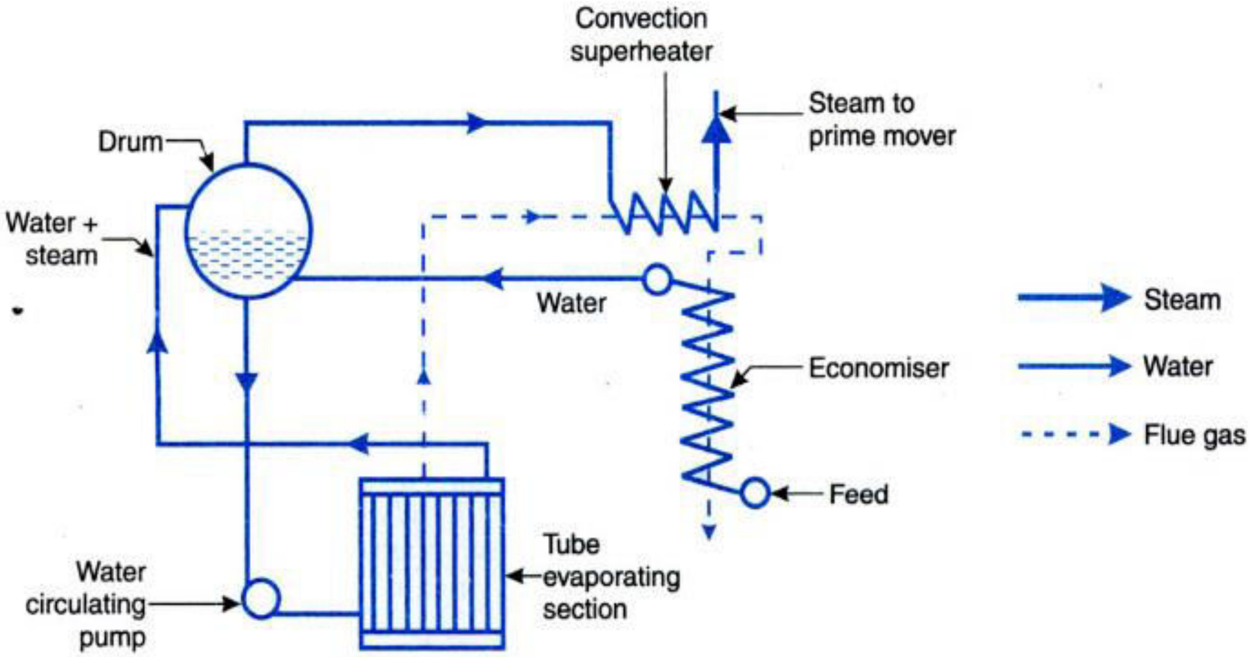


Fig. 11.9. LaMont boiler.

These boilers have been built to generate of 45 to 50 tonnes of superheated steam at a pressure of 130 bar and at a temperature of 500°C.

Benson Boiler

In the LaMont boiler, the main difficult experienced *is the formation and attachment of bubbles on the inner surfaces of the heating tubes. The attached bubbles to the tube surfaces reduce the heat flow and steam generation as it offers high thermal resistance than water film. Benson in 1922 argued that if the boiler pressure was raised to critical pressure (225 atm.), the steam and water have the same density and therefore, the danger of bubble formation can be easily eliminated.* The first high pressure Benson boiler was put into operation in 1927 in West Germany.

This boiler too makes use of forced circulation and uses oil as fuel. It chief novel *principle is that it eliminates the latent heat of water by first compressing the feed to a pressure of 235 bar, it is then above the critical pressure and its latent heat is zero.*

Fig. 11.11 shows a schematic diagram of a Benson boiler. This boiler does not use any drum. The feed water after circulation through the economic tubes flows through the radiant parallel tube section to evaporate partly. The steam water mixture produced then moves to the transit section where this mixture is converted into steam. The steam is now passed through the convection superheater and finally supplied to the prime mover.

Boilers having as high as 650°C temperature of steam had been put into service. The maximum working pressure obtained so far from commercial Benson boiler is 500 atm. The Benson boilers of 150 tonnes/h generating capacity are in use.

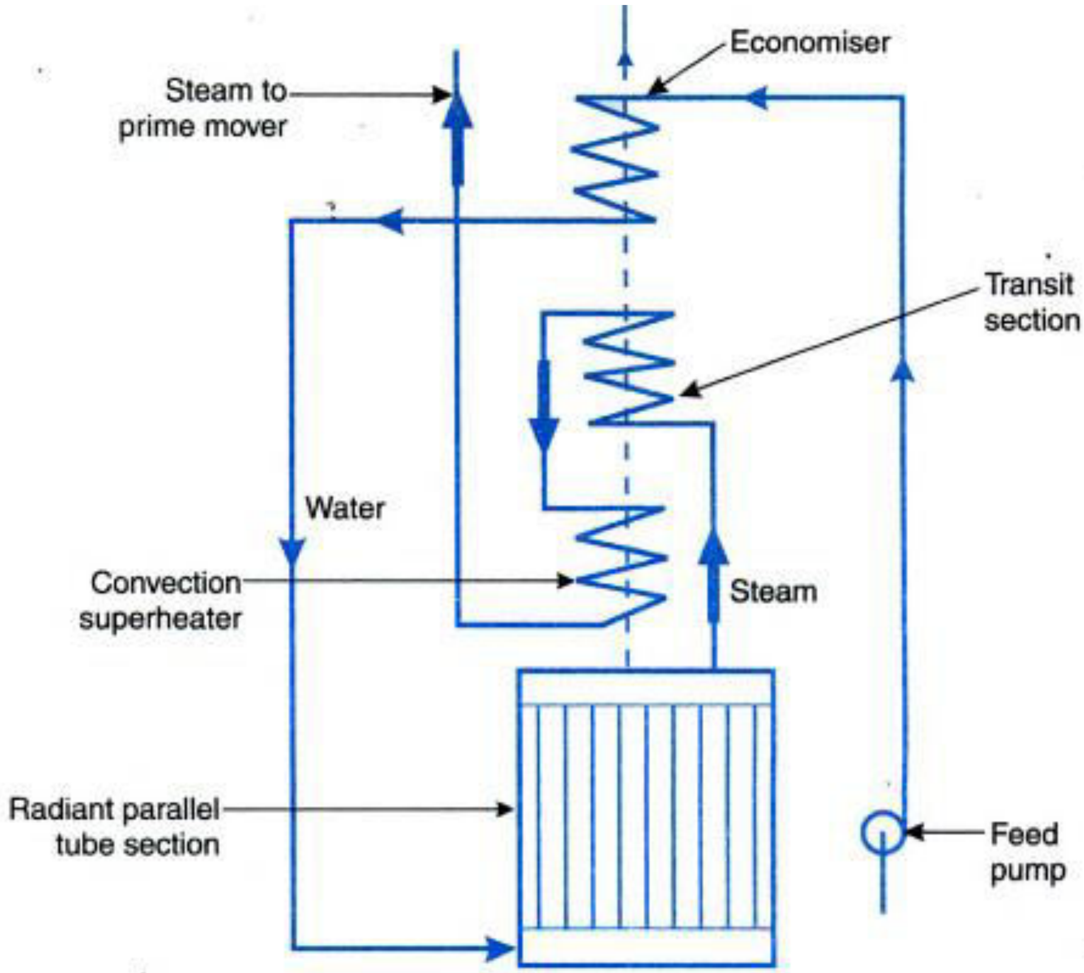


Fig. 11.11. Benson boiler.

Advantages of a Benson Boiler

The Benson boiler possesses the following *advantages* :

1. It can be erected in a comparatively smaller floor area.
2. The total weight of a Benson boiler is 20% less than other boilers, since there are no drums. This also reduces the cost of the boiler.
3. It can be started very quickly because of welded joints.
4. Natural convection boilers require expansion joints but these are not required for Benson boiler as the pipes are welded.
5. The furnace walls of the boiler can be more efficiently protected by using smaller diameter and closed pitched tubes.
6. The transfer of parts of the boiler is easy as no drums are required and majority of the parts are carried to the site without pre-assembly.
7. It can be operated most economically by varying the temperature and pressure at partial loads and overloads. The desired temperature can also be maintained constant at any pressure.
8. The blow-down losses of the boiler are hardly 4% of natural circulation boiler of the same capacity.
9. Explosion hazards are not severe as it consists of only tubes of small diameter and has very little storage capacity.
10. The superheater in a Benson boiler is an integral part of forced circulation system, therefore no special starting arrangement for superheater is required.

Boiler Mountings. These are different fittings and devices which are *necessary for the operation and safety of a boiler*. Usually these devices are *mounted over boiler shell.*

In accordance with the Indian boiler regulation the following *mountings* should be fitted to the boilers.

- Two safety valves
- Two water level indicators
- A pressure gauge
- A steam stop valve
- A feed check valve
- A blow-off cock
- An attachment for inspector's test gauge
- A man hole
- Mud holes or sight holes.

Boilers of Lancashire and Cornish type should be fitted with a *high pressure and low water safety valve.*

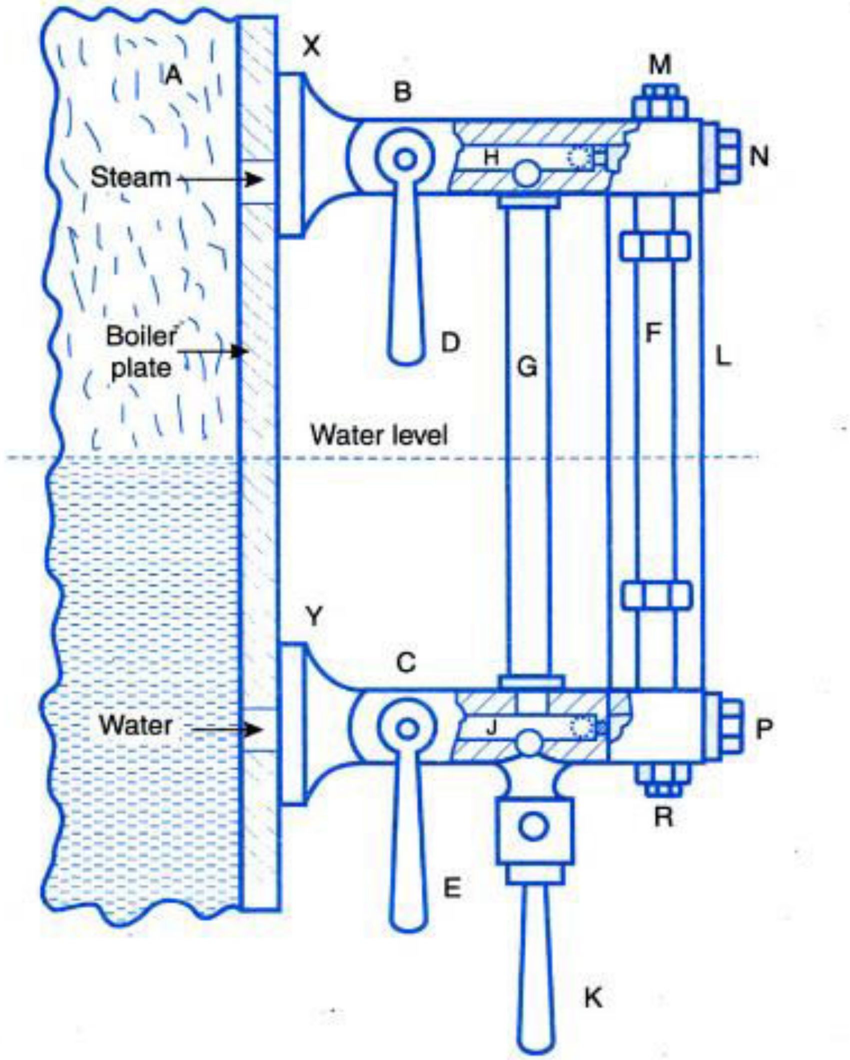
All land boilers should have a *fusible plug* in each furnace.

Boiler Accessories. These are *auxiliary plants required for steam boilers for their proper operation and for the increase of their efficiency*. Commonly used boiler *accessories* are :

- Feed pumps
- Injector
- Economiser
- Air preheater
- Superheater
- Steam separator
- Steam trap.

Water Level Indicator

The function of a water level indicator is to indicate the level of water in the boiler constantly. It is also called *water gauge*. Normally two water level indicators are fitted at the front end of every boiler. Where the boiler drum is situated at considerable height from the floor, the water gauge is often inclined to make the water level visible from any position. When the water being heated in the boiler transforms into steam the level of water in the boiler shell goes on decreasing. For the proper working of the boiler, the water must be kept at safe-level. If the water level falls below the safe level and the boiler goes on producing steam without the addition of feed water, great *damage like crack and leak can occur to the parts of the boiler which get uncovered from water*. This can result in the *stoppage of steam generation and boiler operation*.



A = End plate of boiler
B and C = Hollow gun metal castings
D and E = Cocks
F = Gauge glass
G = Hollow metal column
H and J = Two balls
K = Drain cock
L = Guard glass
M, N, P, R = Screwed caps
X, Y = Flanges

Fig. 12.1. Water level indicator.

Fig. 12.1 shows a Hopkinson's water gauge. It is a common form of glass tube water-level gauge. A is the front end plate of the boiler. F is a very hard glass tube indicating water level and is connected to the boiler plate through stuffing boxes in hollow gun metal castings (B, C) having flanges X, Y for bolting the plate.

For controlling the passage of steam and water cocks D and E are provided. When these cocks are opened the water stands in the glass tube at the same level as in the boiler. K is the drain cock to blow out water at intervals so as not to allow any sediments to accumulate. Upper and lower stuffing boxes are connected by a hollow metal column G. Balls J and H rest in the position shown in the normal working of the gauge. When the glass tube breaks due to rush of water in the bottom passage the balls move to dotted positions and shut off the water and steam. Then the cocks D and E can be safely closed and broken glass tube replaced. M, N, P and R are screwed caps for internal cleaning of the passage after dismantling. L is the guard glass ; it is tough and does not give splinters on breaking. Thus when the gauge glass breaks, and this guard glass which normally will hold flying pieces, also gives way, the pieces will not fly one and *hurt* the attendant.

Pressure Gauge

The function of a pressure gauge is to measure the pressure exerted inside the vessel. The gauge is usually mounted on the front top of the shell or the drum. It is usually constructed to indicate upto double the maximum working pressure. Its dial is graduated to read pressures in kgf/cm^2 (or bar) gauge (*i.e.*, above atmospheric). There are two types of pressure gauges : (*i*) Bourdon tube pressure gauge and (*ii*) Diaphragm type pressure gauge. A pointer, which rotates over a circular graduated scale, indicates the pressure.

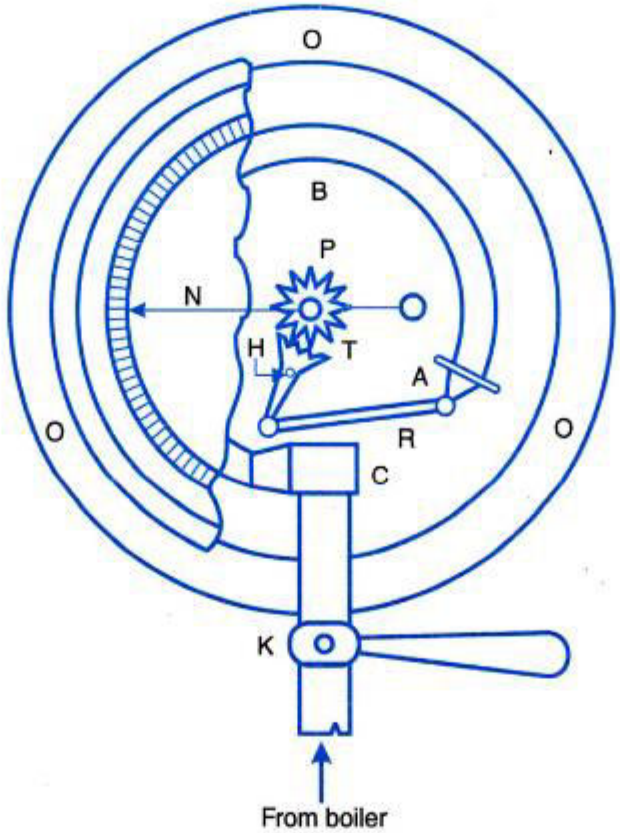


Fig. 12.2. Bourdon pressure gauge.

A pressure gauge is known as *compound pressure gauge* if it is designed in such a fashion so as to measure pressures above and below the atmosphere on the same dial.

Fig. 12.2 shows a Bourdon pressure gauge (single tube) a common type of pressure gauge used. The essential feature of this gauge is the elliptical spring tube which is made of a special quality of bronze and is solid drawn. One end A is closed by a plug and the other is connected with a block C, the block is connected with a syphon tube (which is full of condensed water). The steam pressure forces the water from the syphon tube into elliptical tube and this causes the tube to become circular is cross-section. As the tube is fixed at C, the other end A moves outwards. This outward movement is magnified by the rod R and transmitted to toothed sector T. This toothed sector is hinged at the point H and meshes with the pinion P fixed to the spindle of the pointer N. Thus the pointer moves and registers the pressure on a graduated dial.

The movement of the free end of the elliptical tube *is proportional to the difference between external and internal pressure on the tube.* Since the outside pressure on the tube is atmospheric, the movement of the free end is a measure of the boiler pressure above atmospheric *i.e., gauge pressure.*

Safety Valves

The function of a safety valve is to release the excess steam when the pressure of steam inside the boiler exceeds the rated pressure. As soon as the pressure of steam inside the boiler exceeds the rated pressure the safety valve automatically opens and excess steam rushes out into the atmosphere till the pressure drops down to the normal value. A safety valve is generally mounted on the top of the shell.

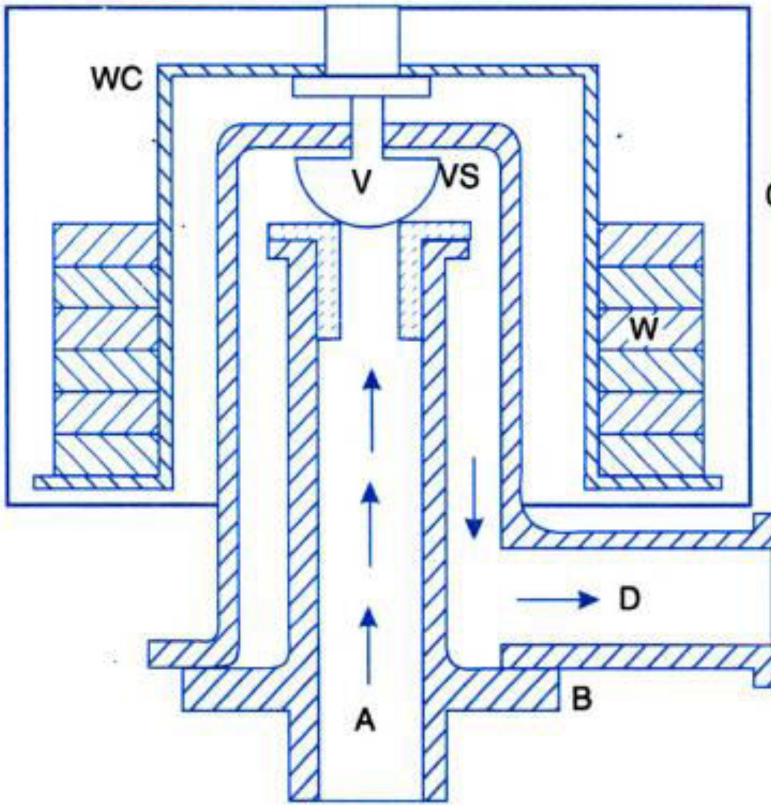
As per boiler regulations every boiler must be fitted at least with two safety valves.

The various types of safety valves are enumerated and discussed as follows :

1. Dead weight safety valve.
2. Lever safety valve.
3. Spring loaded safety valve.
4. High steam and low water safety valve.

Dead Weight Safety Valve

Fig. 12.4 shows a dead weight safety valve. A is the vertical cast iron pipe through which steam pressure acts. B is the bottom flange directly connected to seating block on the boiler shell communicating to the steam space. V is the gun metal valve and VS is the gun metal valve seat. D is another cast iron pipe for discharge of excess steam from the boiler. W are the weights in the form of cylindrical disc of cast iron. WC is the weight carrier carrying the weights W. The cover plate C covers these weights. The steam pressure acts in the upward direction and is balanced by the force of the dead weights W. The total dead-weights consist of the sum of the weights W, weight of the valve V, weight of the weight carrier and weight of the cover plate C.



A = Cast iron pipe
B = Bottom flange
V = Gun metal valve
W = Cast iron weights
WC = Weight carrier
D = Discharge pipe
VS = Gun metal valve seat
C = Cover plate

Fig. 12.4. Dead weight safety valve.

When the steam pressure is greater than the working pressure it lifts the valve with its weights. So the steam escapes from the boiler and the steam pressure thereby decreases.

Merits of dead weight safety valve :

1. Simplicity of design.
2. Gives quite a satisfactory performance during operation.
3. It cannot be easily tempered from the pressure adjustment view-point.

Demerits :

1. Unsuitable for use on any boiler *where extensive vibration and movement are experienced* (*e.g.,* locomotive and marine work).
2. It is not suitable for high pressure boilers because a large amount of weight is required to balance the steam pressure.

Uses. It is mainly used for *low pressures, low capacity, stationary boilers* of the Cornish and Lancashire types.

Lever Safety Valve

Refer Fig. 12.5. It consists of a lever and weight *W*. The valve (made of gun metal) rests on the valve seat (gun metal) which is screwed into the valve body ; the valve seat can be replaced if required. The valve body is fitted on the boiler shell. One end of the lever is hinged while at the other is suspended a weight *W*. The strut presses against the valve on seat against the steam pressure below the valve. The slotted lever guide allows vertical movement to the lever.

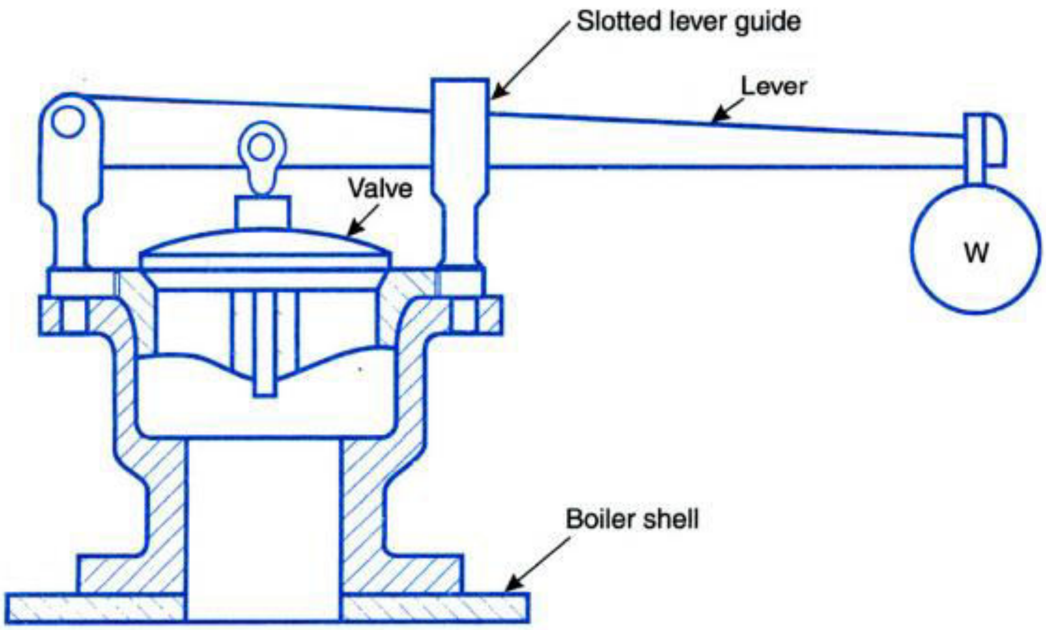


Fig. 12.5. Lever safety valve.

When the steam pressure becomes greater than the normal working pressure, the valve is lifted with the lever and the weight. Consequently, the steam escapes through the passages between the valve and seat and the steam pressure decreases.

The disadvantages of this valve is that it admits of being tempered with, and the effect of a small addition to the weight is magnified considerably in its action on the valve.

Fig. 12.6 shows the loading arrangement on the lever.

Let p = Steam pressure (gauge),

d = Diameter of the valve,

W = Weight suspended on the lever,

W_l = Weight of the lever acting at the centre of gravity G,

W_v = Weight of the valve, and

A = Area of the valve.

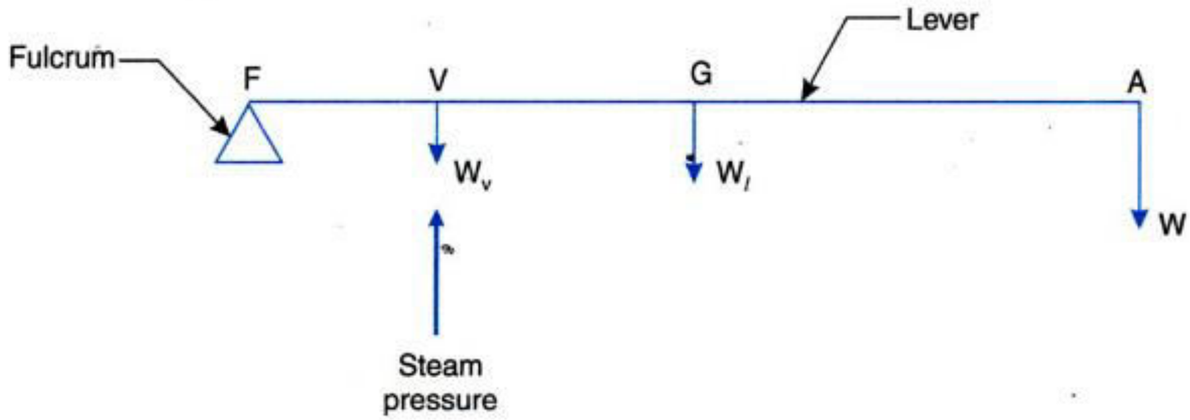


Fig. 12.6. Loading arrangement on the lever.

Taking moments about the fulcrum F, we get

$$W \times AF + W_l \times GF + W_v \times VF = p \times a \times VF, \text{ where } a = \frac{\pi}{4} d^2.$$

Spring Loaded Safety Valve

For locomotives and marine engines both the lever and dead-weight types are unsuitable for abvious reasons, and the valve must be spring loaded, as such valve is *unaffected by vibration or deviations from the vertical*.

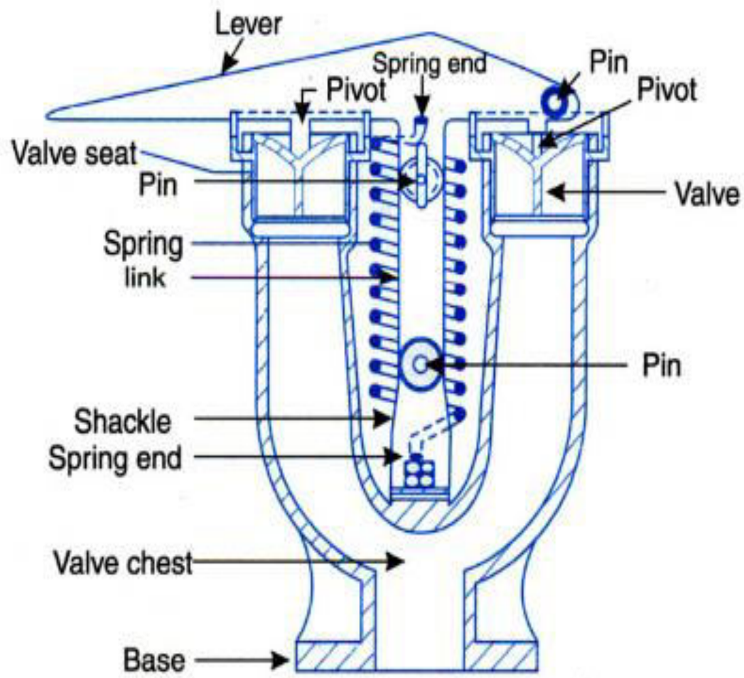


Fig. 12.7. Ramsbottom spring loaded safety valve.

Fig. 12.7 illustrates what is known as Ramsbottom spring loaded safety valve. It consists of two separate valves and seatings having one lever, bearing on the two valves, and loaded by a spring, the spring being placed between the valves. The tension on the spring can be adjusted by the nuts. By pulling or raising the lever the operator/driver can relieve the pressure from either valve separately, and ascertain it is not sticking on the seating.

One *disadvantage* of the spring-loaded safety valve is that *the load on the valve increases as the valve lifts, so that pressure required just to lift the valve is less than that required to open it fully*. From this reason is some cases *it is arranged that the area acted on by the steam is greater when the valve is open than the valve is closed.*

Fusible Plug

The function of a fusible plug is to protect the boiler against damage due to overheating for low water level. It is fitted on the fire box crown plate or over the combustion chamber at its appropriate place.

A common from of fusible plug is illustrated in Fig. 12.9. It consists of a hollow gun metal body screwed into the fire box crown. The body has a hexagonal flange to tighten it into the shell. A gun metal plug having a hexagonal flange is screwed into the gun metal body. There is another hollow gun metal plug separated from the metal plug by an annulus of fusible metal. The fusible metal is protected from fire by flange on the hollow gun metal plug.

Under normal condition when the water-level in the boiler shell is normal, the fusible plug is fully submerged under water. In this case, the heat from the fusible plug is being conducted to water which keeps the fusible metal at an almost constant temperature and *below its melting point.* But when the water level falls below the fusible plug, it gets uncovered from water and is exposed to steam. The heat conduction from the fusible plug to steam is very little compared with that to water. Hence fusible plug becomes overheated and it melts with the result that the hollow gun metal *plug falls down making a hole. The steam and water being under pressure immediately rush to fire box and extinguish the fire.*

The fusible plugs should generally be renewed after a period of about two years as they are liable to become defective over a long period of use (because they are subjected to heat on one side and scale deposits on the other).

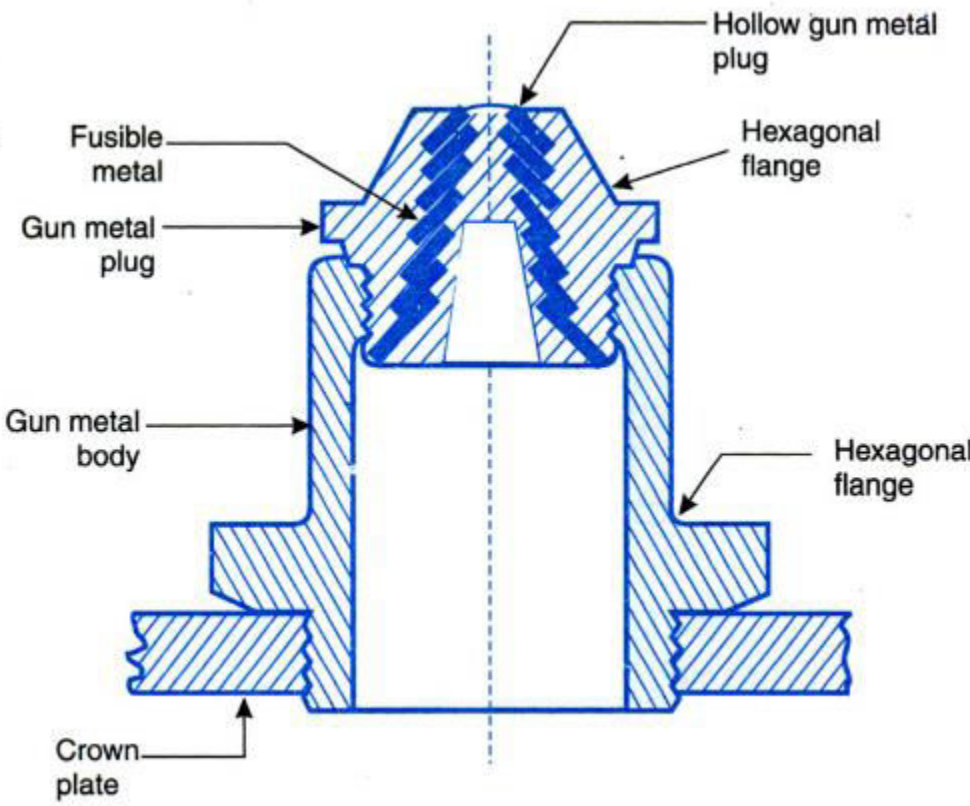


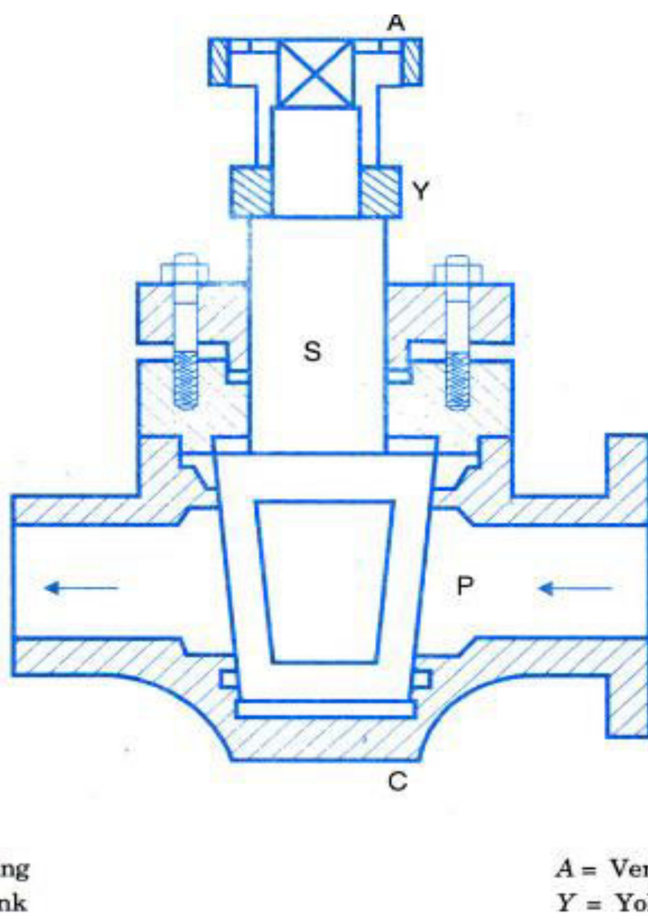
Fig. 12.9. Fusible plug.

Blow-off Cock

A blow-off cock or valve performs the *two functions* : (*i*) *It may discharge a portion of water when the boiler is in operation to blow out mud, scale or sediments perodically.* (*ii*) *It may empty the boiler when necessary for cleaning, inspection and repair.* It is fitted on the boiler shell directly or to a short branch pipe at the *lowest part of the water space*. When more than one boilers are working and they drain in the same waste pipe line, an isolating valve is necessary to prevent the discharge of one boiler, from entering into the other.

Fig. 12.10 shows a common type of plug. The plug P of the cock is conical and fits into the casing C which is packed with asbestos packing in grooves round the top and bottom of the plug. The shank S of the plug passes through a gland and stuffing box in the cover. The plug is held down by a yoke Y and two studs (not shown). A are the vertical slots for fixing the box spanner, on the top of the yoke. The plug spindle S is generally rotated by means of the box spanner.

The plug P has a hole. When this hole is brought in line with the casing hole by rotating the spindle S, the water flows out of it. And the water cannot flow when the solid portion of the plug is infront of casing hole.



C = Casing
S = Shank
P = Plug

A = Vertical slots
Y = Yoke

Fig. 12.10. Blow-off cock.

BOILER ACCESSORIES

The accessories are mounted on the boiler to increase its efficiency. These units are optional on an efficient boiler. The following accessories are normally used on a modern boiler:

(i) Superheater

(ii) Economiser

(iii) Air preheater

(iv) Feed-water pump

(v) Steam injector

(vi) Steam separator

(vii) Steam trap

(viii) Boiler draught equipments

Superheater

It is a heat exchanger in which products of heat of combustion are utilized to dry the wet steam and to make it superheated by increasing its temperature. During superheating of the steam, pressure remains constant, and its volume and temperature increase. A super-heater consists of a set of small-diameter U tubes in which steam flows and takes up the heat from hot flue gases.

The smaller diameter tubes have lower pressure stresses and withstand better. The tube material should be carefully selected, because the tubes are subjected to high temperature, pressure and thermal stresses. The maximum steam temperature at the superheater exit is about 540°C. The superheaters and re-heaters, which are operating at this temperature, are made of special high-strength alloy steels, which have high strength and corrosion resistance.

Superheaters are classified as convective, radiant and of combination types.

In the *convective superheater*, the heat of the hot flue gases is transferred to the surface of the superheater by convection. These are located in the path of hot flue gases.

In a *radiant superheater*, the heat of the combustion is transferred to the surface of the superheater by thermal radiation. These are located in one or more walls of the furnace. These are used in high-pressure boilers.

In a *combination type of superheater*, the heat is transferred to the surface of the tubes by both modes of heat transfer. The radiant superheaters are occasionally used in combination with convective superheaters and are arranged in series.

When the boiler load increases, the rate of energy absorption by the furnace water walls (radiant superheater) does not increase as rapidly as the steam flow rate. Thus, a radiant superheater shows a decrease in steam temperature with an increase in boiler load and steam flow as shown in Fig. 18.11(a) and (b). A convective superheater shows an opposite variation in the steam temperature with change in the boiler load. These opposite characteristics can be utilised to control the final temperature of steam from the combination type of superheaters.

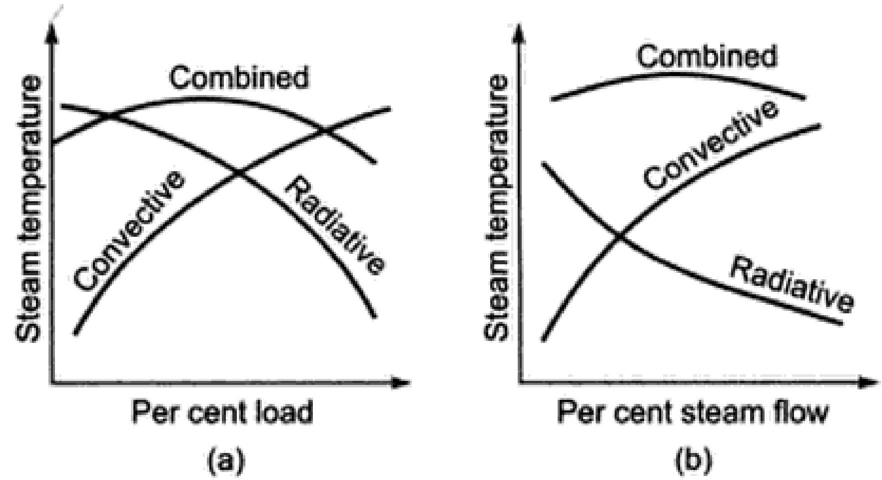


Fig. 18.11 *Superheater characteristics* .

Figure 18.12 shows a schematic of *Surgden's superheater*. It consists of two mild steel headers. The U-shaped steel tubes are connected to these headers.

The steam generated into the boiler passes the valve C and enters the superheater (U) tubes through the intake header. The steam is made dry and superheated in these tubes by supplying heat

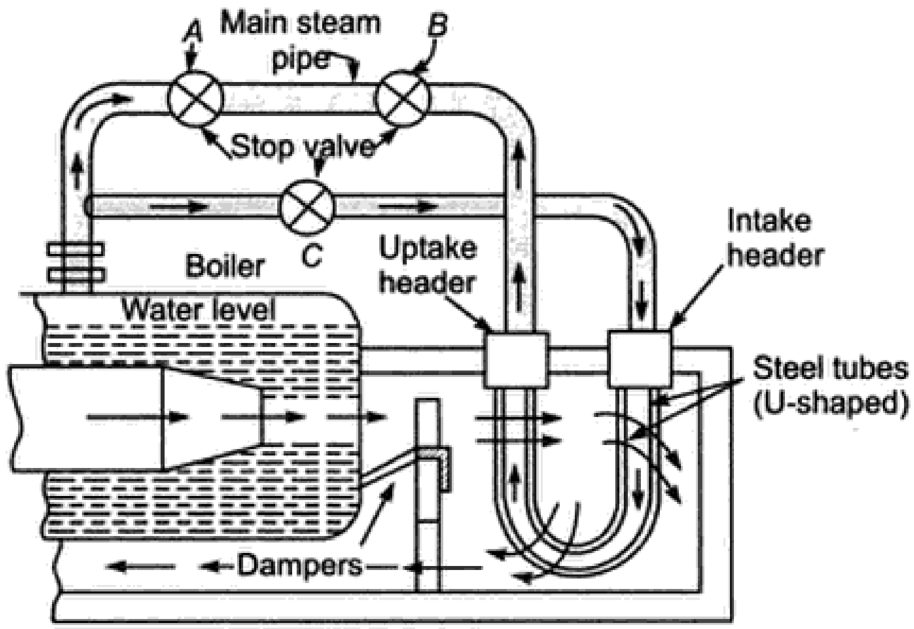


Fig. 18.12 *Superheater*

and then it is taken for use through the valve B via the uptake header. The superheating of steam is controlled by controlling the quantity of flue gases by operating the dampers manually.

If superheated steam is not needed or the superheater is under maintenance, the valves B and C are closed and steam is then taken out through the valve A.

Economizer

The economizer usefully extracts the waste heat of the chimney gases to preheat the water before it is fed into the boiler.

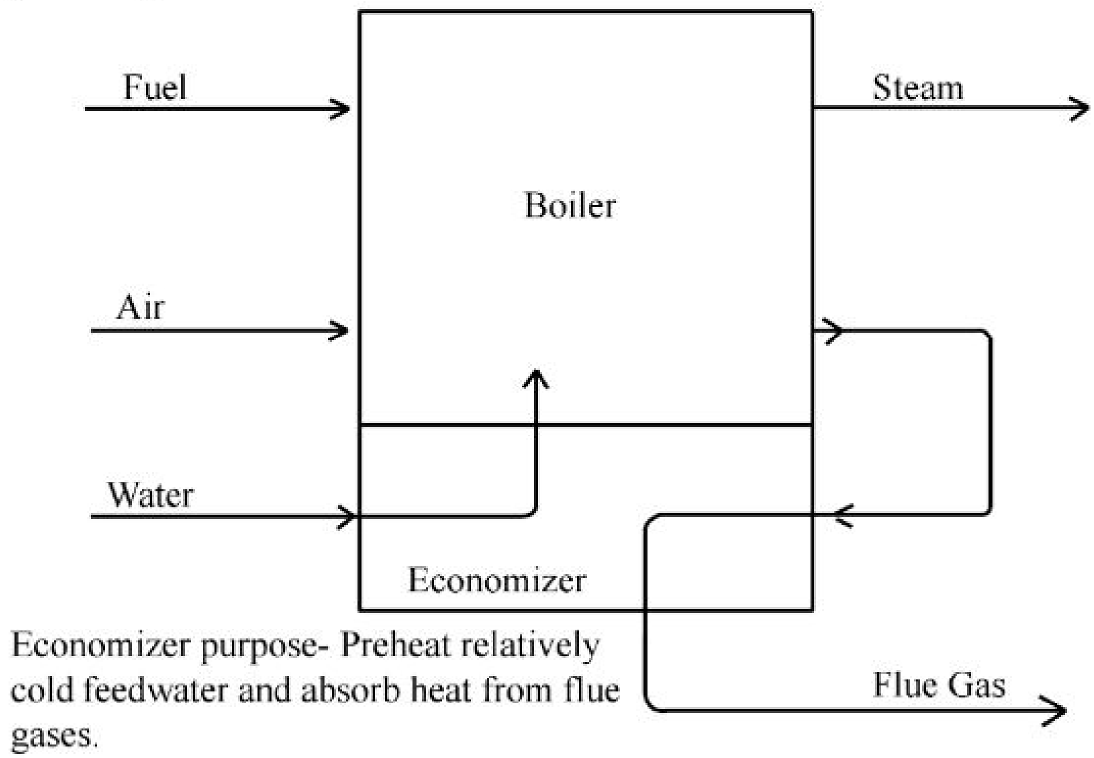


Fig. 3.8: A simple boiler plus economizer

Preheating of the boiler feed water has the following advantages:

i. Fuel saving as waste heat from the flue gas is used for heating the feed water.

ii. Dissolved gases as air and carbon dioxide are removed by pre-heating the feed water reducing corrosion and pitting.

iii. There will be less temperature strain in the boiler plates as the feed water enters the boiler at higher temperature.

iv. Circulation of water is very well maintained as quick evaporation is possible because of hot feed water.

v. This unit improves the overall efficiency of the boiler by reducing the fuel consumption.

Performance of Boiler

Steam boilers are designed to generate the steam according to the requirements. Superheated steam at high pressure is used for power generation, whereas wet steam at low pressure is used for process heating and space heating. The performance of boiler depends upon the following parameters:

1. The condition of the pressure under which boiler has to operate;
2. the quality of steam to be produced;
3. the temperature of feed water supplied to the boiler;
4. grade of fuel burnt;
5. type of firing method;
6. draught used, etc.

Evaporative Capacity

Evaporative capacity gives the information of the rate of generation of the steam in a boiler. It can be provided in one of the following ways:

1. The rate of steam generation at full load in kilogram per hour.
2. The rate of steam generation per unit area of grate (heating surface) in kg/m^2h.
3. The rate of steam generation per unit volume of furnace in kg/m^3h.
4. The rate of steam generation per unit mass of fuel burnt in kilogram per kilogram of fuel.

Equivalent Evaporation

If the boilers are to be compared on the basis of their performance, it becomes necessary that they use the same fuel, have same feed water temperature and working pressure. Therefore, it is necessary to adopt some standard reference for these factors. This is best done by the term "equivalent evaporation". The equivalent evaporation is defined as "the quantity of dry saturated steam that could be generated by the boiler per unit time from feed water at 100°C to dry saturated steam under atmospheric pressure".

$$\text{Equivalent evaporation, } E = \frac{\text{Total heat required to evaporate feed water}}{\text{Latent heat of water at atmospheric pressure}}$$

If the water at its saturation liquid condition is fed into the boiler working at atmospheric pressure, it has to gain only the latent heat of vaporization equivalent to 2257 kJ/kg in order to form the dry and saturated steam. Therefore,

$$E = \frac{H_1}{2257}$$

where, $H_1 = m_s\left(h_s - h_{fw}\right)$ which is equal to heat required to produce m_s kg of steam; h_s is the enthalpy of steam at the generation pressure and temperature in kJ/kg and h_{fw} is the enthalpy of feed water at feed water temperature in kJ/kg.

or,

$$E = \frac{m_e\left(h_s - h_{fw}\right)}{2257}$$

where, m_e is the rate of formation of steam from feed water. If m_e in kg/kg of coal burnt then E will also be in kg/kg of coal burnt.

The enthalpy of the steam produced for unit mass of water can be determined by substituting values of enthalpy from steam table (see Appendix I). One the following formula is used depending upon the quality of steam produced in the boiler.

1. For wet steam $h_s = h_f + xh_{fg}$
2. For dry steam $h_s = h_f + h_{fg} = h_g$
3. For superheated steam $h_s = h_g + C_{ps} \times (T_{sup} - T_s)$

where h_f is the enthalpy of the saturated fluid, h_{fg} is the latent heat, h_g is the enthalpy of dry saturated steam, x is the dryness fraction and C_{ps} is the specific heat at constant pressure.

Calculate the equivalent evaporation of the boiler per kg of coal fired, if the boiler produces 5000 kg of wet steam per hour with a dryness fraction of 0.95 and operating at 10 bar. The coal burnt per hour in the furnace is 5500 kg and feed water temperature is 40°C. *[RGPV Jun 2001]*

Solution: Given $m_s = 5000$ kg/h; $x = 0.95$; $P = 10$ bar; $m_f = 5500$ kg/h; $T_w = 40$°C.
From steam table (temperature basis) at 40°C, we have

$$h_{fw} = h_f = 167.45 \text{ kJ/kg}$$

From steam table (saturation pressure basis) at 10 bar, we have

$$h_f = 762.61 \text{ kJ/kg}$$

$$h_{fg} = 2031.6 \text{ kJ/kg}$$

Enthalpy of wet steam is given by

$$\begin{aligned} h_s &= h_f + xh_{fg} \\ &= 762.61 + 0.95 \times 2031.6 \\ &= 2675.53 \text{ kJ / kg} \end{aligned}$$

Equivalent mass of evaporation is given by

Enthalpy of wet steam is given by

$$\begin{aligned} h_s &= h_f + xh_{fg} \\ &= 762.61 + 0.95 \times 2031.6 \\ &= 2675.53 \text{ kJ / kg} \end{aligned}$$

Equivalent mass of evaporation is given by

$$m_e = \frac{m_s}{m_f} = \frac{5000}{5500} = 0.91$$

Equivalent evaporation is given by

$$\begin{aligned} E &= \frac{m_e\left(h_s - h_{fw}\right)}{2257} \\ &= \frac{0.91 \times (2675.53 - 167.45)}{2257} \\ &= 10.1 \text{ kg / kg of coal} \end{aligned}$$

Boiler Efficiency

Boiler efficiency also serves the purpose of indicating the performance of the boiler. It is defined as the ratio of heat absorbed by the feed water for generation of steam in a certain period to the heat liberated by the fuel in the furnace during the same period. Mathematically, boiler efficiency is given by

$$\eta_{\text{boiler}} = \frac{\text{Heat absorbed by the feed water}}{\text{Heat liberated by fuel in the furnace}}$$

$$\begin{aligned} \eta_{\text{boiler}} &= \frac{m_s\left(h_s - h_{fw}\right)}{m_f \times \text{CV}} \\ &= \frac{m_e\left(h_s - h_{fw}\right)}{\text{CV}} \end{aligned}$$

where, m_s is the rate of steam formation in kg/h; m_f is the rate of coal burnt in kg/h; h_s is the final enthalpy of steam in kJ/kg; h_{fw} is the enthalpy of feed water in kJ/kg; m_s is the rate of formation of steam in kg/h and CV is the calorific value of the fuel.

This shows that the boiler efficiency depends on the hot flue gases discarded through the chimney, radiation from the furnace to the outside, incomplete combustion, presence of moisture in the fuel, unburned fuel going to the ash pit, quality of fuel, etc. Increase in these factors tends to reduce the efficiency of the boiler.

The following readings were taken during the test on a boiler for a period of 1 h: Steam generated 5000 kg; coal burnt 700 kg, CV of coal 31402 kJ/kg and quality of seam 0.92. If the boiler pressure is 1.2 MPa and feed water temperature is 45°C. Find the boiler equivalent evaporation and thermal efficiency.

Solution: Given $m_s = 5000$ kg/h; $m_f = 700$ kg/h; CV = 31402 kJ/kg; $x = 0.92$; $p = 1.2$ MN/m^2 = 12 bar; $T_w = 45$°C.
From steam table (temperature basis) at 45°C, we have

$$h_{fw} = h_f = 188.35 \text{ kJ/kg}$$

From steam table (saturation pressure basis) at 12 bar, we have

$$h_f = 798.43 \text{ kJ/kg}$$
$$h_{fg} = 1984.3 \text{ kJ/kg}$$

Enthalpy of wet steam is given by

$$\begin{aligned} h_s &= h_f + x\,h_{fg} \\ &= 798.43 + 0.92 \times 1984.3 \\ &= 2623.99 \text{ kJ / kg} \end{aligned}$$

Equivalent mass of evaporation is given by

$$m_e = \frac{m_s}{m_f} = \frac{5000}{700} = 7.143$$

Equivalent evaporation is given by

$$E = \frac{m_e\left(h_s - h_{fw}\right)}{2257}$$

$$= \frac{7.143 \times (2623.99 - 188.35)}{2257}$$

$$= 7.7 \text{ kg/kg of coal}$$

Boiler efficiency is given by

$$\eta_{\text{boiler}} = \frac{m_e\left(h_s - h_{fw}\right)}{\text{CV}}$$

$$= \frac{7.143 \times (2623.99 - 188.35)}{31402}$$

$$= 0.554 \text{ or } 55.4\%$$

A boiler is to produce 6000 kg/h of steam at 25 bar and 350°C. The feed water temperature is 40°C. If the calorific value of the fuel oil is 42000 and the expected thermal efficiency is 75%, determine the fuel consumption rate and the equivalent evaporation "from and at 100°C".

Solution: Given $m_s = 6000$ kg/h; $p = 25$ bar; $T_{\text{sup}} = 350°C$; $T_w = 40°C$; CV $= 42000$ kJ/kg; $\eta_{\text{boiler}} = 75\%$.
From steam table (temperature basis) at 40°C, we have

$$h_{fw} = h_f = 167.45 \text{ kJ/kg}$$

From steam table (saturation pressure basis) at 25 bar, we have

$$T_s = 223.94°C$$

$$h_f = 961.96 \text{ kJ/kg}$$

$$h_{fg} = 1839.0 \text{ kJ/kg}$$

$$h_g = 2800.9 \text{ kJ/kg}$$

Assuming specific heat at constant pressure of the steam, C_{ps} as 2.3 kJ/kg, we have
Enthalpy of superheated steam is given by

$$h_s = h_g + C_{ps}\left(T_{sup} - T_s\right)$$

$$= 2800.9 + 2.3 \times (350 - 223.94)$$

$$= 3090.838 \text{ kJ / kg}$$

Boiler efficiency is given by

$$\eta_{\text{boiler}} = \frac{m_s\left(h_s - h_{fw}\right)}{m_f \times \text{CV}}$$

$$0.75 = \frac{6000 \times (3090.838 - 167.45)}{m_f \times 42000}$$

$$m_f = 556.8 \text{ kg / h}$$

Equivalent mass of evaporation is given by

$$m_e = \frac{m_s}{m_f} = \frac{6000}{556.8} = 10.775$$

Equivalent evaporation is given by

$$E = \frac{m_e\left(h_s - h_{fw}\right)}{2257}$$

$$= \frac{10.775 \times \left(3090.838 - 167.45\right)}{2257}$$

$$= 13.96 \text{ kg / kg of oil}$$

The equivalent evaporation of a boiler from and at 100°C is found to be 12 kg of steam per kg of fuel burnt. The calorific value of the fuel is 35000 kJ/kg. Determine the efficiency of the boiler. If the boiler produces 15000 kg/h of steam at 20 bar from feed water at 40°C and the fuel consumption is 1800 kg/h, determine the condition of steam produced.

Solution: Given E = 12 kg/kg of fuel; CV = 35000 kJ/kg; m_s = 15000 kg/h; p = 20 bar; T_w = 40°C; m_s = 1800 kg/h.
Equivalent evaporation is given by

$$E = \frac{m_e\left(h_s - h_{fw}\right)}{2257}$$

$$12 = \frac{m_e\left(h_s - h_{fw}\right)}{2257}$$

$$m_e\left(h_s - h_{fw}\right) = 12 \times 2257$$

$$= 27084 \text{ kJ / kg of coal}$$

Boiler efficiency is given by

$$\eta_{\text{boiler}} = \frac{m_e\left(h_s - h_{fw}\right)}{\text{CV}}$$

Steam Nozzles

13.1 INTRODUCTION

A nozzle is a device of varied cross-sectional area in which the potential energy of steam is converted into the kinetic energy. The increased velocity of steam jet at the exit of the nozzle is due to the decrease in enthalpy of steam.
The nozzles are used in the following engineering applications:

(a) Steam and gas turbines
(b) Jet engines
(c) Rocket motors
(d) In flow measurement
(e) In injectors for pumping feed water into the boiler.
(f) In injectors for removing air from condensers.
(g) In water sprinklers.

TYPES OF STEAM NOZZLES

The following are the three main types of nozzles:
(a) Convergent nozzles (b) Divergent nozzle (c) Convergent-divergent nozzles.

Convergent Nozzle

When the cross-sectional area of the nozzle decreases continuously from entrance to exit, it is called a convergent nozzle. It is shown in Fig. 13.1(a) convergent nozzle is used when the back pressure is equal or more than the critical pressure ratio. It is also used for non-compressible fluids.

Divergent Nozzle

When the cross-sectional area of the nozzle increases continuously from entrance to exit, it is called a *divergent nozzle* as shown in Fig. 13.1(b). When the back pressure is less than critical pressure divergent nozzle is used.

Convergent-divergent Nozzle

When the cross-sectional area of the nozzle first decreases from its entrance to throat, and then increases from throat to exit, it is called a *convergent-divergent nozzle* as shown in Fig. 13.1(c). When the back pressure is less than critical pressure convergent divergent nozzle is used.

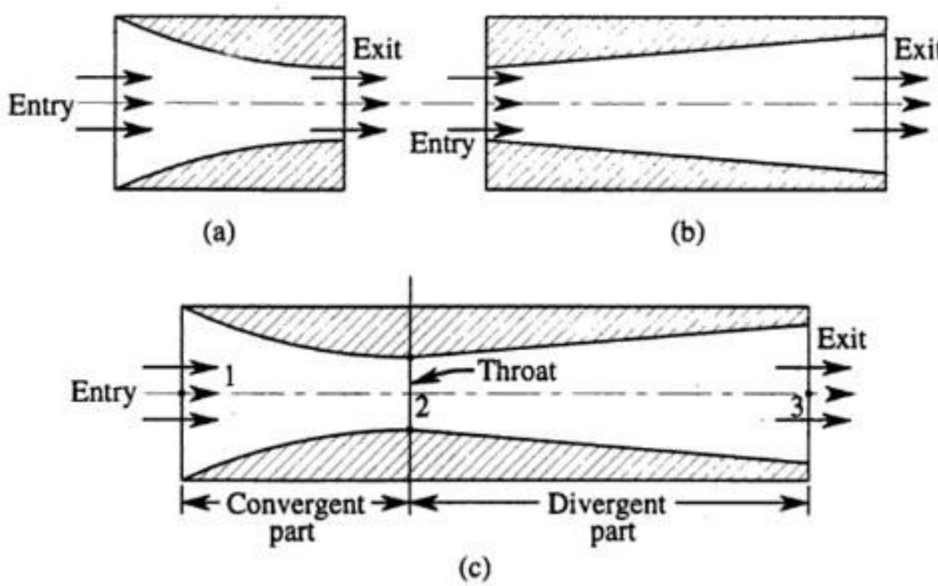

Exit
Entry
(a)
Exit
Entry
(b)
Exit
1
Throat
Entry
2
3
Convergent part
Divergent part
(c)

Fig. 13.1

FLOW OF STEAM THROUGH NOZZLES

The expansion of steam through a nozzle is considered as adiabatic and as there is no external work done during the flow of steam, both the heat transfer and work done are zero. Consider the nozzle shown in Fig. 13.2.

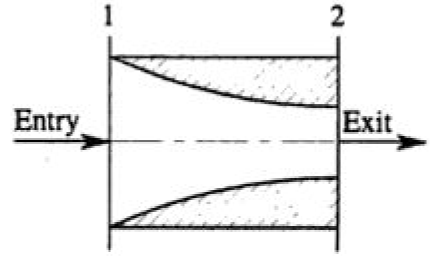


Fig. 13.2

Applying energy equation at Sections 1 and 2 for a flow of 1 kg/s.

$$\frac{V_2^2 - V_1^2}{2 \times 1000} = H_1 - H_2$$

where H_1 and H_2 are the enthalpies at Sections 1 and 2 in kJ/kg. V_1 is the velocity of steam entering the nozzle at Section 1 and V_2 is the velocity of steam at exit from nozzle at Section 2 in m/s.

$$V_2^2 = 2000(H_1 - H_2) + V_1^2 \quad \therefore \quad V_2 = \sqrt{2000(H_1 - H_2) + V_1^2}$$

As the velocity of steam entering the nozzle is very small, V_1 can be neglected.

$$\therefore \qquad V_2 = \sqrt{2000(H_1 - H_2)} = 44.72\sqrt{(H_1 - H_2)} \text{ m/s}$$

If frictional losses are taken into account then

$$V_2 = 44.72\sqrt{(H_1 - H_2)\eta_n} \text{ m/s}$$

CONDITION FOR MAXIMUM DISCHARGE THROUGH NOZZLE

The nozzle is always designed for maximum discharge.

From Eq. 13.4 the rate of mass flow per unit area is given by

$$\frac{m}{A}=\sqrt{2000\frac{n}{n-1}\times\frac{P_1}{v_1}\left[\left(\frac{P_2}{P_1}\right)^{\frac{2}{n}}-\left(\frac{P_2}{P_1}\right)^{\frac{n+1}{n}}\right]}$$

The mass flow per unit area will be maximum at the throat because the throat area is minimum.

It is seen from the above equation that the discharge through a nozzle is a function of $\frac{P_2}{P_1}$ only, as the expansion index is fixed according to the steam supplied to the nozzle.

Therefore, $\frac{m}{A}$ is maximum when

$$\left[\left(\frac{P_2}{P_1}\right)^{\frac{2}{n}}-\left(\frac{P_2}{P_1}\right)^{\frac{n+1}{n}}\right] \text{ is minimum}$$

Differentiating with respect to $\left(\frac{P_2}{P_1}\right)$ and equating to zero for maximum discharge

$$\frac{d}{d\left(\frac{P_2}{P_1}\right)}\left[\left(\frac{P_2}{P_1}\right)^{\frac{2}{n}}-\left(\frac{P_2}{P_1}\right)^{\frac{n+1}{n}}\right]=0$$

$$\therefore\quad \frac{2}{n}\left(\frac{P_2}{P_1}\right)^{\frac{2}{n}-1}-\frac{n+1}{n}\left(\frac{P_2}{P_1}\right)^{\frac{n+1}{n}-1}=0 \quad \text{or,}\quad \frac{2}{n}\left(\frac{P_2}{P_1}\right)^{\left(\frac{2-n}{n}\right)}=\frac{n+1}{n}\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}}$$

$$\text{or,}\quad \frac{2}{n}\times\frac{n}{n+1}=\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}}\left(\frac{P_2}{P_1}\right)^{-\frac{2-n}{n}} \quad \text{or,}\quad \frac{2}{n+1}=\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}-\frac{2-n}{n}}=\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}$$

$$\therefore\frac{P_2}{P_1}=\left(\frac{2}{n+1}\right)^{\frac{n}{n-1}}$$

CRITICAL PRESSURE RATIO

This pressure ratio $\frac{P_2}{P_1}$ is known as critical pressure ratio and depends upon the value of index n.

(a) For air $n = 1.4$ and $\frac{P_2}{P_1} = 0.528$

(b) For wet steam $n = 1.113$ and $\frac{P_2}{P_1} = 0.582$

(c) For dry saturated steam $n = 1.135$ and $\frac{P_2}{P_1} = 0.578$

(d) For superheated or supersaturated steam $n = 1.3$ and $\frac{P_2}{P_1} = 0.546$

Dr. Zeuner's equation for calculating the value of index n is given by

$n = 1.035 + 0.1\,x$ where x = Initial dryness fraction of steam.

Critical Pressure

The pressure at which the area is minimum and discharge per unit area is maximum is known as *critical pressure.*

Physical significance of critical pressure ratio

Let us consider two vessels A and B as shown in Fig. 13.3. The vessel contains steam at a high and steady pressure P_1. The pressure in the vessel B may be

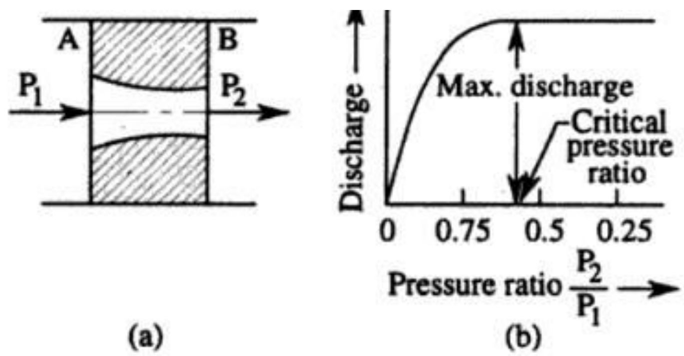


Fig. 13.3

varied at will. The two vessels are connected by a convergent nozzle as shown in Fig. 13.3 (a). Let the pressure P_2 in vessel B be initially equal to P_1. It will be observed that in this position, no flow takes place. Now P_2 is gradually reduced. This will cause the discharge through the nozzle to increase gradually as shown by the curve in Fig. 13.3 (b). As the pressure P_2 approaches the critical value the discharge rate gradually approaches its maximum value. If P_2 is now further reduced below the critical value, the discharge rate will not increase but remains equal the value at the critical pressure.

The critical pressure gives the velocity of steam at the throat equal to the velocity of sound (sonic velocity). The flow of steam in the convergent portion of the nozzle is subsonic. Thus to increase the velocity of steam above sonic velocity (supersonic), by expanding steam below critical pressure, divergent portion is necessary.

EFFECT OF FRICTION IN A NOZZLE

Most of the friction occurs in the diverging part of a convergent-divergent nozzle as the length of the converging part is very small. The effect of friction is to reduce the available enthalpy drop by about 10 to 15 per cent. The velocity of steam will be then $V_2 = 44.72\sqrt{K(H_1 - H_2)}$ where K is the coefficient which allows for friction loss. It is also known as nozzle efficiency (η_n)

$$\therefore \quad V_2 = 44.72\sqrt{(H_1 - H_2)\eta_n}$$

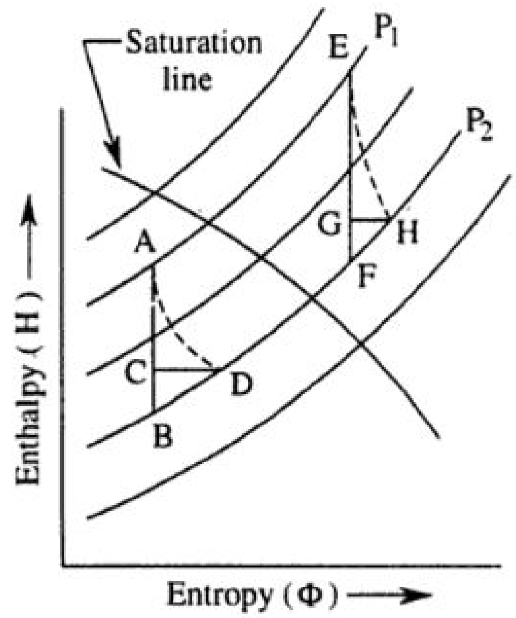


Fig. 13.6

The kinetic energy lost in friction is transformed into heat which makes the steam dry or superheated. ~~Therefore, it~~ effects the final condition of steam issuing from the nozzle. The effect is represented on the $H - \phi$ diagram as shown in Fig. 13.6. Let the point A represent the initial condition of steam. The steam expands from pressure P_1 to P_2. AB is the total enthalpy drop. Enthalpy drop due to friction is BC.

$$K = \frac{\text{Actual enthalpy drop}}{\text{Isentropic enthalpy drop}} = \frac{AC}{AB}$$

If the value of K is known then AC can be obtained from the above relation. The line CD is drawn horizontally to meet P_2 pressure line at D because the expansion must end on the same pressure line P_2. Point D represents the final condition of steam. The value of dryness fraction at D is greater than that at point B. Therefore, friction partially dries the steam.

Let the steam be initially superheated and be represented by the point E.

The steam is expanded up to pressure P_2 and is represented by point F. Actual enthalpy drop is EG. Here the effect of friction is to further superheat the steam. Due to friction, the velocity of steam is reduced but the dryness fraction or degree of superheat is increased.

1. Steam at a pressure of 10 bar and 210°C is supplied to a convergent divergent nozzle with a throat area of 15 cm^2. The exit is below critical pressure. Find the coefficient of discharge, if flow is 7200 kg of steam per hour.

Solution: From steam table at 10 bar and 210°C, $v = 0.2074$ m^3/kg.

As steam is initially superheated $n = 1.3$

Now

$$m = A\sqrt{1000\, n \times \frac{P_1}{v_1} \times \left(\frac{2}{n+1}\right)^{\frac{n+1}{n-1}}} = 15 \times 10^{-4}\sqrt{1000 \times 1.3 \times \frac{10 \times 10^2}{0.2074} \times \left(\frac{2}{1.3+1}\right)^{\frac{1.3+1}{1.3-1}}} = 2.1977.$$

$$\text{Co-efficient of discharge} = \frac{\text{Actual discharge}}{\text{Theoretical discharge}} = \frac{7200}{2.1977 \times 3600} = 0.91$$

2. Dry saturated steam at a pressure of 8 bar enters a convergent divergent nozzle and leaves it at a pressure of 1.5 bar. If the flow is isentropic and if the corresponding expansion index is 1.133, find the ratio of cross-sectional area at exit and throat for maximum discharge.

Solution: Throat pressure $P_2 = P_1\left(\frac{2}{n+1}\right)^{\frac{n}{n-1}} = 8\left(\frac{2}{1.133+1}\right)^{\frac{1.133}{1.133-1}} = 4.6227$ bar.

At 8 bar, from steam tables $\phi_s = 6.6628$ kJ/kg K, $H = 2769.1$ kJ/kg

At 4.6227 bar, $V_s = 0.40366\ m^3/kg$, $h = 627.542\ kJ/kg$

$L = 2117.63\ kJ/kg$, $\phi_w = 1.8308\ kJ/kg\ K$, $\phi_s = 6.8475\ kJ/kg\ K$.

At 1.5 bar, $V_s = 1.1593\ m^3/kg$, $h = 467.11\ kJ/kg$ $L = 2226.5\ kJ/kg$, $\phi_w = 1.4336$, $\phi_s = 7.2233\ kJ/kg\ K$.

For throat, entropy before expansion = entropy after expansion

$$\phi_1 = \phi_2$$

$$\phi_{s_1} = \phi_{w_2} + x_2(\phi_{s_2} - \phi_{w_2}),\ 6.6628 = 1.8308 + x_2(6.8475 - 1.8308)$$

$$\therefore x_2 = 0.963$$

$$H_2 = h_2 + x_2 L_2 = 627.542 + 0.963 \times 2117.63 = 2666.8197\ kJ/kg.$$

Enthalpy drop form entry to throat = $H_1 - H_2$

Velocity at throat $V_2 = 44.72\sqrt{H_1 - H_2} = 44.72\sqrt{2769.1 - 2666.8197} = 452.27\ m/s.$

$$\therefore m = \frac{A_2 V_2}{v_2}, \quad \therefore \quad A_2 = \frac{m \times v_2}{V_2} = \frac{m \times x_2 \times v_{s_2}}{V_2}$$

$$A_2 = \frac{m \times 0.963 \times 0.40366}{452.27} = 0.0008594\ m$$

For exit, $\phi_1 = \phi_3, \phi_{s_1} = \phi_{w_3} + x_3(\phi_{s_3} - \phi_{w_3})$

$$6.6628 = 1.4336 + x_3(7.2233 - 1.4336),\ \therefore x_3 = 0.903$$

$$.H_3 = h_3 + x_3 L_3 = 467.11 + 0.903 \times 2226.5 = 2477.64\ kJ/kg.$$

$\therefore$ Enthalpy drop from entrance to exit = $H_1 - H_3$

Velocity at exit, $V_3 = 44.72\sqrt{H_1 - H_3} = 44.72\sqrt{2769.1 - 2477.64} = 763.468\ m/s.$

Now, $m = \frac{A_3 V_3}{v_3} = \frac{A_3 V_3}{x_3 \times v_{s_3}}$

$$\therefore A_3 = \frac{m \times x_3 \times v_{s_3}}{V_3} = \frac{m \times 0.903 \times 1.1593}{763.468} = 0.001371\ m$$

$$\therefore \frac{\text{Area at exit}}{\text{Area at throat}} = \frac{A_3}{A_2} = \frac{0.001371\ m}{0.0008594\ m} = 1.595$$

A convergent divergent adiabatic steam nozzle is supplied with steam at 10 bar and 250°c.the discharge pressure is 1.2 bar.assuming that the nozzle efficiency is 100% and initial velocity of steam is 50 m/s. find the discharge velocity.

Given Data:-

Initial pressure(p_1)=10bar Initial

Temperature(T_1)=250°c

Exit pressure(p_2)=1.2 bar

Nozzle efficiency(η_{nozzle})=100%

Initial velocity of steam (v_1)=50m/s

Solution:-

From steam table, For 10 bar, 250°c, h_1=2943 KJ/kg s_1=6.926 KJ/kgk

From steam table, For 1.2 bar,

h_{f2} =439.3 KJ/kg ; h_{fg2}=2244.1 KJ/kg;

s_{f2}=1.3 61 KJ/kg K ; s_{fg2}=5.937 KJ/kgK.

Since $s_1=s_{2,}$

S1=sf 2+x2sfg2

$6.926=1.361+x_2(5.937)$

X_2=0 .9373

We know that,

$$h2=hf2+x2hfg2$$

$$= 439.3+(0.9373)2244.1$$

$$h_2 = 2542KJ/Kg$$

Exit velocity $(V2) = Rt[(2000(2943\ \ 2542) + 50^2]$

$$= 896.91m/s.$$

Dry saturated steam at 6.5 bar with negligible velocity expands isentropically in a convergent divergent nozzle to 1.4 bar and dryness fraction 0.956. De termine the final velocity of steam from th e nozzle if 13% heat is loss in friction. Find th e % reduction in the final velocity.

Given data:

Exit pressure (P2) = 1.4 bar

Dryness fract ion (X2) = 0.956

Heat loss = 13%

To Find:

The percent reduction in final velocity

Solution:

From steam table for initial pressure P1 = 6.5bar, take values h_1 =

$$h_1 = 2758.8 KJ/Kg$$

Similarly, at 1.4 bar,

$$h_{fg2} = 2231.9 \text{ KJ/Kg}$$

$$h_{f2} = 458.4 KJ/Kg$$

$$h2 = hf2 + X2\ hfg2$$

$$= 458.4 + (0.956)\ 2231.6$$

$$h_2 = 2592.1 \text{ KJ/Kg}$$

$h_2 = 2592.1$ KJ/Kg

Final velocit y (V2) = Rt (2000(h1-h2))

V2 = 577.39 m/s

Heat drop is 13%= 0.13

Nozzle efficiency (η) = 1- 0.13 = 0.87

Velocity of s team by considering the nozzle efficiency,

y (V2) = Rt (2000(h1-h2)) x η

V2 = 538.55 m/s

= % reductio n in final velocity = 6.72%

'.A convergent divergent nozzle receives steam at 7bar and 200^{o}c and it expands isentropically into a space of 3bar neglecting the inlet velocity calculat e the exit area required for a mass flow of 0.1Kg/sec . when the flow is in equilibrium through all and super saturated with $PV^{1.3}$=C.

Given Data:

Initiall pressure $(P_1) = 7bar = 7\times 10^5 N/m^2$

Initiall temperature $(T_1) = 200^oC$

Press ure $(P_2) = 3bar = 3\times 10^5 N/m^2$

Mass flow rate (m) = 0.1Kg/sec

$PV^{1.3} = C$

To Find:

Exit area

Solution:

From st eam table for P1 = 7bar and T1 = 200^{o}C V_1 =

0.2999

$h_1 = 2844.2$

S1 = 6.886

Similarly for P2 = 3bar

Vf2 = 0.001074 Vg2 = 0.60553 hf2 =

561.5 hfg2 = 2163.2

Sf2 = 1.672 Sfg2 = 5.319

We know that, S1 = S2 = St

$S_1 = S_{f2} + X_2 S_{fg2}$

$6.886 = 1.672 + X_2 (5.319)$ X_2 =

0.98

$$h2 = hf2 + X2\ hfg2$$

$$h_2 = 561.5 + 0.98\ (2163.2)$$

(i) <u>Flow is in equilibriu m through all:</u>

$$V_2=569.56$$

$$v_2 = X_2 \times v_{g2}$$

$$= 0.98 \times 0.60553 = 0.5934$$

$$V_2 = \sqrt{2000\,(h_1 - h_2)}$$

$$V_2 = \sqrt{2000\,(2844.2 - 2681.99)}\ V_2 =$$

569.56

$$v_2 = X_2 \times v_{g2}$$

$= 0.98 \times 0.60553 = 0.5934$

$$m = \frac{[(A]_2 \times V_2)}{v_2}$$

$$A_2 = \frac{[m \times V_2)}{v_2} = \frac{0.5934 \times 0.1}{569.56}$$

$$A_2 = 1.041 \times 10^{-4} m^2$$

(ii) <u>For saturated flow:</u>

$$v_2 = \sqrt{\frac{2n}{n-1}\ (P_1 v_1)(1 - (\frac{P_2}{P_1})^{\frac{n-1}{n}}}$$

$$v_2 = \sqrt{\frac{2(1.3)}{1.3-1}\ (7 \times 10^5 \times 0.2999)\ (1 - \frac{3 \times 10^5}{7 \times 10^5})^{\frac{1.2-1}{1.2}}}$$

$v2 = 568.69$ m/s

specific volume of steam at exit. For super saturated flow, $P_1 v_1^n = P_2$

$$\left(\frac{v_2}{v_1}\right)^n = \frac{P1}{P2}$$

$$\left(\frac{v_2}{v_1}\right)^n = \frac{P1}{P2}$$

$$v_2 = \left(\frac{7}{3}\right)^{\frac{1}{1.3}} \times 0.2999$$

$$v_2 = 0.5754$$

$$A_2 = \frac{[m \times V_2)}{v_2}$$

$$= \frac{0.1 \times 0.5754}{568.69}$$

$$A_2 = 1.011 \times 10^{-4} m^2$$

III - UNIT

Steam Turbine

14.1 INTRODUCTION

In steam engines, the pressure energy of steam is utilised. Steam engines can be operated without any drop in pressure in the cylinder, but the operation of steam turbine purely depends on the dynamic action of steam.

The first steam turbine was made by Hero of Alexandria in about 120 BC. This turbine worked on pure reaction principle and had no provision for driving anything. It consisted of a hollow ball mounted on between the two pivots. Steam which was produced in a cauldron placed beneath the turbine was supplied through one pivot. Two converging nozzles were provided in the ball. The steam was expanded through these nozzles to the atmosphere which caused reactive force on the ball and thus initiated rotation.

It has undergone several changes in its design since. The steam turbine has been used as a prime mover in all thermal power stations. Now, single steam turbine of 1000 MW capacity is built in many countries.

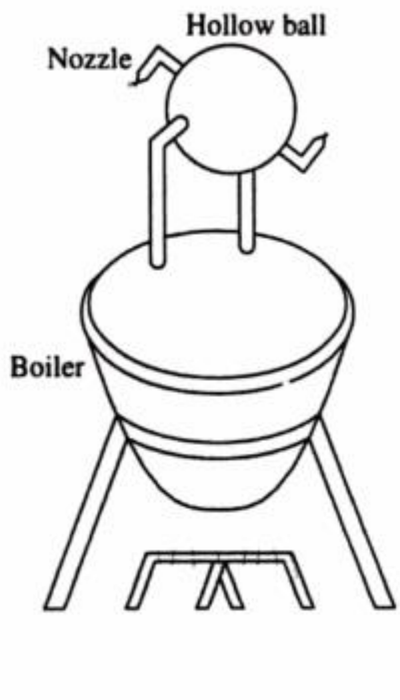


Fig. 14.1 *Working of the hero's turbine*

CLASSIFICATION OF TURBINES

Steam turbines are mainly divided into two groups:
(a) Impulse turbine and (b) impulse reaction turbine (in practice known as reaction turbine).

Impulse Turbine

In impulse turbine, steam coming out at a very high velocity through the fixed nozzle strikes the blades fixed on the periphery of a rotor. The blades change the direction of the steam flow without changing its pressure. The force due to change of momentum causes the rotation of the turbine shaft.

The examples of impulse turbine are De-Laval, Curties and Rateau.

Impulse-Reaction Turbine

In an impulse reaction turbine, steam expands both in fixed and moving blades continuously as the steam passes over them. The pressure drop occurs continuously over both moving and fixed blades.

The example of such turbine is Parsons's turbine.

DIFFERENCE BETWEEN IMPULSE AND REACTION TURBINE

(a) In an impulse turbine the steam completely expands in the nozzle and its pressure remains constant during its flow through the blade passages.

In reaction turbines steam expands partially in the nozzle and further expansion takes place in the rotor blades.

(b) The relative velocity of steam passing over the blade of impulse turbine remains constant (assuming there is no friction).

The relative velocity of steam passing over the blade of reaction turbine increases as the steam passing over the blade expands.

(c) The impulse turbine blades are symmetrical but the reaction turbine blades are asymmetrical.

(d) The pressure on both ends of the moving blade of an impulse turbine is the same but in a reaction turbine it is different.

(e) The number of stages required for reaction turbine are more compared with impulse turbine for the same power developed as the pressure drop in each stage is small.

(f) The blade efficiency curve for the reaction turbine is more flat compared to that of the impulse turbine.

(g) The steam velocity in a reaction turbine is not very high and therefore the speed of the turbine is low.

SIMPLE IMPULSE TURBINE

The simple impulse turbine is called De-Laval turbine after the name of its inventor. It consists of one set of nozzles and a set of moving blades as shown in Fig. 14.2. The expansion of steam from its initial pressure to final pressure takes place in one set of nozzles. Due to high drop in pressure in the nozzles the velocity of the steam increases in the nozzle.

The steam leaves the nozzle with a very high velocity and strikes the blades mounted on a wheel with this high velocity. The pressure of the steam when it moves over the blades remains constant but the velocity decreases.

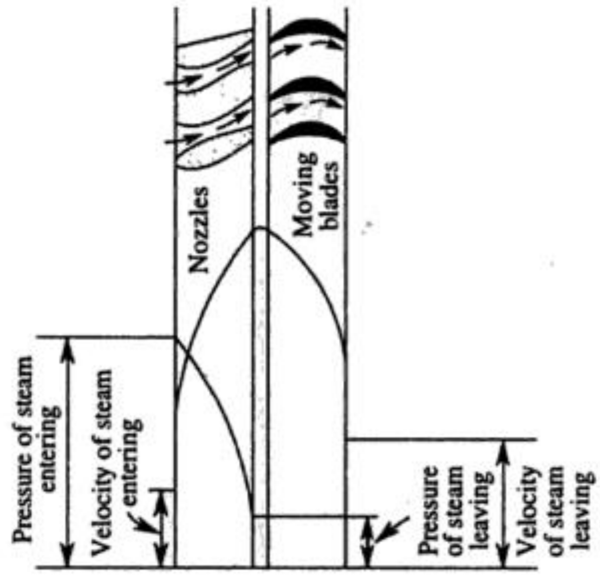


Fig. 14.2 *Diagrammatic sketch of a simple impulse turbine*

But this turbine is not commonly used due to the following disadvantages.

(a) Since all the kinetic energy of the high velocity steam has to be absorbed in only one ring of moving blades, so the velocity of the turbine is too high i.e. up to 30,000 r.p.m. for practical purposes.

(b) The veloctiy of steam at exit is sufficiently high which means that there is a considerable loss of kinetic energy.

VELOCITY DIAGRAM FOR MOVING BLADES FOR AN IMPULSE TURBINE

Figure 14.3 shows the velocity diagram of a single stage turbine.

V_1 = Absolute velocity of steam

at inlet in m/s

α_1 = Nozzle inlet angle

u = Blade velocity in m/s

V_{r_1} = Relative velocity of steam at inlet in m/s

V_{w_1} = Tangential velocity of steam at inlet m/s.

V_{a_1} = Axial velocity of steam at inlet in m/s

β_1 = Blade inlet angle

β_2 = Blade outlet angle

V_{r_2} = Relative velocity of steam at outlet in m/s.

V_{w_2} = Tangential velocity of steam at outlet in m/s.

V_{a_2} = Axial velocity of steam at outlet in m/s.

K = Blade velocity coefficient $= \frac{V_{r_2}}{V_{r_1}}$

V_2 = Absolute velocity of steam at outlet in m/s.

α_2 = Angle made by absolute velocity V_2 with the tangent of the wheel at outlet.

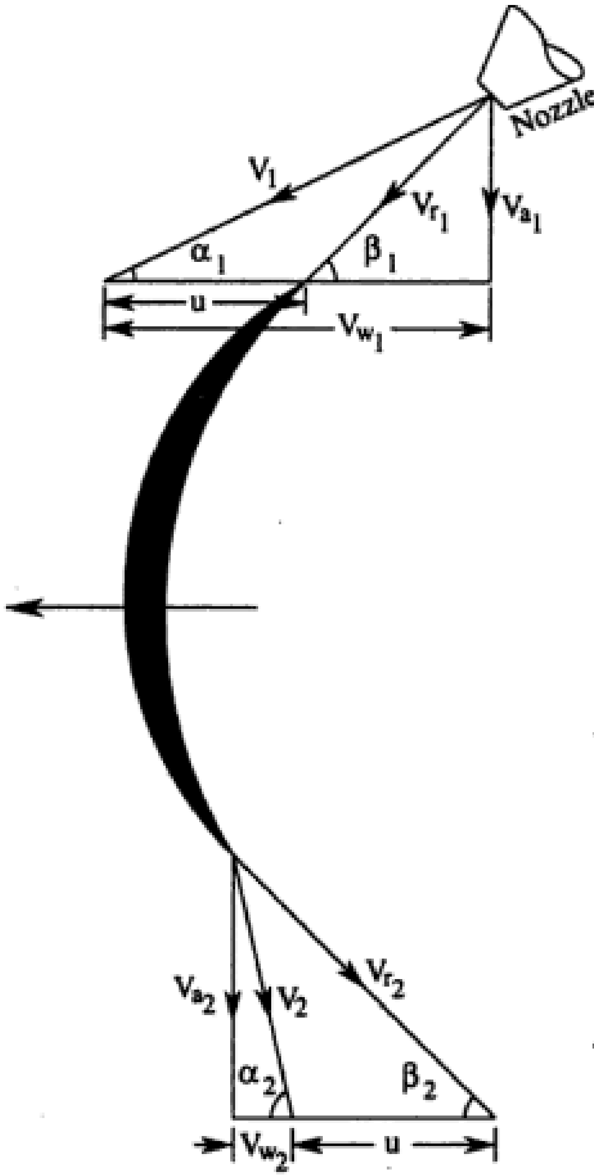


Fig. 14.3 *Velocity diagram for moving blades for an impulse turbine*

COMBINED VELOCITY DIAGRAM

The procedure for drawing the combined velocity diagram is given below:

1. Draw a horizontal line AB equal to blade velocity u to some suitable scale.
2. Draw a line AC at an angle α_1 with AB. Cut $AC = V_1$.

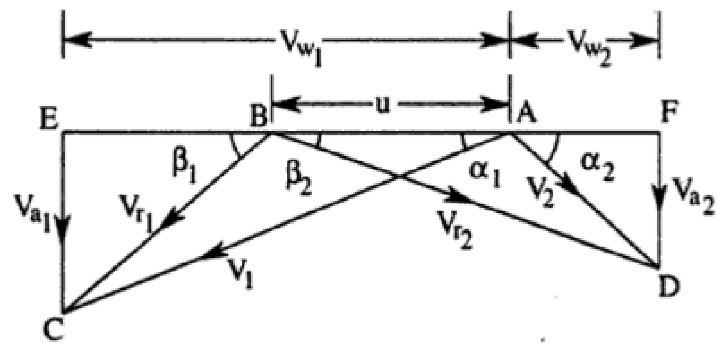


Fig. 14.4 *Combined velocity diagram*

3. Join B and C. The line BC represents the relative velocity at inlet. The blade inlet angle β_1 is measured and its value is noted down.
4. From point C draw a perpendicular CE on AB produced. CE represents axial velocity at inlet and AE represents tangential velocity at inlet.
5. From point B draw a line BD at an angle β_2 (the blade outlet angle). Cut $BD = V_{r_2} = KV_{r_1}$. Join A and D. AD represents the absolute velocity at outlet. The angle α_2 is measured and noted down.
6. From point D draw a perpendicular DF on BA. Then AF represents the tangential velocity of steam at outlet and DF represents the axial velocity outlet. This completes the velocity triangle.
   (a) Force in the tangential direction = Rate of change of momentum in the tangential direction.

   $$= \text{mass per second} \times \text{change of velocity} = m\left(V_{w_1} \pm V_{w_2}\right) \text{ N} \qquad (14.1)$$

   (b) Force in the axial direction or Axial thrust

   = Rate of change of momentum in axial direction

   $$= m\left(V_{a_1} - V_{a_2}\right) \text{ N} \qquad (14.2)$$

   (c) Work done by steam on blades $= m\left(V_{w_1} \pm V_{w_2}\right)u$ Nm/s (14.3)

   (d) Power developed by the turbine $= \dfrac{m(V_{w_1} \pm V_{w_2})u}{1000}$ kW (14.4)

(e) Blade efficiency $= \dfrac{\text{Work done on the blade/s}}{\text{Energy supplied to the blade/s}}$

$$\eta_b = \frac{m(V_{w_1} \pm V_{w_2})u}{\frac{1}{2} \times m \times V_1^{\;2}} = \frac{2u(V_{w_1} \pm V_{w_2})}{V_1^2} \tag{14.5}$$

(f) Energy lost due to blade friction $= \dfrac{1}{2} m\left(V_{r_1}^2 - V_{r_2}^2\right)$ Nm/s (14.6)

(g) Stage efficiency $= \dfrac{\text{Work done on the blade/s}}{\text{Total energy supplied per stage}}$

$$= \frac{m(V_{w_1} \pm V_{w_2})u}{m(H_1 - H_2)} = \frac{(V_{w_1} \pm V_{w_2})u}{H_d} \tag{14.7}$$

where $H_d = H_1 - H_2 =$ heat drop in the nozzle ring.

MAXIMUM WORK AND MAXIMUM DIAGRAM EFFICIENCY

From the combined velocity diagram, Fig. 14.4

$V_{w_1} = V_1 \cos\alpha_1 = V_{r_1} \cos\beta_1 + u$, $V_{w_2} = V_2 \cos\alpha_2 = V_{r_2} \cos\beta_2 - u$

$$\therefore\ V_{w_1} + V_{w_2} = Vr_1 \cos\beta_1 + V_{r_2} \cos\beta_2 = V_{r_1} \cos\beta_1 \left[1 + \frac{V_{r_2}}{V_{r_1}} . \frac{\cos\beta_2}{\cos\beta_1}\right] = Vr_1 \cos\beta_1 [1 + KC]$$

where $K = \dfrac{V_{r_2}}{V_{r_1}}$ and $c = \dfrac{\cos\beta_2}{\cos\beta_1}$

$$V_{w_1} + V_{w_2} = (V_1\cos\alpha_1 - u)(1 + KC)$$

Rate of doing work per kg of steam per second $= (V_1\cos\alpha_1 - u)(1 + KC)u$

Diagram efficiency $\eta_b = \dfrac{(V_1\cos\alpha_1 - u)(1 + KC)2u}{V_1^2}$

Let $\rho = \dfrac{u}{V_1}$ = Blade speed ratio

$$\eta_b = 2(\rho\cos\alpha_1 - \rho^2)(1 + KC) \tag{14.8}$$

It is obvious from Eq. (14.8) that the value of diagram efficiency depends upon the following factors:
(i) Nozzle angle α_1 (ii) Blade speed ratio ρ (iii) Blade angles β_1 and β_2 (iv) Blade velocity co-efficient K.
If α_1, K and C are assumed to be constant, the diagram efficiency depends upon the value of ρ. In order to determine the optimum value of ρ for maximum diagram efficiency the first differential of the equation.

$\eta_b = 2(\rho\cos\alpha_1 - \rho^2)(1 + KC)$ be equated to zero

$$\therefore\ 2(1 + KC)(\cos\alpha_1 - 2\rho) = 0$$

or $\rho = \frac{\cos\alpha_1}{2}$ since$(1 + KC)$ is not equal to zero

$$\therefore \eta_{b(\max)} = 2(1+KC)\left[\frac{\cos\alpha_1}{2}.\cos\alpha_1 - \frac{\cos^2\alpha_1}{4}\right] = (1+KC)\frac{\cos^2\alpha_1}{2} \qquad (14.9)$$

Assuming that the blades are symmetrical and friction is absent. $\beta_1 = \beta_2, \therefore C = 1$, and $K = 1$

$$\eta_{b(\max)} = \cos^2\alpha_1 \qquad (14.10)$$

Rate of doing work per kg of steam per second

$$= (V_1\cos\alpha_1 - u)(1+KC)u$$

$$\rho = \frac{u}{V_1} = \frac{\cos\alpha_1}{2} \quad \therefore V_1 = \frac{2u}{\cos\alpha_1}$$

Maximum rate of doing work per kg of steam per second

$$= \left(\frac{2u}{\cos\alpha_1}.\cos\alpha_1 - u\right)(1+1.1)u = 2u^2 \qquad (14.11)$$

METHODS OF REDUCING ROTOR SPEED

In case of simple impulse turbine, the steam is expanded from the boiler pressure to the condenser pressure in one stage only. Hence the speed of the rotor becomes very high for practical purposes. In order to make the rotor speed practicable compounding of steam turbine is done. Compounding is the method in which multiple system of rotors are keyed to a common shaft in series and the steam pressure or jet velocity is absorbed in stages as it flows over the rotor blades. The rotor speed can be reduced by the following methods of compounding steam turbine.

(a) Velocity compounding
(b) Pressure compounding
(c) Pressure-velocity compounding.

Velocity Compounding

Figure 14.5 (a) shows a section of velocity compounded turbine. It consists of a set of nozzles and a few rows of moving blades which are fixed to the shaft and rows of fixed blades which are attached to the casing. In the figure two rows of moving blades are separated by a row of fixed blades. The steam is expanded from the boiler pressure to the condenser pressure in the nozzles only. Due to the decrease in pressure the steam acquires a very high velocity. This high velocity steam first enters the first row of moving blades, where some portion of the velocity is absorbed. Then it enters the ring of fixed blades where the direction of steam is changed to suit the second ring of moving blades. There is no

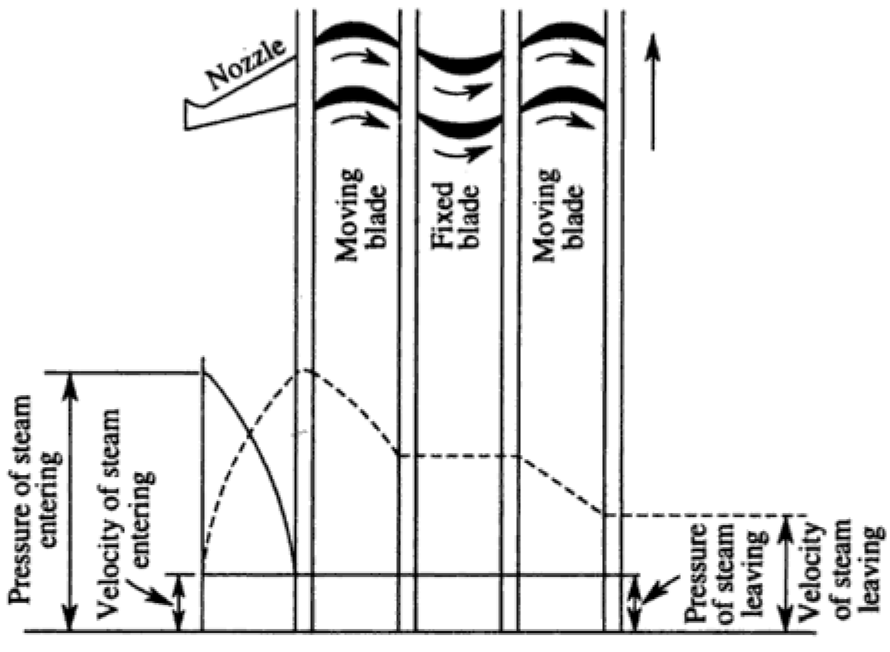


Fig. 14.5(a) *Velocity compounding*

change in the velocity as the steam passes over the fixed blades.

The steam then passes on to the second row of moving blades where velocity is further reduced. Thus a fall in velocity occurs everytime when steam passes over the row of moving blades. Steam thus leaves the turbine with a low velocity. The variation of pressure and velocity of steam as it passes over the moving and fixed blades is shown in the figure. It is clear that the pressure drop takes place only in the nozzle and there is no further drop of pressure as the steam passes over the blades. This method of velocity compounding is used in Curtis turbine.

Advantages of velocity compounded turbines

(a) The arrangement has less number of stages and hence less initial cost.

(b) The arrangement requires little space.

(c) The system is reliable and easy to operate.

(d) The fall of pressure in the nozzle is considerable, so the turbine itself need not work in high pressure surroundings and the turbine housing need not be very strong.

Disadvantages of velocity compounded turbines

(a) Due to very high steam velocity in the nozzles, friction losses are large.

(b) The efficiency is also low because the ratio of blade velocity to steam velocity is not optimum for all the wheels.

(c) Power developed in the later rows is only a fraction of power developed in the first row. Still all the stages require same space, material and cost of fabrication.

Velocity Diagram for Velocity Compounded Turbine

Figures 14.5(d) and (e) shows the velocity diagram for a two stage velocity compounded turbine.

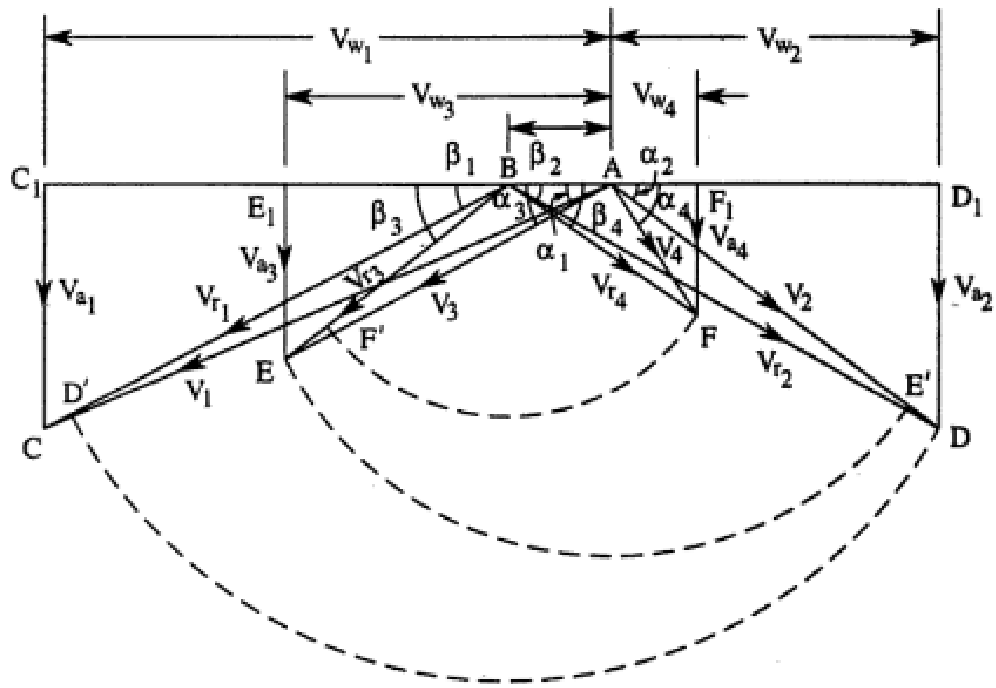


Fig. 14.5 (e) *Velocity diagram for velocity compounded two stage turbine*

Given,

u = Blade velocity in m/s

V_1 = Absolute velocity of steam at inlet to blade in m/s.

V_{r_1} = Velocity of steam relative to the 1st row of moving blades at entrance in m/s.

α_1 = Nozzle angle at entrance.

β_1 = Entrance angle to the 1st row of moving blades

K = Friction factor

V_{r_2} = Velocity of steam relative to the 1st row of moving blade at exit in m/s.

= KVr_1

β_2 = Exit angle from 1st row of moving blades.

V_2 = Absolute velocity of steam at exit from 1st row of moving blade in m/s.

α_2 = Entrance angle to the fixed blade.

V_3 = Absolute velocity of steam at exit from the fixed blades in m/s = $K V_2$

= Absolute velocity of steam entering the second row of moving blades.

α_3 = Exit angle from the fixed blades.

V_{r_3} = Velocity of steam relative to the second row of moving blades at entrance in m/s.

β_3 = Entrance angle to the 2nd row of moving blades.

Vr_4 = Velocity of steam relative to the second row of moving blades at exit in m/s.

β_4 = Exit angle from 2nd row of moving blades.

V_4 = Absolute velocity of steam at exit from 2nd row of moving blades in m/s.

α_4 = Entrance angle to the next row of fixed blades.

$\left(V_{w_1} \pm V_{w_2}\right)$ = C_1D_1 = Tangential velocity or velocity of whirl for the 1st row of moving blades in m/s.

$\left(V_{w_3} \pm V_{w_4}\right)$ = E_1F_1 = Tangential velocity or velocity of whirl for the 2nd row of moving blades in m/s.

V_{a_1} = Axial velocity of steam at entrance to the 1st row of moving blades in m/s.

m/s.

V_{a_3} = Axial velocity of steam at entrance to the 2nd row of moving blade in m/s.

V_{a_4} = Axial velocity of steam at exit to the 2nd row of moving blades in m/s.

Let AB represent the blade velocity u to any scale. At A draw $AC = V_1$ at an angle α_1 to AB. Join BC. Then, $BC = Vr_1$ and $\angle CBC_1 = \beta_1$. Mark BD' such that $BD' = KBC = Vr_2$. Draw BD such that $\angle DBA = \beta_2$. Join AD; then $AD = V_2$ and $\angle DAD_1 = \alpha_2$. From C draw perpendicular CC_1 on AB produced. Then $AC_1 = Vw_1$ and $CC_1 = Va_1$. From D draw perpendicular DD_1 on BA produced. Then $AD_1 = Vw_2$ and $DD_1 = Va_2$. At B draw $AE = V_3 = KV_2$ at an angle α_3. Join BE. Then $BE = Vr_3$.

From E draw perpendicular EE_1 on AB produced. Then $EE_1 = Va_3$ and $\angle EBE_1 = \beta_3$. Mark BF' such that $BF' = KBE = Vr_4$. Draw BF such that $\angle FBA = \beta_4$. Join AF, then $AF = V_4$. From F draw perpendicular on BA produced. Then $FF_1 = Va_4$, $AE_1 = Vw_3$ and $AF_1 = Vw_4$.

Work done from the 1st row of moving blades $= m(Vw_1 \pm Vw_2)u$ Nm/s

$$= m.C_1D_1.u \text{ Nm/s} \tag{14.12}$$

Work done from the 2nd row of moving blades $= m(Vw_3 \pm Vw_4).u$ Nm/s

$$= m.E_1F_1.u \text{ Nm/s} \tag{14.13}$$

$\therefore$ (a) Work done from the two stages $= m.C_1D_1.u + m.E_1F_1.u$

$$= m(C_1D_1 + E_1F_1).u \text{ Nm/s or J/s} \tag{14.14}$$

(b) Power developed by a two stage impulse turbine $= m.u.(C_1D_1 + E_1F_1)$ W

$$= m.u.\frac{(C_1D_1 + E_1F_1)}{1000} \text{ kW} \tag{14.15}$$

(c) Diagram or blade efficiency

$$\eta_b = \frac{u(C_1D_1 + E_1F_1)}{\frac{V_1^2}{2}} = \frac{2u(C_1D_1 + E_1F_1)}{V_1^2} \times 100\% \tag{14.16}$$

(d) Stage efficiency $= \dfrac{u(C_1D_1 + E_1F_1)}{1000\,H}$ (14.17)

where $H = H_1 - H_2$ = enthalpy drop in nozzle in kJ/kg.

(e) Total axial thrust $= m[(Va_1 - Va_2) + (Va_3 - Va_4)]$ N (14.18)

In order to have maximum efficiency of the turbine, the out going absolute velocity, V_4 of the steam should be at right angles to the blade having no tangential component i.e. $\alpha_4 = 90°$.

The velocity diagrams at the inlet and outlets for the different blades are drawn on the same side as shown in Fig. 14.5 (f) for frictionless flow and symmetrical blading under maximum efficiency condition.

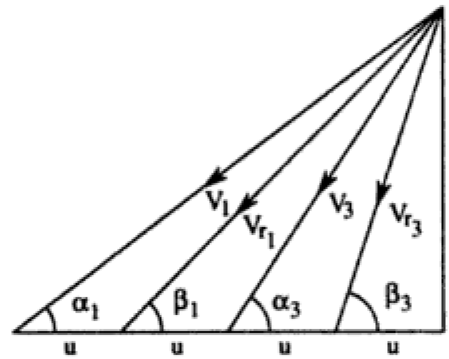


Fig. 14.5(f)

$$V_1\cos\alpha_1 = 4u \text{ or } u = \frac{V_1\cos\alpha_1}{4} = \frac{V_1\cos\alpha_1}{2 \times \text{No. of stages}}$$

From Fig. 14.5 (f) we find that, $V_2\cos\alpha_2 = 2u$

$$Vr_4\cos\beta_4 = Vr_3\cos\beta_3 = u$$

$\therefore$ Maximum work in the 1st stage $= m(V_1\cos\alpha_1 + V_2\cos\alpha_2)u = m(4u + 2u)u = m.6u^2$

Maximum work in the 2nd stage $= m(Vr_3\cos\beta_3 + Vr_4\cos\beta_4)u = m(u + u)u = m.2u^2$

$\therefore$ Total work $= 6mu^2 + 2mu^2 = 8mu^2$

Maximum blade efficiency, $\eta_{b(\max)} = \dfrac{8mu^2}{\frac{1}{2}mV_1^2}$

$$\eta_{b(\max)} = \frac{8mu^2 \times 2}{m \times \frac{16u^2}{\cos^2\alpha_1}} = \cos^2\alpha_1 \qquad (14.19)$$

Thus the velocity compounded turbine has the same maximum efficiency as the single stage turbine.

1. Steam with absolute velocity 360 m/s enters the stage of an impulse turbine provided with a single row wheel. The nozzles are inclined at 20° to the plane of the wheel. The blade rotor with diameter 95.5 cm rotates with a speed of 3000 r.p.m. Find (a) suitable inlet and outlet angle for the moving blade so that there is no axial thrust on the blade. It may be assumed that friction in blade passages is 19% of the kinetic energy corresponding to relative velocity at inlet to blades, (b) Power developed in blading for a steam flow of 1 kg/s, and (c) Kinetic energy of steam finally leaving the stage.

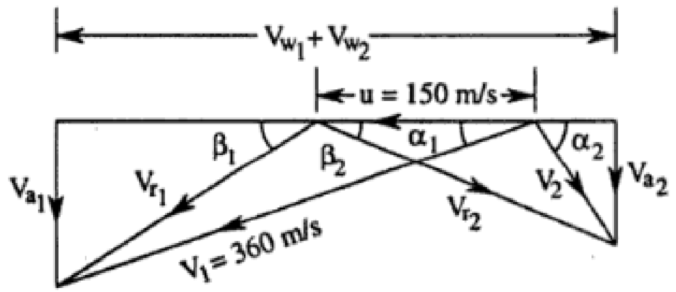


Fig. 14.10

Solution: Given, $V_1 = 360$ m/s, $\alpha_1 = 20°$, $d = 95.5$ cm, $N = 3000$ r.p.m, $V_{a_1} = V_{a_2}$

Mean blade speed, $u = \dfrac{\pi d N}{60} = \dfrac{\pi \times 0.955 \times 3000}{60} = 150$ m/s

From the given relation

$$\frac{V_{r_2}^2}{2} = (1 - 0.19)\frac{V_{r_1}^2}{2} \quad \therefore V_{r_2} = 0.9 V_{r_1}$$

Scale of velocity diagram, 1 cm = 50 m/s

From the velocity triangle

(a) Blade inlet angle, $\beta_1 = 34°$

Blade outlet angle, $\beta_2 = 38°$

$V_{w_1} + V_{w_2} = 348$ m/s

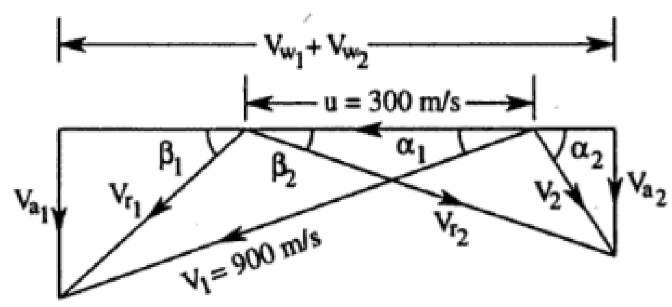
(b) Power output for a steam flow of 1 kg/s

$$= \frac{(V_{w_1} + V_{w_2})u}{1000} \text{ kW.} = \frac{348 \times 150}{1000} = 52.2 \text{ kW}$$

(c) Absolute velocity of steam at exit, $V_2 = 126$ m/s

Kinetic energy of steam finally leaving the stage $= \dfrac{V_2^2}{2} = \dfrac{(126)^2}{2} = 7938$ Nm/kg.

2. The blade speed of a single ring of an impulse turbine is 300 m/s and the nozzle angle is 20°. The isentropic heat drop is 473 kJ/kg and the nozzle efficiency is 0.85. Given that the blade velocity co-efficient is 0.7 and the blades are symmetrical, draw the vector diagrams and calculate for a mass flow of 1 kg/s:



(a) axial thrust on the blading.
(b) steam consumption per B.P. hour if the mechanical efficiency is 90 per cent.
(c) blade efficiency, stage efficiency and maximum blade efficiency.
(d) heat equivalent of the friction of blading.

Solution: $V_1 = 44.7\sqrt{(H_1 - H_2)\eta_n} = 44.7\sqrt{473 \times 0.85} = 900$ m/s

Given, $\alpha_1 = 20°$, $u = 300$ m/s, $\beta_1 = \beta_2$, $V_{r_2} = 0.7V_{r_1}$

From this data the velocity diagram is drawn and the following results are obtained.

$\beta_1 = \beta_2 = 29.5°$; $V_{w_1} + V_{w_2} = 927.5$ m/s; $V_{a_1} - V_{a_2} = 92.5$ m/s; $V_{r_1} = 630$ m/s; $V_{r_2} = 441$ m/s

(a) Axial thrust per kg $= V_{a_1} - V_{a_2} = 92.5$ N

(b) $$\text{Power} = \frac{m(V_{w_1} + V_{w_2})u}{1000} \text{ kW} = \frac{1 \times 927.5 \times 300}{1000} = 278.25 \text{ kW}$$

Brake power $= 278.25 \times 0.9 = 250.425$ kW

$$\therefore \text{Steam consumption per B.P. hour} = \frac{3600}{250.425} = 14.33 \text{ kg}$$

(c) $$\text{Blade efficiency} = \frac{2u(V_{w_1} + V_{w_2})}{V_1^2} = \frac{2 \times 300 \times 927.5}{(900)^2} = 0.688 = 68.8\%$$

Maximum blade efficiency, $\eta_{b(\max)} = \cos^2\alpha_1 = \cos^2 20 = 0.88 = 88\%$

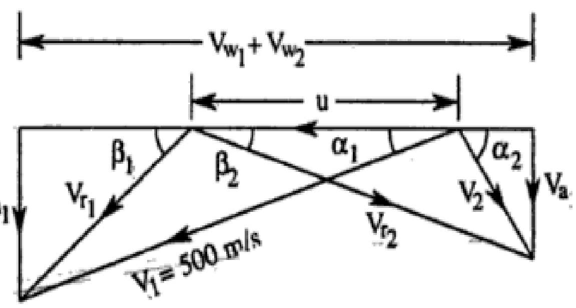
$$\text{Stage efficiency} = \frac{\text{Work done on blade}}{\text{Total energy supplied to blade}} = \frac{278.25}{473} = 0.588 = 58.8\%$$

(d) Energy lost due to blade friction

$$= \frac{Vr_1^2 - Vr_2^2}{2} = \frac{(630)^2 - (441)^2}{2} = 101209.5 \text{ J} = 101.2095 \text{ kJ.}$$

3. The data pertaining to impulse turbine is as follows:
Steam velocity = 500 m/s; Blade speed = 200 m/s; Exit angle of moving blade = 25°; Nozzle

angle = 20°. Neglecting the effect of friction when passing through the blade passages, calculate (a) inlet angle of moving blade, (b) exit velocity and direction, (c) work done per kg of steam, (d) axial thrust and power for a steam flow rate of 5 kg/s, and (e) diagram efficiency.



Solution: Given, $V_1 = 500$ m/s; $u = 200$ m/s; $\beta_2 = 25°$; $\alpha_1 = 20°$ $V_{r_1} = V_{r_2}$

The following results are obtained from the velocity diagram

(a) Inlet angle of moving blade, $\beta_1 = 33°$

(b) Exit velocity, $V_2 = 162.5$ m/s
Direction of exit velocity, $\alpha_2 = 56°$

(c) $V_{w_1} + V_{w_2} = 555$ m/s

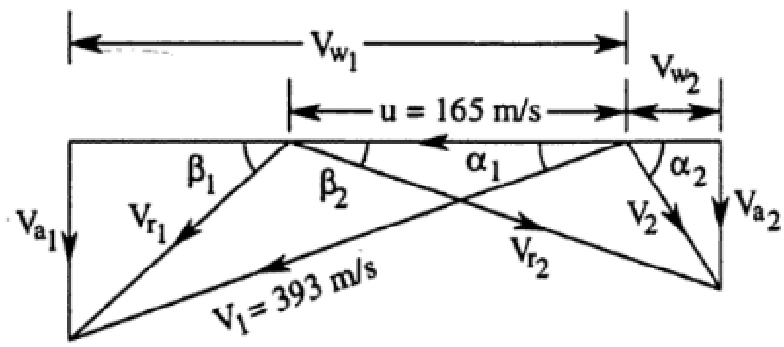
Work done per kg of steam $= m(V_{w_1} + V_{w_2})u = 1 \times 555 \times 200 = 111000$ Nm/s

(d) Axial thrust $= m(V_{a_1} - V_{a_2}) = 5 \times 45 = 225$ N

$$\text{Power} = \frac{5 \times 111000}{1000} = 555 \text{ kW.}$$

(e) $$\text{Diagram efficiency} = \frac{2u(V_{w_1} + V_{w_2})}{V_1^2} = \frac{2 \times 200 \times 555}{(500)^2} = 0.888 = 88.8\%$$

5. The mean diameter of the blades of an impulse turbine with a single row wheel is 105 cm and the speed is 3000 r.p.m. The nozzle angle is 18°, the ratio of blade speed to steam speed is 0.42 and the ratio of the relative velocity at outlet from the blades to that at inlet is 0.84. The outlet angle of the blade is to be made 3° less than the inlet angle. The steam flow is 8 kg per sec. Draw the velocity diagram for the blades and derive the (a) resultant t[illegible]n the blades, (b) tangential thrust on the blades, (c) axial thrust on the blades, (d) power developed in blades, and (e) blading efficiency



Solution: Given, $d = 105$ cm $= 1.05$ m; $N = 3000$; $\alpha_1 = 18°$

$$\frac{u}{V_1} = 0.42; \quad \frac{V_{r_2}}{V_r} = 0.84; \quad \beta_2 = \beta_1 - 3°; \quad m = 8 \text{ kg/s}$$

Blade speed, $u = \dfrac{\pi d N}{60} = \dfrac{\pi \times 1.05 \times 3000}{60} = 165$ m/s

Absolute velocity of steam at inlet, $V_1 = \dfrac{165}{0.42} = 393$ m/s

The velocity diagram is shown in Fig. 14.14.

The following results are obtained from the velocity diagram:

$\beta_1 = 30.2°$; $V_{w_1} + V_{w_2} = 389$ m/s; $V_{a_1} = 122$ m/s; $V_{a_2} = 93$ m/s

(a) Resultant thrust $= \sqrt{(3112)^2 + (232)^2} = 3120.56$ N

(b) Tangential thrust $= m(V_{w_1} + V_{w_2}) = 8 \times 389 = 3112$ N

(c) Axial thrust $= m(V_{a_1} - V_{a_2})$ N $= 8(122 - 93) = 232$ N

(d) Power developed $= \dfrac{m(V_{w_1} + V_{w_2})u}{1000}$ kW $= \dfrac{8 \times 389 \times 165}{1000} = 513.48$ kW

(e) Blading efficiency $= \dfrac{2u(V_{w_1} + V_{w_2})}{V_1^2} = \dfrac{2 \times 165 \times 389}{(393)^2} = 0.83 = 83\%.$

REACTION TURBINE

A turbine in which steam pressure decreases gradually while expanding through the moving blades as well as through the fixed blades is known as *reaction turbine*. It consists of a large number of stages, each stage consisting of set of fixed and moving blades. The heat drop takes place throughout in both fixed and moving blades. No nozzles are provided in a reaction turbine. The fixed blades act both as nozzles in which the velocity of steam is increased and direct the steam to enter the ring of moving blades. As the pressure drop takes place both in the fixed and moving blades all blades are nozzle shaped. The steam expands while flowing over the moving blades and thus gives reaction to the moving blades. Hence the turbine is known as reaction turbine. The fixed blades are attached with the casing whereas moving blades are fixed with rotor. It is also called Pearson's Reaction turbine.

The work done per kg of steam in the stage

(Per pair) $= u(Vw_1 + Vw_2)$ Nm or joules.

Work done by the steam per second per pair $= m.u(Vw_1 + Vw_2)$ N.m/s or J/s

$$\text{Power developed per pair} = \frac{m.u(Vw_1 + Vw_2)}{1000} \text{ kW} \tag{14.20}$$

where, m = mass of steam flowing over blades in kg/s.

$$\text{Efficiency} = \frac{\text{Work done per pair per kg of steam}}{\text{Enthalpy drop per pair}} = \frac{u(Vw_1 + Vw_2)}{1000H} \tag{14.21}$$

where, H = enthalpy drop per pair in kJ/kg.

Degree of Reaction (R)

The degree of reaction is defined as the ratio of isentropic heat drop in the moving blades to isentropic heat drop in the entire stage of reaction turbine.
The degree of Reaction R is given by,

$$R = \frac{\text{Enthalpy drop in the moving blade}}{\text{Enthalpy drop in the stage}} = \frac{dH_2}{dH_1 + dH_2} \tag{14.22}$$

where dH_1 = enthalpy drop in the fixed blade per kg of steam $= \dfrac{V_1^2 - V_2^2}{2}$ kJ/kg.

dH_2 = enthalpy drop in the moving blade per kg of steam $= \dfrac{V_{r_2}^2 - V_{r_1}^2}{2}$ kJ/kg $= H_2 - H_3$.

Also, $dH_1 + dH_2$ = enthalpy drop in the stage per kg of steam

$= H_1 - H_3$ = work done by the steam in the stage.

$$dH_1 + dH_2 = u(Vw_1 + Vw_2) \quad \therefore R = \frac{Vr_2^2 - Vr_1^2}{2u(Vw_1 + Vw_2)} \tag{14.23}$$

In Parsons Reaction turbine, the degree of reaction is 50%, then $\alpha_1 = \beta_2, \alpha_2 = \beta_1$. Which means that the moving blade and fixed blade have the same shape. When degree of reaction $R = 0$, we have the simple impulse turbine. When degree of reaction $R = 1$, we have the pure reaction turbine.
For symmetrical triangles, $V_{r_2} = V_1, V_{r_1} = V_2$.

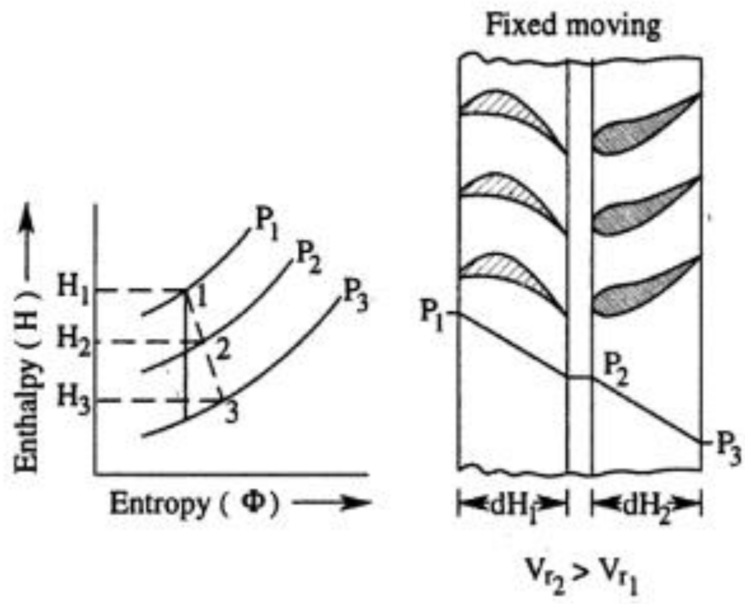


Fig. 14.6 *Reaction turbine stage isentropic expansion*

Efficiency of Reaction Turbine

The condition for maximum efficiency is calculated considering the following assumptions.

(a) The degree of reaction is 50%.

(b) The fixed and moving blades are symmetrical.

The kinetic energy supplied to the fixed blade per kg of steam $= \frac{V_1^2}{2}$

The kinetic energy supplied to the moving blade per kg of steam $= \dfrac{V_{r_2}^2 - V_{r_1}^2}{2}$

$$\therefore \text{ Total energy supplied} = \frac{V_1^2}{2} + \frac{V_{r_2}^2 - V_{r_1}^2}{2} \tag{14.24}$$

As $V_{r_2} = V_1$

$$\text{Total energy supplied} \qquad = V_1^2 - \frac{V_{r_1}^2}{2}$$

From velocity triangle, $\quad V_{r_1}^2 = V_1^2 + u^2 - 2.u.V_1\cos\alpha_1$

$$\therefore \text{ Total energy supplied} = V_1^2 - \frac{V_1^2 + u^2 - 2.u.V_1\cos\alpha_1}{2} \tag{14.25}$$

The work done per kg of steam is given by, work done $= u(Vw_1 + Vw_2)$

$$= u(V_1\cos\alpha_1 + Vr_2\cos\beta_2 - u) = u(2V_1\cos\alpha_1 - u) \quad \text{As } \alpha_1 = \beta_2 \text{ and } V_1 = Vr_2$$

$$\text{Diagram efficiency} = \frac{\text{Work done}}{\text{Total energy supplied}}$$

$$= \frac{u(2V_1\cos\alpha_1 - u)}{V_1^2 - \frac{V_1^2 + u^2 - 2u.V_1\cos\alpha_1}{2}} = \frac{2u(2V_1\cos\alpha_1 - u)}{V_1^2 - u^2 + 2.u.V_1\cos\alpha_1}$$

$$= \frac{2u\,V_1\left(2\cos\alpha_1 - \frac{u}{V_1}\right)}{V_1^2\left(1 - \frac{u^2}{V_1^2} + 2.\frac{u}{V_1}\cos\alpha_1\right)} = \frac{2.\frac{u}{V_1}\left(2\cos\alpha_1 - \frac{u}{V_1}\right)}{\left(1 - \frac{u^2}{V_1^2} + 2.\frac{u}{V_1}.\cos\alpha_1\right)}$$

$$= \frac{2\rho(2\cos\alpha_1 - \rho)}{(1 - \rho_2 + 2\rho\cos\alpha_1)} \qquad (14.26)$$

where $\rho = \frac{u}{V_1}$, the blade speed ratio.

The efficiency is maximum when $1 - \rho^2 + 2\rho\cos\alpha_1$ is minimum or when its differential with respect to ρ is zero.

$\therefore$ For maximum efficiency $\frac{d}{d\rho}(1 - \rho^2 + 2\rho\cos\alpha_1) = 0$

$$-2\rho + 2\cos\alpha_1 = 0 \text{ or, } \rho = \cos\alpha_1 \qquad (14.27)$$

Putting the value of ρ in Eq. 14.26.

In a Parson's turbine of 50% degree of reaction running at 25 r.p.s. the available enthalpy drop for an expansion is 62.8 kJ/kg. If the mean diameter of the rotor is 1 m, find the number of rows of moving blades required. The blade outlet angle is 20° and speed ratio is 0.7. Assume stage efficiency as 80%.

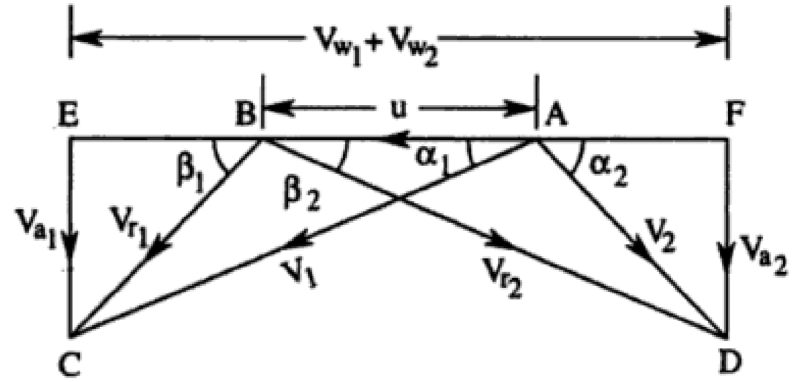


Fig. 14.31

Solution: Given, $\beta_2 = \alpha_1 = 20°$,

Blade speed, $u = \dfrac{\pi D_m N}{60} = \dfrac{\pi \times 1 \times 25 \times 60}{60} = 78.5$ m/s

Speed ratio $= \dfrac{u}{V_1} = 0.7 \quad \therefore \quad V_1 = \dfrac{78.5}{0.7} = 112$ m/s

$$AF = V_{r_2}\cos\beta_2 - u = V_1\cos\beta_2 - u = 112\cos 20° - 78.5 = 27$$

$$V_{w_1} + V_{w_2} = V_1\cos\alpha_1 + AF = 112 \times \cos 20° + 27 = 132.5 \text{ m/s}$$

Work done per stage $= \dfrac{m(V_{w_1} + V_{w_2})u}{1000} = \dfrac{1 \times 132.5 \times 78.5}{1000} = 10.4675$ kJ/kg.

$$\text{Stage efficiency} = \frac{\text{Work done per stage}}{\text{Enthalpy drop per stage}}$$

$\therefore$ Enthalpy drop per stage $= \dfrac{10.4675}{0.8} = 13.084$ kJ/kg.

$\therefore$ Number of rows required $= \dfrac{\text{Total enthalpy drop}}{\text{Enthalpy drop per stage}} = \dfrac{62.8}{13.084} = 5$ stages.

At a stage of a reaction turbine, the mean rotor diameter is 140 cm. The speed ratio is 0.7. Find the inlet angle of the blade if the outlet angle of the blade is 20°. The speed of the turbine is 3000 r.p.m. Find the diagram efficiency.

If the rotor is designed to run at the best theoretical speed and the exit angle remains the same, find the percentage increase in diagram efficiency and rotor speed.

Solution: $u = \dfrac{\pi DN}{60} = \dfrac{\pi \times 1.4 \times 3000}{60} = 220$ m/s

Speed ratio, $\rho = 0.7 = \dfrac{u}{V_1}$ $\therefore$ $V_1 = \dfrac{220}{0.7} = 314$ m/s

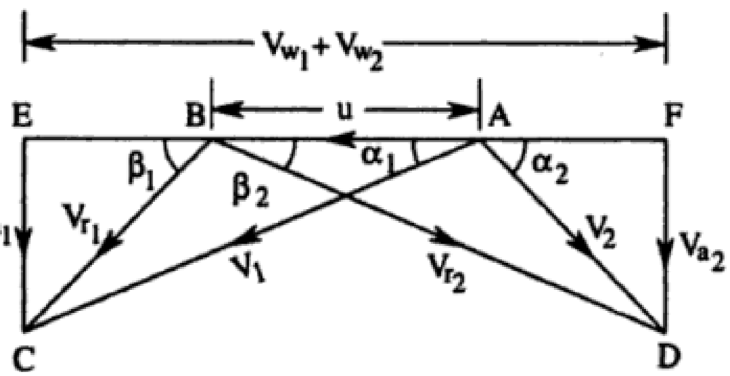


Fig. 14.32

$\alpha_1 = \beta_2 = 20°$

From ΔABC

$$V_{r_1}^2 = V_1^2 + u^2 - 2V_1 u \cos\alpha_1 = (314)^2 + (220)^2 - 2 \times 314 \times 220 \times \cos 20° = 17168.068$$

$\therefore V_{r_1} = 131.02$ m/s.

Work done per kg of steam $= m(V_{w_1} + V_{w_2})u$

$= m(2V_1 \cos\alpha_1 - u)u = 1(2 \times 314\ \cos 20° - 220)220$ Nm $= 81521.1$ Nm.

$$\text{Energy supplied} = \frac{V_1^2 + V_{r_2}^2 - V_{r_1}^2}{2} = \frac{2V_1^2 - V_{r_1}^2}{2} \quad [\therefore V_1 = Vr_2]$$

$$= \frac{2 \times (314)^2 - (131.02)^2}{2} = 90012.88 \text{ Nm.}$$

$$\therefore \text{Diagram efficiency} = \frac{\text{Work done}}{\text{Energy supplied}} = \frac{81521.1}{90012.88} = 0.9056 = 90.56\%$$

For maximum efficiency, $u = V_1 \cos\alpha_1 = 314 \ \cos 20° = 295$ m/s

For this blade speed of 295 m/s, $V_{r_1}^2 = (314)^2 + (295)^2 - 2 \times 314 \times 295 \ \cos 20°$

$\therefore Vr_1 = 110$ m/s.

$$\therefore \text{Diagram efficiency} = \frac{\text{Work done}}{\text{Energy supplied}}$$

$$= \frac{2u(2V_1 \cos\alpha_1 - u)}{\left(V_1^2 + V_{r_2}^2 - V_{r_1}^2\right)} = \frac{2 \times 295(2 \times 314 \cos 20° - 295)}{(314)^2 + (314)^2 - (110)^2} = 0.938 = 93.8\%$$

$$\therefore \text{Percentage increase in diagram efficiency} = \frac{0.938 - 0.905}{0.905} = 0.0365 = 3.65\%$$

$\therefore$ The best rotor speed

$$N = \frac{60u}{\pi D} = \frac{60 \times 295}{\pi \times 1.4} = 4025 \text{ r.p.m.}$$

In a power plant the steam is supplied at a pressure of 30 bar and temperature 300°C to the high pressure side of the turbine where it is expanded to 5 bar. The steam is then removed and reheated to 300°C at constant pressure. It is then expanded to the low pressure side of the turbine to 0.05 bar. Find the efficiency of the cycle with and without reheating.

Solution: Given, $P_1 = 30$ bar,

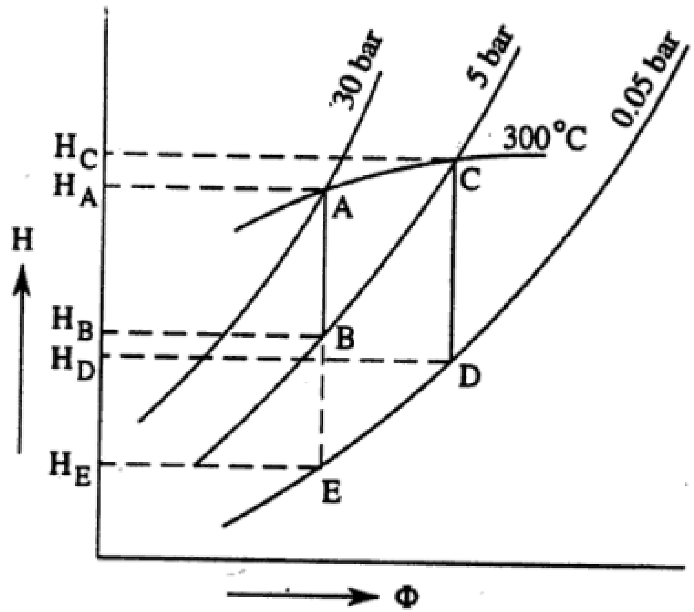


Fig. 14.33

$T_1 = 300°C$, $P_2 = 5$ bar, $P_3 = 0.05$ bar

Temperature of reheated steam = 300°C

From Mollier diagram

$H_A = 2990$ kJ/kg, $H_b = 2625$ kJ/kg, $H_C = 3075$ kJ/kg,

$H_D = 2595$ kJ/kg, $H_E = 2280$ kJ/kg.

From steam table $h_D = h_E = 340.6$ kJ/kg.

With reheating

$$\eta = \frac{(H_A - H_B) + (H_C - H_D)}{H_A + H_C - H_B - h_D} = \frac{(2990 - 2625) + (3075 - 2595)}{2990 + 3075 - 2625 - 340.6} = 0.273 = 27.3\%.$$

Steam Condensers

1. INTRODUCTION

A **steam condenser** *is a device or an appliance in which steam condenses and heat released by steam is absorbed by water*. It serves the following purposes :

1. It *maintains a very low back pressure* on the exhaust side of the piston of the steam engine or turbine. Consequently, the steam expands to a greater extent which results in an *increase in available heat energy* for converting into mechanical work. The shaded area in Fig. 1. (*i.e.*, area 44′5′5) shows the increase in work obtained by fitting a condenser to a non-condensing engine. *The thermal efficiency of a condensing unit therefore is higher than that of non-condensing unit for the same available steam.*

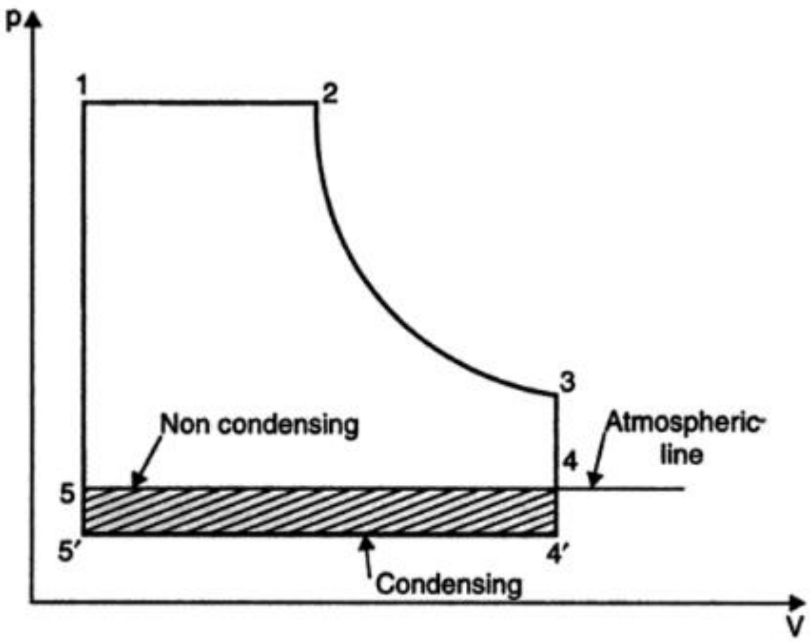


Fig. 1

2. It supplies to the boiler pure and hot feed water as the condensed steam which is discharged from the condenser and collected in a hot well, can be used as feed water for the boiler.

2. VACUUM

Vacuum is *sub-atmospheric pressure*. It is measured as the *pressure depression below atmospheric*. The condensation of steam in a closed vessel produces a partial vacuum by reason of the great reduction in the volume of the low pressure steam or vapour. The back pressure in steam engine or steam turbine can lowered from 1.013 to 0.2 bar abs. or even less. Since the steam engines are intermittent flow machines and as such cannot take the advantage of a very low vacuum, therefore, for most steam engines the exhaust pressure is about 0.2 to 0.28 bar abs. On the other hand, in steam turbines, which are continuous flow machines, the back pressure may be about 0.025 bar abs.

3. ORGANS OF A STEAM CONDENSING PLANT

A steam condensing plant mainly consists of the following *organs/elements* :

1. Condenser (To condense the steam).
2. Supply of cooling (or injection) water.
3. Wet air pump (To remove the condensed steam, the air and uncondensed water vapour and gases from the condenser ; separate pumps may be used to deal with air and condensate).
4. Hot well (where the condensate can be discharged and from which the boiler feed water is taken).
5. Arrangement for recooling the cooling water in case surface condenser is employed.

4. CLASSIFICATION OF CONDENSERS

Mainly, condensers are of two types : (1) Jet condensers, (2) Surface condenser.

In ***jet condensers***, the *exhaust steam and water come in direct contact with each other and temperature of the condensate is the same as that of cooling water leaving the condenser*. The cooling water is usually sprayed into the exhaust steam to cause, rapid condensation.

In ***surface condensers***, *the exhaust steam and water do not come into direct contact*. The steam passes over the outer surface of tubes through which a supply of cooling water is maintained. There may be single-pass or double-pass. In single-pass condensers, the water flows in one direction only through all the tubes, while in two-pass condenser the water flows in one direction through the tubes and returns through the remainder.

A jet condenser is simpler and cheaper than a surface condenser. It should be installed when the cooling water is cheaply and easily made suitable for boiler feed or when a cheap source of boiler and feed water is available. A surface condenser is most commonly used because the condensate obtained is not thrown as a waste but returned to the boiler.

4.1. Jet Condensers

These condensers may be classified as :

(*a*) Parallel flow type

(*b*) Counter flow type

(*c*) Ejector type.

Parallel flow and counter flow condensers are further sub-divided into two types : (*i*) Low level type (*ii*) High level type.

In *parallel-flow type* of condenser, both the exhaust steam and cooling water find their entry at the top of the condenser and then flow downwards and condensate and water are finally collected at the bottom.

In *counter-flow type,* the steam and cooling water enter the condenser from opposite directions. Generally, the exhaust steam travels in upward direction and meet the cooling water which flows downwards.

Low level jet condenser (Parallel-flow)

In the Fig. 2 is shown a line sketch of a low level parallel flow condenser. The exhaust steam is entering the condenser from the top and cold water is being sprayed on its way. The baffle plate provided in it ensures the proper mixing of the steam and cooling water. An extraction pump at the bottom discharges the condensate to the hot well from where it may be fed to the boiler if the cooling water being used is free from impurities. A separate dry pump may be incorporated to maintain proper vacuum.

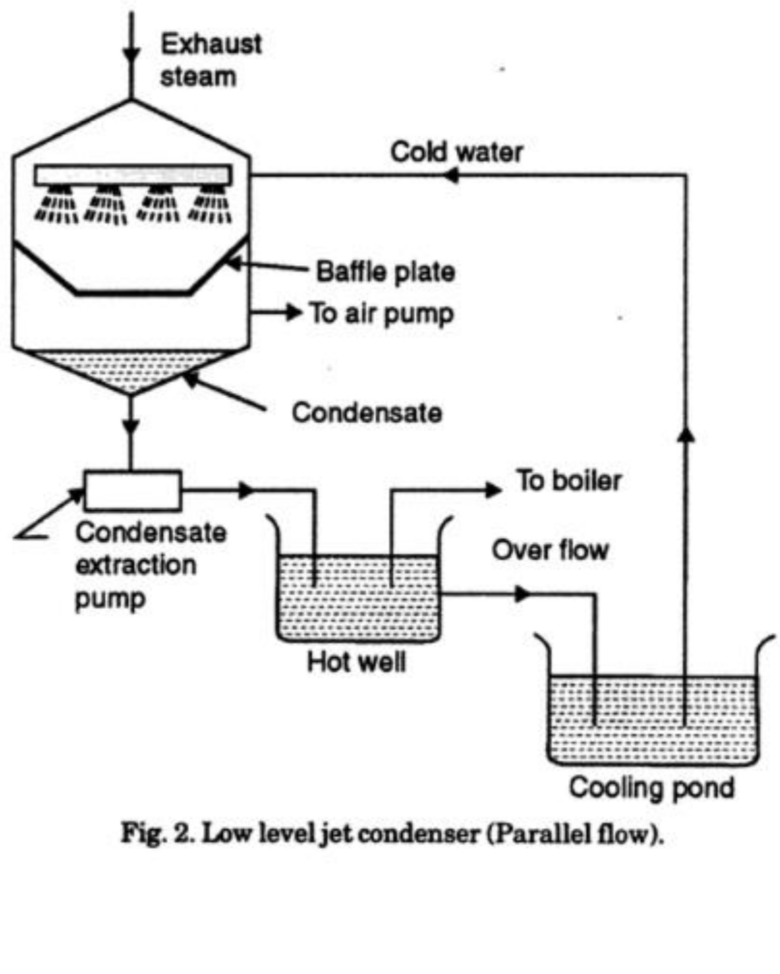


Fig. 2. Low level jet condenser (Parallel flow).

Low level jet condenser (Counter-flow)

Refer Fig. 3 (on next page). *L, M* and *N* are the perforated trays which break up water into jets. The steam moving upwards comes in contact with water and gets condensed. The condensate and water mixture is sent to the hot well by means of an extraction pump and the air is removed by an air suction pump provided at the top of the condenser.

High level jet condenser (Counter-flow type)

In Fig. 4 is shown a high level counter-flow jet condenser. It is also called ***barometric condenser***. In this case the shell is placed at a height about 10.363 metres above hot well and thus the necessity of providing an extraction pump can be obviated. However provision of own injection pump has to be made if water under pressure is not available.

Ejector condenser

Fig. 5 shows the schematic sketch of an ejector condenser. Here the exhaust steam and cooling water mix in hollow truncated cones. The cold water having a head of about 6 metres flow down through the number of cones and as it moves its velocity increases and drop in pressure results. Due to this decreased pressure exhaust steam along with associated air is drawn through

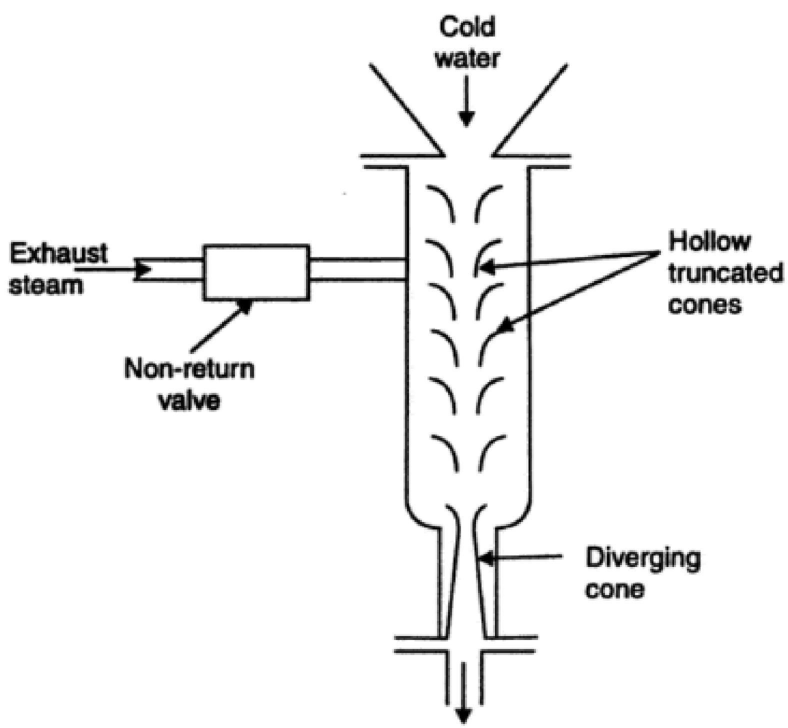


Fig. 5. Ejector condenser.

the truncated cones and finally lead to diverging cone. In the diverging cone, a portion of kinetic energy gets converted into pressure energy which is more than the atmospheric so that condensate consisting of condensed steam, cooling water and air is discharged into the hot well. The exhaust steam inlet is provided with a non-return valve which does not allow the water from hot well to rush back to the engine in case a failure of cooling water supply to condenser.

4.2. Surface Condensers

Most condensers are generally classified on the direction of flow of condensate, the arrangement of the tubing and the position of the condensate extraction pump. The following is the main classification of surface condensers :

(*i*) Down-flow type
(*ii*) Central-flow type
(*iii*) Inverted-flow type
(*iv*) Regenerative type
(*v*) Evaporative type.

(*i*) Down-flow type

In Fig. 6. is shown a down flow type of surface condenser. It consists of a shell which is generally of cylindrical shape ; though other types are also used. It has cover plates at the ends and furnished with number of parallel brass tubes. A baffle plate partitions the water box into two

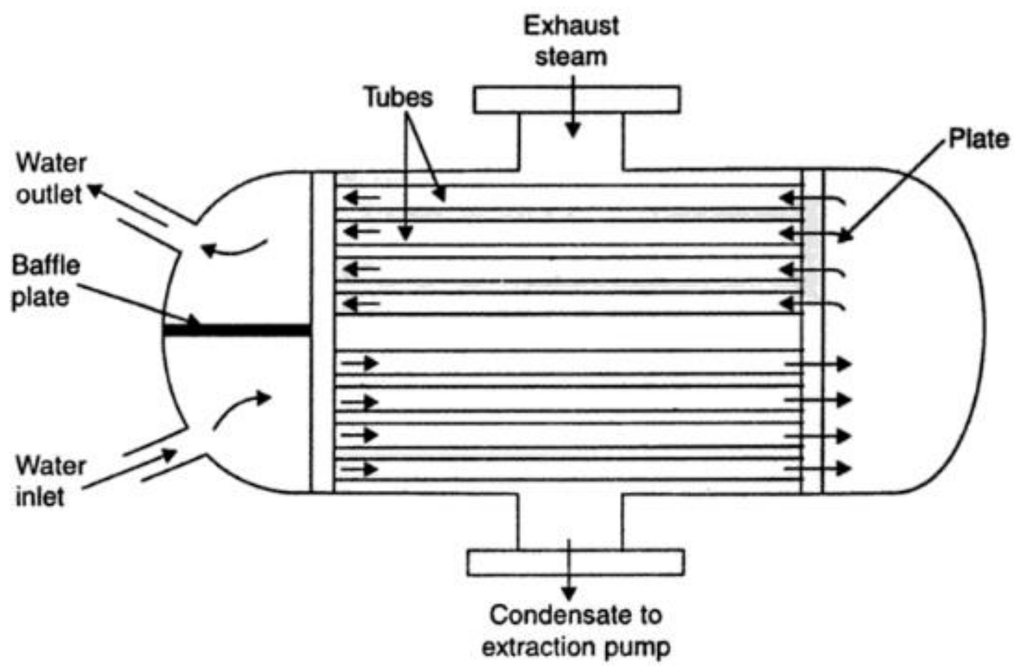


Fig. 6. Down flow type.

sections. The cooling water enters the shell at the lower half section and after travelling through the upper half section comes out through the outlet. The exhaust steam entering shell from the top flows down over the tubes and gets condensed and is finally removed by an extraction pump. Due to the fact that steam flows in a direction right angle to the direction of flow of water, it is also called *cross-surface condenser*.

(*ii*) **Central-flow type**

Refer Fig. 7 (on next page). In this type of condenser, the suction pipe of the air extraction pump is located in the centre of the tubes which results in radial flow of the steam. The better contact between the outer surface of the tubes and steam is ensured, due to large passages the pressure drop of steam is reduced.

(*iii*) **Inverted-flow type**

This type of condenser has the air suction at the top, the steam after entering at the bottom rises up and then again flows down to the bottom of the condenser, by following a path near the outer surface of the condenser. The condensate extraction pump is at the bottom.

(*iv*) **Regenerative type**

This type is applied to condensers adopting a regenerative method of heating of the condensate. After leaving the tube nest, the condensate is passed through the entering exhaust steam from the steam engine or turbine thus raising the temperature of the condensate, for use as feed water for the boiler.

(*v*) **Evaporative type**

Fig. 8 shows the schematic sketch of an *evaporative condenser*. The underlying principle of this condenser is that when a limited quantity of water is available, its quantity needed to condense the steam can be reduced by causing the circulating water to evaporate under a small partial pressure.

The exhaust steam enters at the top through gilled pipes. The water pump sprays water on the pipes and descending water condenses the steam. The water which is not evaporated falls into the open tank (cooling pond) under the condenser from which it can be drawn by circulating water pump and used over again. The evaporative condenser is placed in open air and finds its application in small size plants.

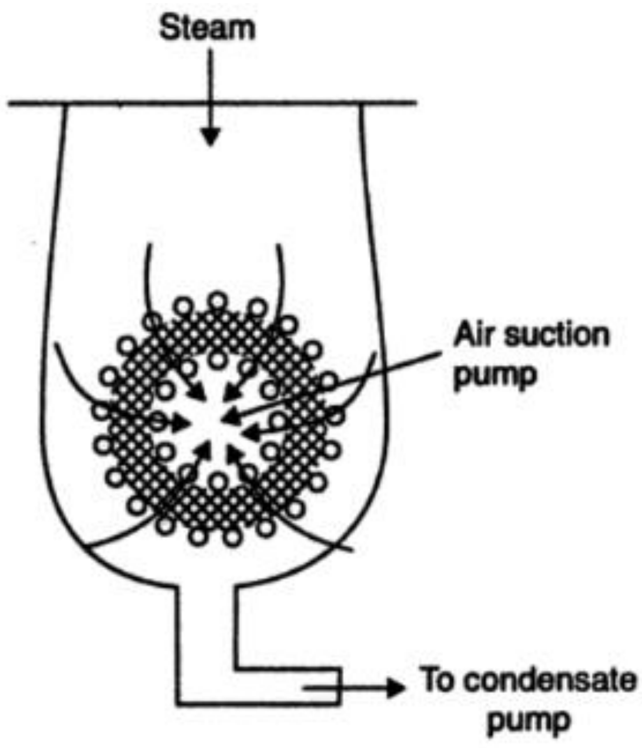


Fig. 7. Central flow type.

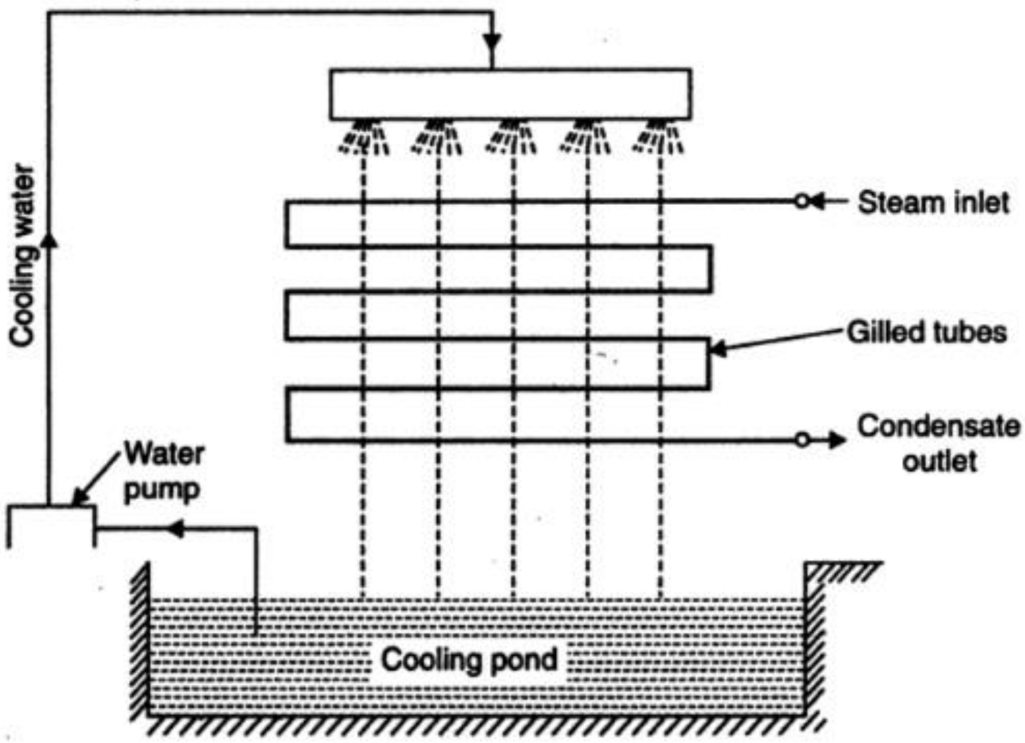


Fig. 8. Evaporative condenser.

4.4. Comparison between Jet and Surface Condensers

	Jet Condenser	Surface Condenser
1.	Cooling water and steam are mixed up.	Cooling water and steam are not mixed up.
2.	Low manufacturing cost.	High manufacturing cost.
3.	Lower up keep.	Higher upkeep.
4.	Requires small floor space.	Requires large floor space.
5.	The condensate cannot be used as feed water in the boilers unless the cooling water is free from impurities.	Condensate can be reused as feed water as it does not mix with the cooling water.
6.	More power is required for air pump.	Less power is needed for air pump.
7.	Less power is required for water pumping.	More power is required for water pumping.
8.	It requires less quantity of cooling water.	It requires large quantity of cooling water.
9.	The condensing plant is simple.	The condensing plant is complicated.
10.	Less suitable for high capacity plants due to low vacuum efficiency.	More suitable for high capacity plants as vacuum efficiency is high.

IV - UNIT

THE SIMPLE GAS TURBINE CYCLE

The schematic details of a simple gas turbine are shown in Fig. 5.1. Figure 5.2 shows the various processes on a p-V diagram whereas Fig. 5.3 gives the details on a T-s diagram. Figures 5.4 and 5.5 show the performance curves of the cycle.

From the thermodynamic analysis the relevant steady flow energy equation has been shown to be (refer Eq. 2.54)

$$w_s = \Delta h = h_2 - h_1 \tag{5.1}$$

where w_s is the work transfer per unit mass flow, and h stands for enthalpies.

Applying Eq. 5.1 to each component and assuming unit mass flow of the working fluid, we can write the work input to the compressor (process $1 \rightarrow 2$) as

Compressor work $[W_C]$

$$W_{12} = (h_2 - h_1) = C_p(T_2 - T_1) \tag{5.2}$$

Heat addition $[Q]$

$$Q_{23} = (h_3 - h_2) = C_p(T_3 - T_2) \tag{5.3}$$

Turbine work $[W_T]$

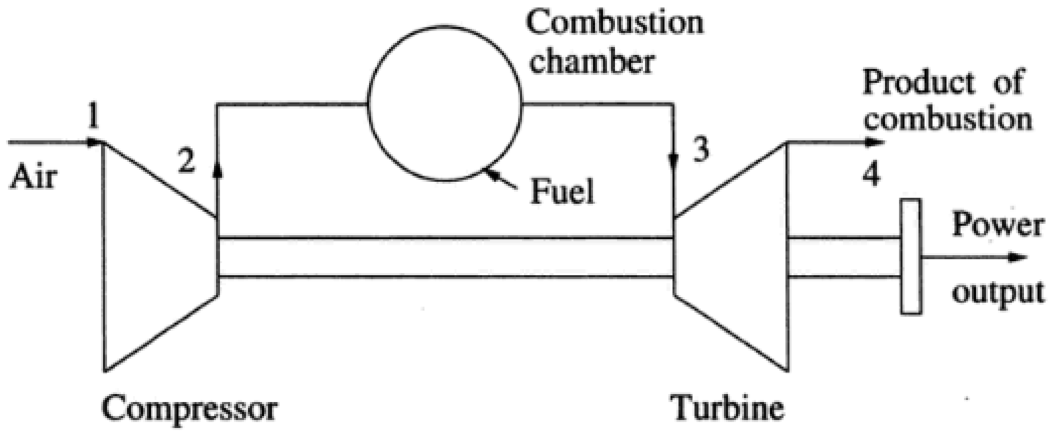


Fig. 5.1 Schematic arrangement of a simple gas turbine

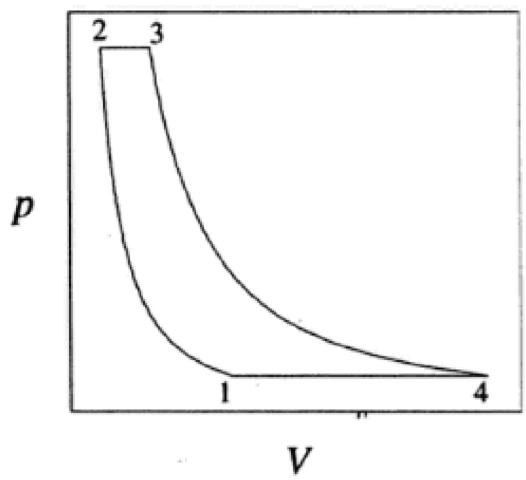


Fig. 5.2 p-V diagram

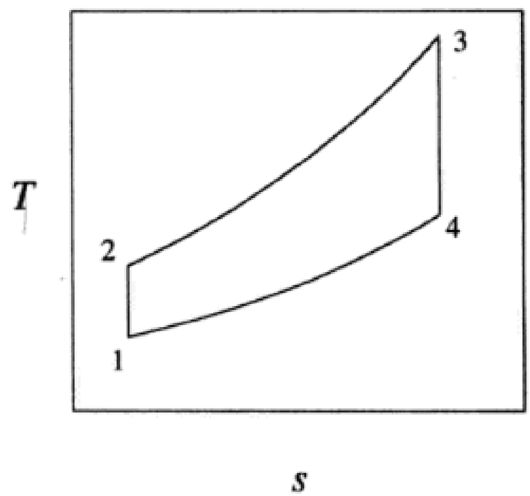


Fig. 5.3 T-s diagram

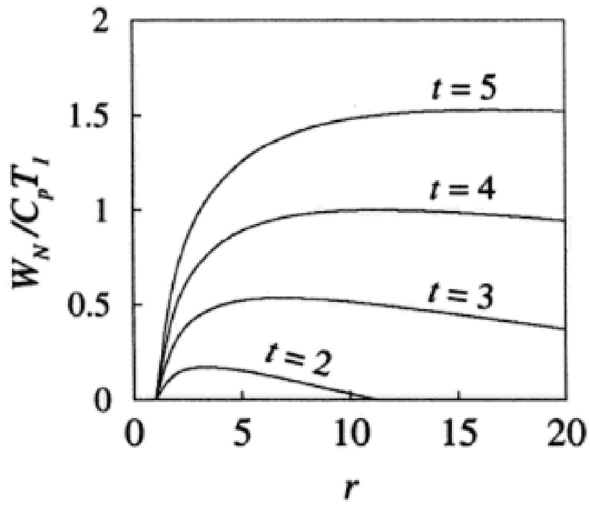


Fig. 5.4 r vs $\frac{W_N}{C_p \ T_1}$

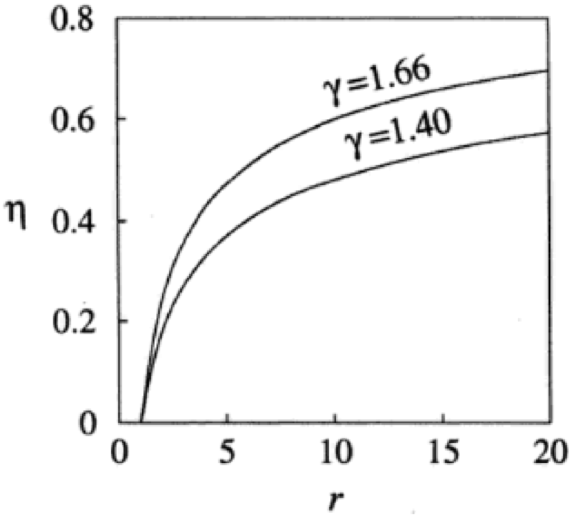


Fig. 5.5 r vs efficiency

$$W_{34} = (h_3 - h_4) = C_p(T_3 - T_4) \tag{5.4}$$

Net work output $[W_N]$

$$= W_T - W_C$$

$$= C_p(T_3 - T_4) - C_p(T_2 - T_1) \tag{5.5}$$

$$= C_pT_1\left(\frac{T_3}{T_1} - \frac{T_4}{T_1} - \frac{T_2}{T_1} + 1\right) \tag{5.6}$$

Let

$$\frac{T_3}{T_1} = t \quad \text{and} \quad \frac{p_2}{p_1} = r \tag{5.7}$$

then

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

Let $r^{\frac{\gamma-1}{\gamma}} = c$. From Eq. 5.6

$$\begin{aligned}\frac{W_N}{C_pT_1} &= \frac{T_3}{T_1} - \frac{T_4}{T_3}\frac{T_3}{T_1} - \frac{T_2}{T_1} + 1 \\ &= t - \frac{t}{c} - c + 1 \qquad (5.8)\end{aligned}$$

$$\frac{W_N}{C_pT_1} = t\left(1 - \frac{1}{c}\right) - (c-1) \qquad (5.9)$$

Equation 5.9 shows that the specific work output $\frac{W_N}{C_p\ T_1}$, upon which the size of the plant depends is a function of not only the pressure ratio, (r), but also of maximum cycle temperature T_3. Now,

$$\eta = \frac{\text{Net work output}}{\text{Heat input}} = \frac{W_N}{Q} \qquad (5.10)$$

$$= \frac{C_pT_1\left[t(1-\frac{1}{c}) - (c-1)\right]}{C_p(T_3 - T_2)} \qquad (5.11)$$

$$= \frac{C_pT_1\left[(t-c) - \left(\frac{t-c}{c}\right)\right]}{C_pT_1\left[\frac{T_3}{T_1} - \frac{T_2}{T_1}\right]} = \frac{(t-c) - \frac{t-c}{c}}{t-c}$$

$$\eta = 1 - \frac{1}{c} = 1 - \frac{1}{r^{\frac{\gamma-1}{\gamma}}} \qquad (5.12)$$

The efficiency of a simple gas turbine thus depends only on the pressure ratio and the nature of the gas, (γ).

Figure 5.4 shows the plot of work output in non-dimensional form $\frac{W_N}{C_pT_1}$ as a function of r and t. It may be noted that the value of t depends on the maximum cycle temperature, known as the *metallurgical limit*. The highly stressed parts of the turbine should withstand this temperature during the required working life. For an industrial plant it might be between *3.5 and 4.0*, whereas a value of *5.0 to 5.5* is permissible for an aircraft engine with cooled turbine blades.

A glance at the specific output curves (Fig. 5.4) show that a constant t curve exhibits a maximum at a certain pressure ratio; $W = 0$ at $r = 1$ and also at the value of the pressure ratio, $r = t^{\gamma/(\gamma-1)}$ (i.e., $c = t$) for which the compression and expansion processes coincide (refer Eq. 5.9).

For any given value of t, there must be an optimum pressure ratio to give a maximum specific output. This can be found by differentiating the Eq. 5.9 with respect to c and equating it to zero; the result is,

$$r_{opt}^{(\gamma-1)/\gamma} = \sqrt{t} \tag{5.13}$$

Squaring both sides

$$t = \left(r^{\frac{\gamma-1}{\gamma}}\right)^2$$

since,

$$r^{(\gamma-1)/\gamma} = \frac{T_2}{T_1} = \frac{T_3}{T_4} \qquad (5.14)$$

it can be written as

$$t = \frac{T_2}{T_1}\frac{T_3}{T_4} = \frac{T_2}{T_4}\, t \qquad (5.15)$$

If Eq. 5.15 is to be true then it follows that T_2 must be equal to T_4, i.e., $T_2 = T_4$.

This means that the specific work output is maximum when the pressure ratio is such that the compressor and turbine outlet temperatures are equal. For all values of r between 1 and $t^{\gamma/2(\gamma-1)}$, T_4 will be greater than T_2 and a heat exchanger can be incorporated to utilize the energy in the exhaust gas to heat the air coming out of the compressor and thereby, the efficiency improvement can be achieved.

Figure 5.5 shows the relation between η and r when the working fluid is air ($\gamma = 1.4$), or a mono-atomic gas such as argon ($\gamma = 1.66$) is used. It can be seen that the efficiency increases with pressure ratio but the rate of increase reduces with the increase in pressure ratio.

5.3 THE HEAT EXCHANGE CYCLE

A schematic arrangement of a simple gas turbine cycle incorporating exhaust heat exchanger is shown in Fig. 5.6. The corresponding p-V and T-s diagrams are shown in Fig. 5.7 and Fig. 5.8 respectively. Figures 5.9 and 5.10 show the performance curves of the cycle.

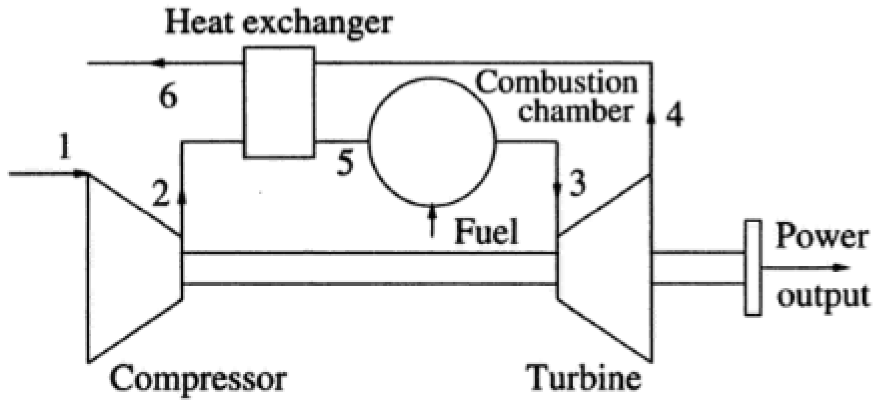
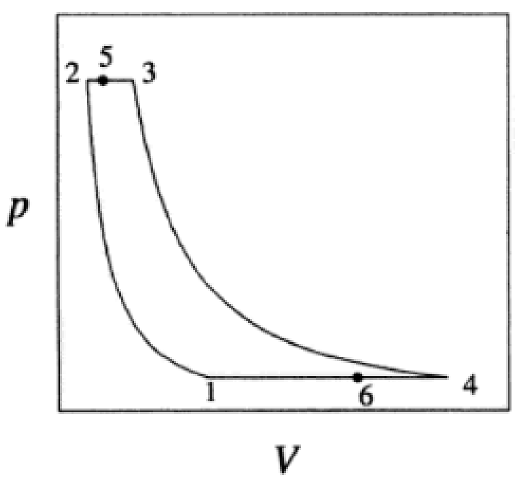


Fig. 5.6 Schematic arrangement of a heat exchange cycle



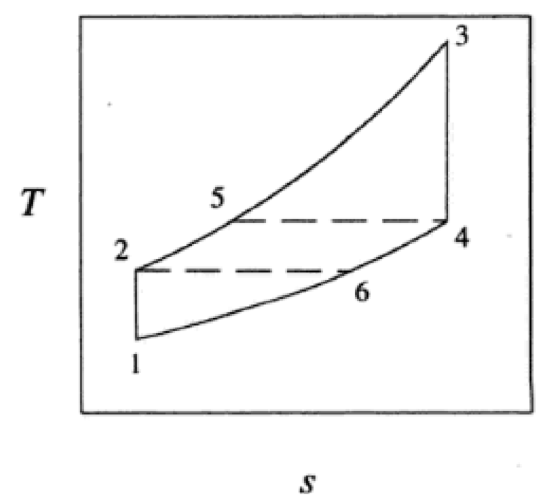


Fig. 5.7 p-V diagram

Fig. 5.8 T-s diagram

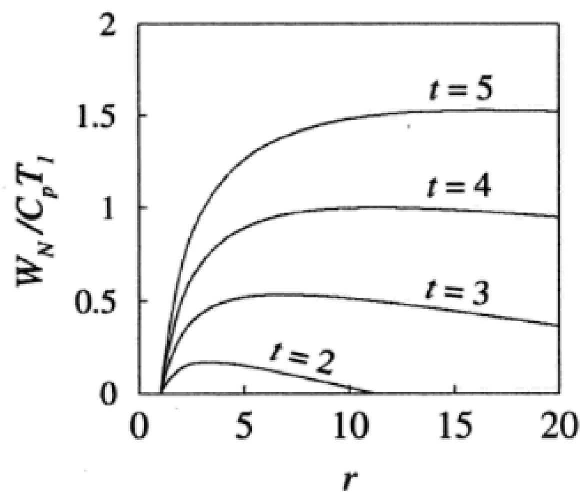


Fig. 5.9 r vs $\frac{W_N}{C_p\ T_1}$

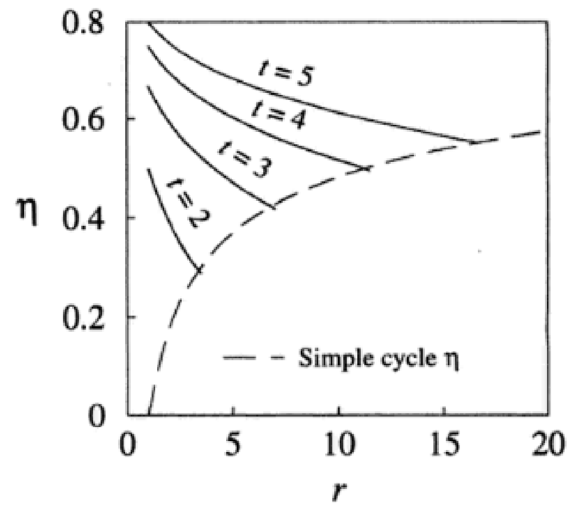


Fig. 5.10 r vs efficiency

It may be noticed that the T-s diagram is unchanged in outline from that of the simple gas turbine cycle, as can be seen from Figs. 5.3 and 5.8, except for the presence of the heat exchanger as indicated by the two dotted lines 4-5 and 2-6. It may be seen that the temperature of the compressed air has been raised from T_2 to T_5 in the heat exchanger resulting in the fall in temperature of the exhaust gases from T_4 to T_6. Now, for a unit mass flow rate

Compressor work $[W_C]$

$$W_{12} = (h_2 - h_1) = C_p(T_2 - T_1) \tag{5.16}$$

Heat addition $[Q]$

$$Q_{53} = (h_3 - h_5) = C_p(T_3 - T_5) \tag{5.17}$$

Turbine work $[W_T]$

$$W_{34} = (h_3 - h_4) = C_p(T_3 - T_4) \tag{5.18}$$

Since, the expressions $W_N = W_T - W_C$ is identical compared to a simple cycle (Eq.5.5), we have,

$$\frac{W_N}{C_pT_1} = t\left(1-\frac{1}{c}\right)-(c-1)$$

However, the expression for η will be different

$$\frac{W_N}{Q} = \eta = \frac{C_pT_1\left[t\left(1-\frac{1}{c}\right)-(c-1)\right]}{C_p(T_3-T_5)}$$

Since, $T_5 = T_4$

$$\eta = \frac{C_pT_1\left[t\left(1-\frac{1}{c}\right)-(c-1)\right]}{C_pT_1\left(\frac{T_3}{T_1}-\frac{T_4}{T_3}\ \frac{T_3}{T_1}\right)}$$

$$= \frac{t\left(1-\frac{1}{c}\right)-(c-1)}{t-\frac{t}{c}} = \frac{t-c}{t}$$

$$\eta = 1-\frac{c}{t} \qquad (5.19)$$

As can be seen from Eq. 5.19 the efficiency of the heat exchange cycle is not independent of maximum cycle temperature and clearly increases as t is increased. Further, it is evident that for a given value of t the efficiency increases with decrease in pressure ratio and *not* with increase in pressure ratio as for the simple cycle.

In Fig. 5.10 the solid line curves represent the above equation (Eq. 5.19). Each curve for t starts at $r = 1$ with a value of $\eta = 1-\frac{1}{t}$, which is the Carnot efficiency. This is to be expected because in this limiting case the Carnot requirement of complex external heat reception and rejection at the upper and lower cycle temperatures is satisfied. The curves fall with increasing pressure ratio until a value corresponding to the value of $\sqrt{t}$ equals c is reached, and at this point the efficiency becomes that of the simple cycle. At this pressure ratio $T_4 = T_2$ and the output is maximum. For higher values of r a heat exchanger would *cool* the air leaving the compressor reducing the efficiency.

The specific output is unchanged by the additions of a heat exchanger. Compare the curves of work output of Fig. 5.4 and Fig. 5.9. From the curves of efficiency for heat exchange cycle (Fig. 5.10) it can be concluded that to obtain an appreciable improvement in efficiency by heat exchange cycle we must have a value of r appreciably less than that for the optimum specific work output. It may be noted that it is not necessary to use a higher cycle pressure ratio as the maximum cycle temperature is increased.

The following observations can be made from the performance curves:

(i) With heat exchanger cycle, the cycle efficiency reduces as the pressure ratio increases, which is opposite to that of a simple cycle. This is due to the fact that as the pressure ratio increases the delivery temperature from the compressor increases and ultimately will exceed that of the exhaust gas from the turbine. Then heat in the heat exchanger will be lost from the air to the exhaust gases instead of desired gain. The efficiency with lower temperatures, say at $t = 2$, is seen to become negative soon after the pressure ratio 11.3 is exceeded (refer Fig. 5.10 $W_N < 0$). The reason is that the temperature at compressor outlet actually exceeds the assumed combustion temperature in this case.

(ii) In many cases, regeneration is not desirable. With high pressure ratios, efficiencies are higher without regeneration, again because loss of heat from the compressed air to the exhaust gases.

(iii) Efficiency with heat exchanger cycle rises very rapidly with increase in maximum temperature of the cycle.

(iv) Lower pressure ratios and high cycle temperatures are favourable for the regenerative cycle, since a large heat recovery is then possible.

(v) For a given temperature-ratio, the curve falls with increasing value of pressure ratio until a value of c given by $c^2 = t$ is reached (Fig. 5.10). After this, the efficiency is equal to the ideal cycle without regeneration. Any further increase of pressure ratio will yield an efficiency which is lower than this value and is of no interest.

5.4 THE REHEAT CYCLE

A good improvement in specific work output can be obtained by splitting the expansion and reheating the gas between the high pressure and low pressure turbines. Figure 5.11 shows the schematic arrangement of the reheat cycle along with the p-V (Fig. 5.12) and T-s (Fig. 5.13) diagrams. Figures 5.14 and 5.15 show the performance curves of the cycle. The turbine work increase is obvious from the fact that the vertical distance between any pair of constant-pressure lines increases as the entropy increases. Thus

$$(T_3 - T_4) + (T_5 - T_6) > (T_3 - T_{4'})$$

Consider the T-s diagram in Fig. 5.13 in which the expansion is carried out in two stages, reheating of the working fluid to the upper limit of temperature T_3 taking place between the stages. Let the pressure ratio in compression be r and the pressure ratios of the expansion stages be r_3 and r_4 so that

$$r = r_3 \times r_4 \tag{5.20}$$

$$c = r^{(\gamma-1)/\gamma} = \frac{T_2}{T_1} \tag{5.21}$$

$$c_3 = r_3^{(\gamma-1)/\gamma} = \frac{T_3}{T_4} \tag{5.22}$$

$$c_4 = r_4^{(\gamma-1)/\gamma} = \frac{T_5}{T_6} \tag{5.23}$$

Therefore,

$$c = c_3 \times c_4 \tag{5.24}$$

For a unit quantity of fluid flow

Compressor work $[W_C]$

$$W_{12} = (h_2 - h_1) = C_p(T_2 - T_1) \tag{5.25}$$

Heat addition $[Q]$

$$Q_{23} = (h_3 - h_2) = C_p(T_3 - T_2) \tag{5.26}$$

$$Q_{45} = (h_5 - h_4) = C_p(T_5 - T_4) \tag{5.27}$$

Turbine work $[W_T]$

$$W_{34} = (h_3 - h_4) = C_p(T_3 - T_4) \tag{5.28}$$

$$W_{56} = (h_5 - h_6) = C_p(T_5 - T_6) \tag{5.29}$$

Net work output $[W_N]$

$$\begin{aligned} &= W_T - W_C \\ &= C_p(T_3 - T_4) + C_p(T_5 - T_6) - C_p(T_2 - T_1) \\ &= C_pT_1\left[\left(\frac{T_3}{T_1} - \frac{T_4}{T_1}\right) + \left(\frac{T_5}{T_1} - \frac{T_6}{T_1}\right) - \left(\frac{T_2}{T_1} - 1\right)\right] \end{aligned}$$

$$\frac{W_N}{C_pT_1} = \frac{T_3}{T_1} - \frac{T_4}{T_3}\frac{T_3}{T_1} + \frac{T_5}{T_1} - \frac{T_6}{T_5}\frac{T_5}{T_1} - \frac{T_2}{T_1} + 1 \tag{5.30}$$

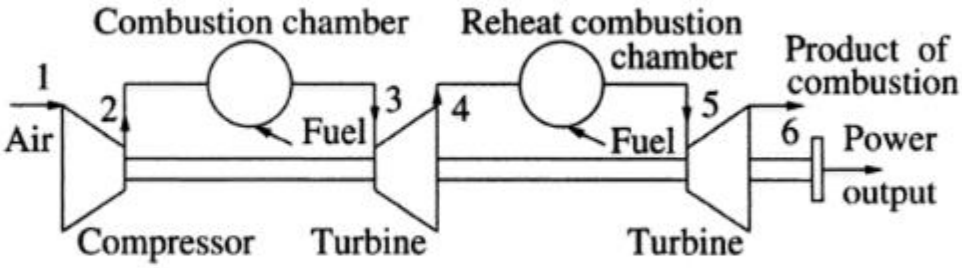
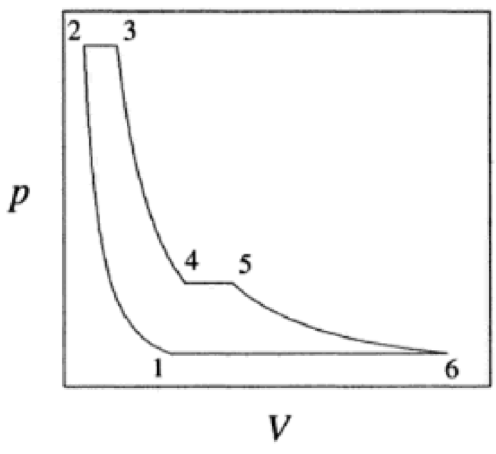
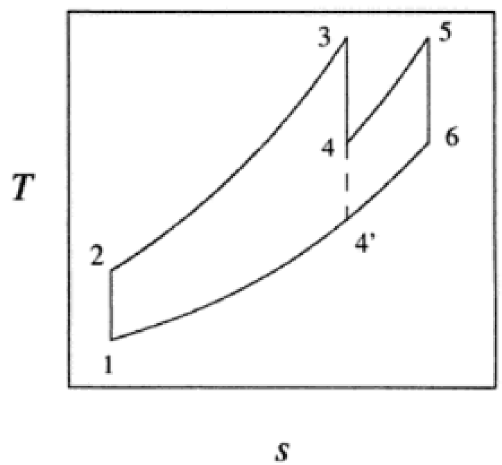


Fig. 5.11 Schematic arrangement of a reheat cycle





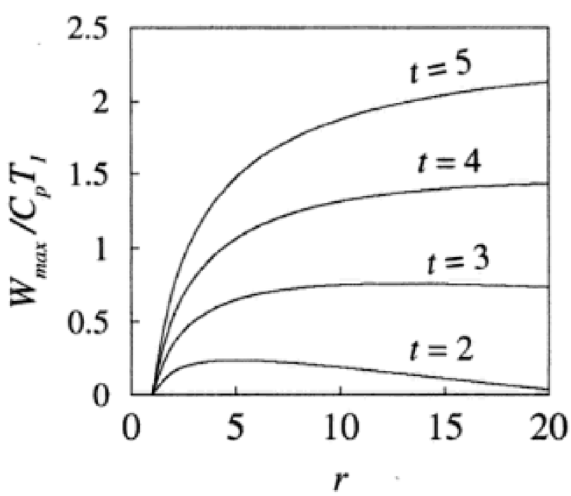


Fig. 5.14 r *vs* $\frac{W_N}{C_p \; T_1}$

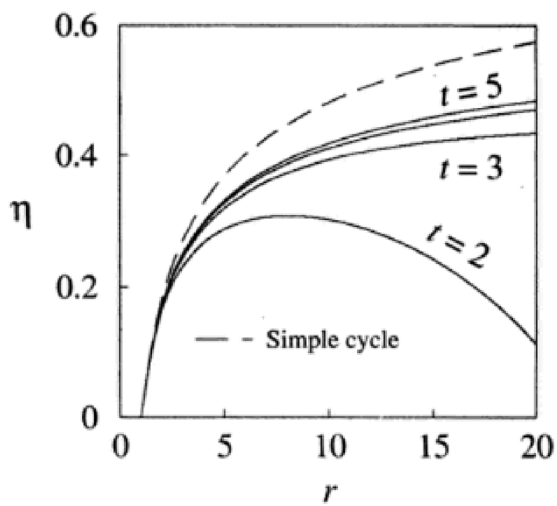


Fig. 5.15 r *vs efficiency*

Since $T_5 = T_3$ and $c_3 \cdot c_4 = c$, the above equation simplifies to

$$\frac{W_N}{C_p T_1} = 2t - \frac{t}{c_3} - \frac{t}{c_4} - c + 1 \tag{5.31}$$

For maximum output

$$\frac{d}{dc_3}\left(\frac{W}{C_p T_1}\right) = 0 \tag{5.32}$$

writing c_4 in terms of c and c_3 and treating c, T and t as constants with respect to c_3

$$\frac{t}{c_3^2} - \frac{t}{c} = 0$$

$$c_3 = \sqrt{c} = c_4 \tag{5.33}$$

Equation 5.33 shows that to obtain maximum output the stage pressure ratios must be same and this should be the square root of the overall pressure ratio for two stage expansion. Now,

$$\frac{W_{max}}{C_p T_1} = 2t - \frac{2t}{\sqrt{c}} - c + 1 \tag{5.34}$$

$$= 2t\left(1 - \frac{1}{\sqrt{c}}\right) - (c-1) \tag{5.35}$$

$$\eta_{max} = \frac{W_{max}}{Q} = \frac{C_pT_1\left[2t - c + 1 - \frac{2t}{\sqrt{c}}\right]}{C_p[T_3 - T_2] + C_p[T_5 - T_4]}$$

$$= \frac{2t\left(1 - \frac{1}{\sqrt{c}}\right) - (c-1)}{\frac{T_3}{T_1} - \frac{T_2}{T_1} + \frac{T_5}{T_1} - \frac{T_4}{T_1}}$$

$$\eta_{max} = \frac{2t\left(1 - \frac{1}{\sqrt{c}}\right) - (c-1)}{2t - c - \frac{t}{\sqrt{c}}} \tag{5.36}$$

Comparisons of the $\frac{W_N}{C_pT_1}$ curves (Fig. 5.14) with the simple cycle (Fig. 5.4) indicates that reheat markedly increase the specific output, but the curves for efficiency indicate that this is achieved at the expense of efficiency (Fig. 5.15). This is to be expected because a less efficient cycle ($4'456$) (refer Fig. 5.13) is added to the simple cycle – less efficient because it operates over a smaller temperature range. Note that the rate of reduction in η becomes less as maximum cycle temperature is increased.

5.5 THE REHEAT AND HEAT EXCHANGE CYCLE

It is seen that improvement in specific power output is achieved in reheat cycle at the expense of the efficiency. This can be overcome by adding a heat exchanger to the reheat cycle. The schematic arrangements of the reheat cycle with heat exchanger is given in Fig. 5.16. The corresponding p-V and T-s diagrams are shown in Figs. 5.17 and 5.18 respectively. Figures 5.19 and 5.20 show the performance curves of the cycle. For a unit quantity of mass flow,

Compressor work $[W_C]$

$$W_{12} = C_p(T_2 - T_1) \tag{5.37}$$

Heat addition $[Q]$

$$Q_{73} + Q_{45} = C_p(T_3 - T_7) + C_p(T_5 - T_4) \tag{5.38}$$

Turbine work $[W_T]$

$$W_{34} + W_{56} = C_p(T_3 - T_4) + C_p(T_5 - T_6) \tag{5.39}$$

Net work output $[W_N]$

$$\begin{aligned} &= W_T - W_C \\ &= C_p(T_3 - T_4) + C_p(T_5 - T_6) - C_p(T_2 - T_1) \end{aligned}$$

$$= \quad C_pT_1\left(\frac{T_3}{T_1} - \frac{T_4}{T_1} + \frac{T_5}{T_1} - \frac{T_6}{T_1} - \frac{T_2}{T_1} + 1\right) \qquad (5.40)$$

The maximum work output expression will be same as Eq. 5.34 since heat exchanger will improve only the efficiency and not the work output. Hence,

$$\frac{W_{max}}{C_pT_1} = 2t - c + 1 - \frac{2t}{\sqrt{c}}$$

$$\eta_{max} = \frac{W_{max}}{Q} = \frac{C_pT_1\left[2t - c + 1 - \frac{2t}{\sqrt{c}}\right]}{C_p[T_3 - T_7] + C_p[T_5 - T_4]} \qquad (5.41)$$

$$= \frac{2t - c + 1 - \frac{2t}{\sqrt{c}}}{\frac{T_3}{T_1} - \frac{T_7}{T_1} + \frac{T_5}{T_1} - \frac{T_4}{T_1}}$$

Since $T_7 = T_4$ and $T_5 = T_3$, we have

$$\eta_{max} = \frac{2t - c + 1 - \frac{2t}{\sqrt{c}}}{\frac{T_3}{T_1} - \frac{T_4}{T_1} + \frac{T_3}{T_1} - \frac{T_4}{T_1}} = \frac{2t - c + 1 - \frac{2t}{\sqrt{c}}}{2\left(\frac{T_3}{T_1} - \frac{T_4}{T_1}\right)}$$

$$= \frac{2t - c + 1 - \frac{2t}{\sqrt{c}}}{2t - \frac{2t}{\sqrt{c}}}$$

$$\eta_{max} = 1 - \frac{c - 1}{2t - \frac{2t}{\sqrt{c}}} \qquad (5.42)$$

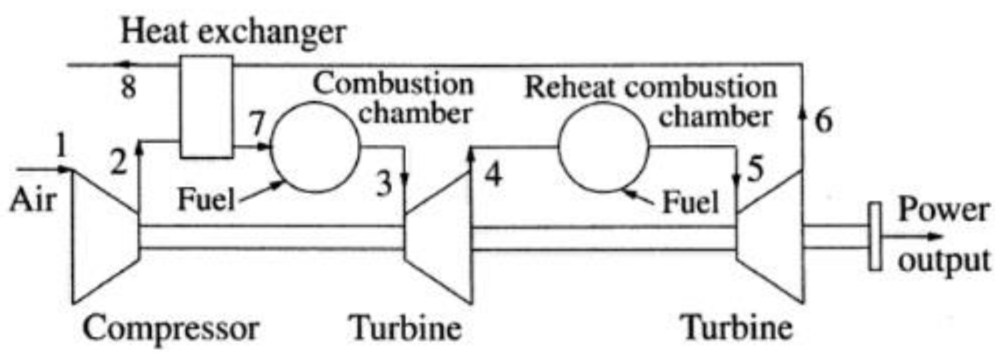


Fig. 5.16 Schematic arrangement of a reheat cycle with heat exchanger

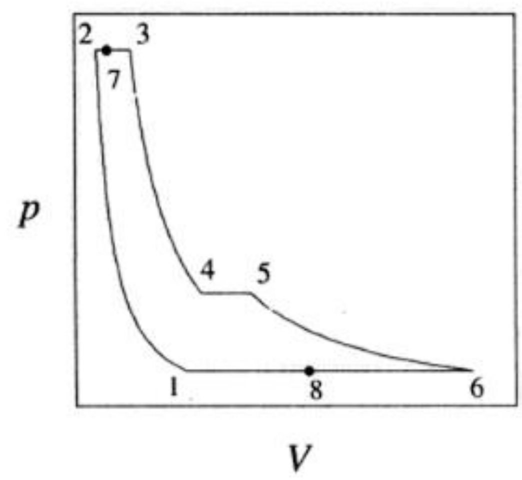


Fig. 5.17 p-V diagram

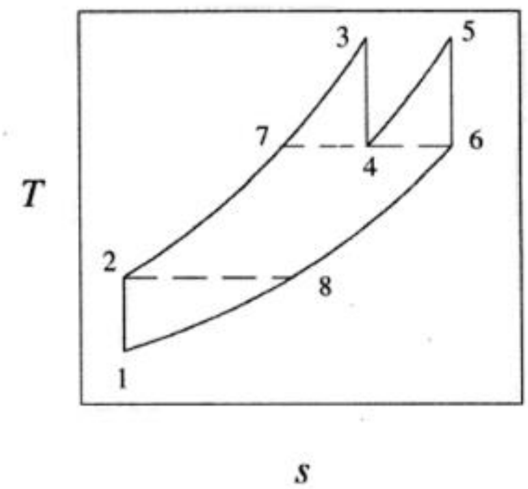


Fig. 5.18 T-s diagram

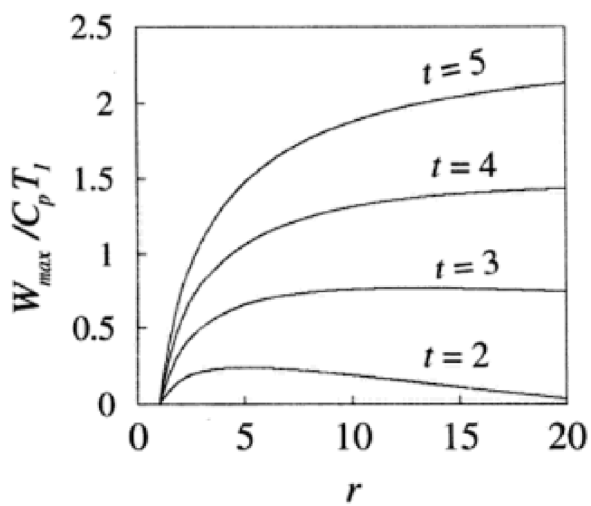


Fig. 5.19 r vs $\frac{W_{max}}{C_p\ T_1}$

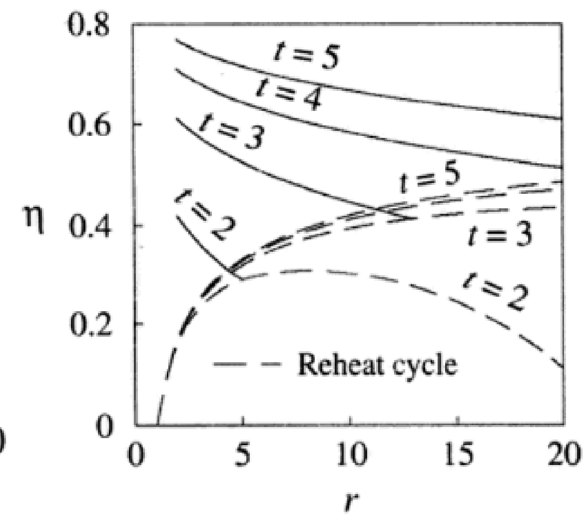


Fig. 5.20 r *vs efficiency*

The higher exhaust gas temperature is now fully utilized in the heat exchanger. In fact, when a heat exchanger is employed, the efficiency is higher with reheat than without. The family of constant t lines exhibit the same features as those for simple cycle with heat exchanger. Each curve starts with the Carnot values at $r = 1$ and falls with increasing r to meet the corresponding curve of the reheat cycle without heat exchanger at a value of r corresponding to maximum specific output.

5.6 THE INTERCOOLED CYCLE

We have seen in Section 5.4 that specific output of the cycle can be improved by increasing the turbine work output incorporating the reheat cycle. Another way of achieving the same is reducing the work of compression, i.e., compression in more than one stage and using an intercooler in between. That is, the compression of the working fluid is cut off at some intermediate pressure and the fluid is cooled by passing it through a heat exchanger supplied with coolant from some external source before being compressed in a second stage to the required pressure ratio, a certain improvement in overall output can be achieved. The details of such an arrangement are shown in Fig. 5.21. The corresponding p-V and T-s diagrams are as shown in Figs. 5.22 and 5.23 respectively. Figures 5.24 and 5.25 show the performance curves of the cycle. Now, for a unit quantity of mass flow

Compressor work $[W_C]$

$$W_{12} + W_{34} = C_p(T_2 - T_1) + C_p(T_4 - T_3) \tag{5.43}$$

Heat addition $[Q]$

$$Q_{45} = C_p(T_5 - T_4) \tag{5.44}$$

Turbine work $[W_T]$

$$W_{56} = C_p(T_5 - T_6) \tag{5.45}$$

Net work output $[W_N]$

$$\begin{aligned} &= W_T - W_C \\ &= C_p(T_5 - T_6) - C_p(T_2 - T_1) - C_p(T_4 - T_3) \\ &= C_pT_1\left(\frac{T_5}{T_1} - \frac{T_6}{T_1} - \frac{T_2}{T_1} + 1 - \frac{T_4}{T_1} + \frac{T_3}{T_1}\right) \end{aligned} \tag{5.46}$$

Let,

$$\frac{T_5}{T_1} = t; \qquad \frac{T_5}{T_6} = c$$

$$\frac{T_2}{T_1} = c_1; \qquad \frac{T_4}{T_3} = c_2 \tag{5.47}$$

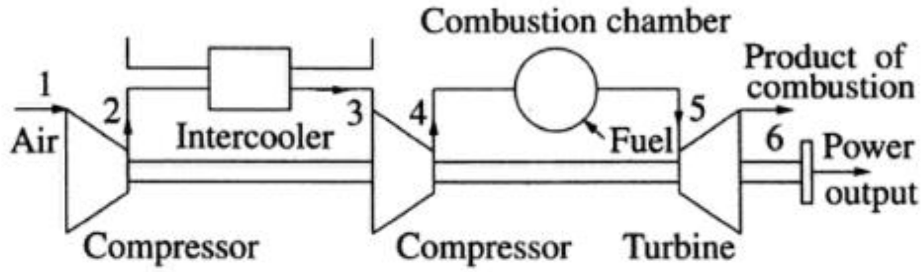


Fig. 5.21 Schematic arrangement of a intercooled cycle

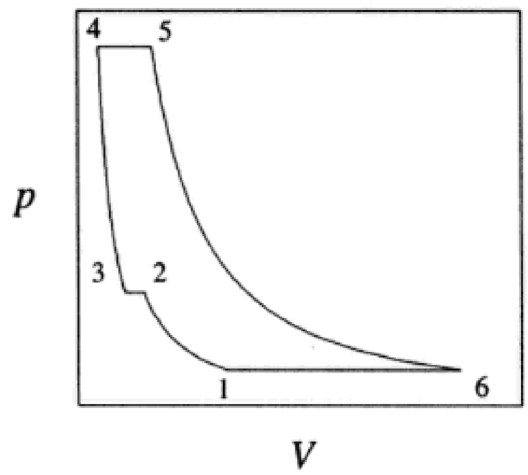


Fig. 5.22 p-V diagram

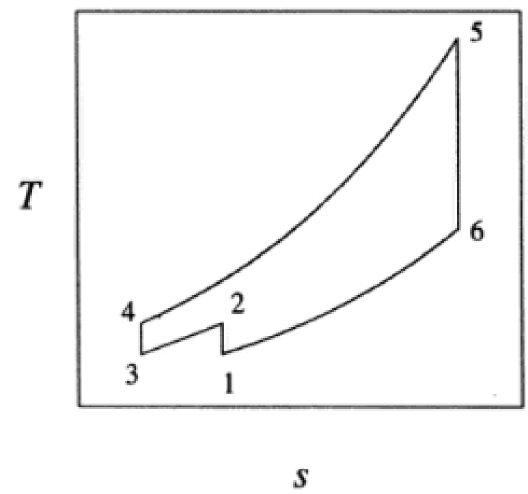


Fig. 5.23 T-s diagram

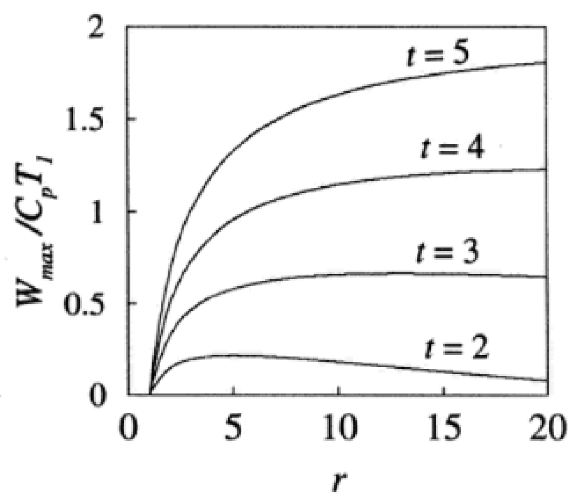


Fig. 5.24 r vs $\frac{W_{max}}{C_p\ T_1}$

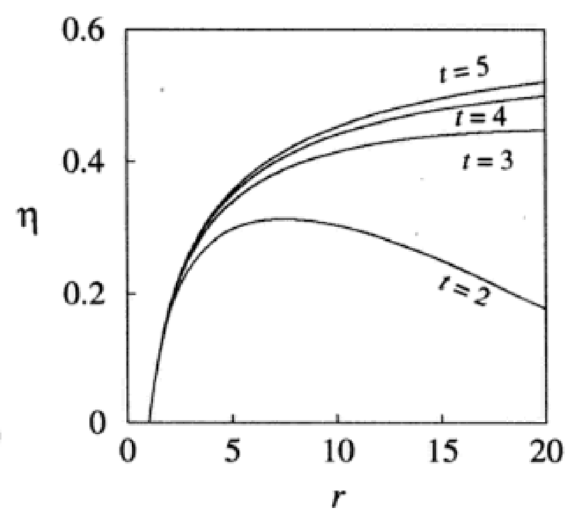


Fig. 5.25 r vs efficiency

It can be shown that for maximum power output and perfect intercooling, (following the similar procedure as we have done for reheat cycle)

$$c_1 = \sqrt{c} = c_2 \tag{5.48}$$

$$\begin{aligned}
\frac{W_{max}}{C_p T_1} &= \frac{T_5}{T_1} - \frac{T_6}{T_5}\frac{T_5}{T_1} - \frac{T_2}{T_1} + 1 - \frac{T_4}{T_3} + 1 \\
&= t - \frac{t}{c} - \sqrt{c} + 1 - \sqrt{c} + 1
\end{aligned}$$

$$\frac{W_{max}}{C_p T_1} = t - \frac{t}{c} - 2\sqrt{c} + 2 \tag{5.49}$$

$$\begin{aligned}
\eta_{max} &= \frac{W_{max}}{Q} = \frac{C_p T_1 (t - \frac{t}{c}) - 2(\sqrt{c} - 1)}{C_p (T_5 - T_4)} \\
&= \frac{(t - \frac{t}{c}) - 2(\sqrt{c} - 1)}{\frac{T_5}{T_1} - \frac{T_4}{T_3}}
\end{aligned}$$

Since $T_3 = T_1$

$$\eta_{max} = \frac{t - \sqrt{c} - \frac{t}{c} - \sqrt{c} + 2}{t - \sqrt{c}} \tag{5.50}$$

$$= 1 - \frac{\frac{t}{c} + \sqrt{c} - 2}{t - \sqrt{c}} \tag{5.51}$$

The specific work output and the efficiency curves are as shown in Figs. 5.24 and 5.25 respectively. Intercooling will help to increase the net work output of the cycle. Because of the lower compressor outlet temperature, the fuel flow rate to obtain a given turbine inlet temperature will increase. Therefore, the thermal efficiency of the intercooled cycle will be less than that for a simple cycle.

Intercooling is useful when the pressure ratios are high and the efficiency of the compressor is low. Due to lower compressor outlet temperature there will be more scope for adding a heat exchanger. Due to regeneration, a substantial amount of heat from exhaust gases can be recovered. This will be seen in the next section.

5.7 THE INTERCOOLED CYCLE WITH HEAT EXCHANGER

As seen in the previous section, by intercooling we can improve the work output of the cycle. Suppose we have to improve the efficiency also, then we can add a heat exchanger to the intercooled cycle. The basic schematic arrangements along with the p-V and T-s diagrams are shown in Figs. 5.26, 5.27 and 5.28 respectively. Figures 5.29 and 5.30 show the performance curves of the cycle. Now, for an unit quantity of mass flow

Compressor work $[W_C]$

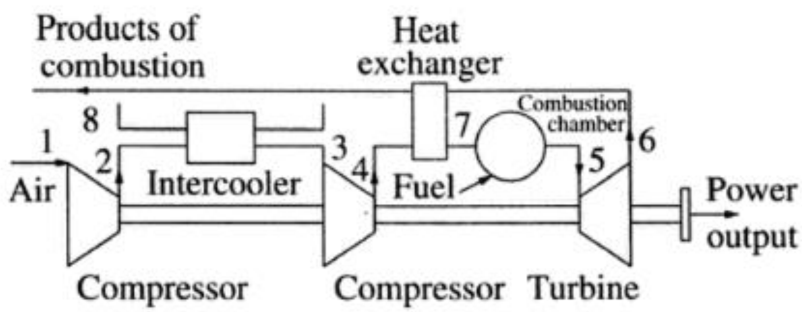


Fig. 5.26 Schematic arrangement of an intercooled cycle with heat exchanger

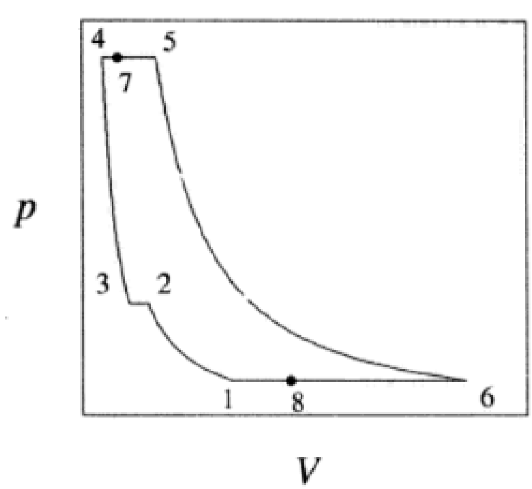


Fig. 5.27 p–V diagram

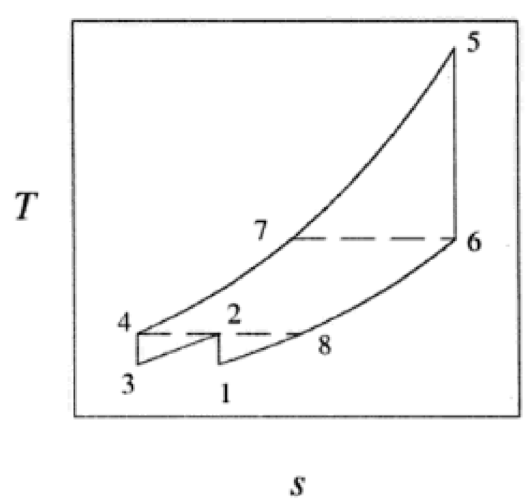


Fig. 5.28 T-s diagram

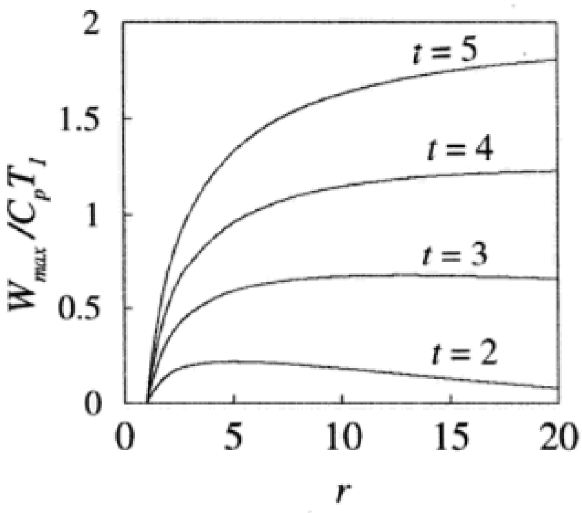


Fig. 5.29 r vs $\frac{W_{max}}{C_p\ T_1}$

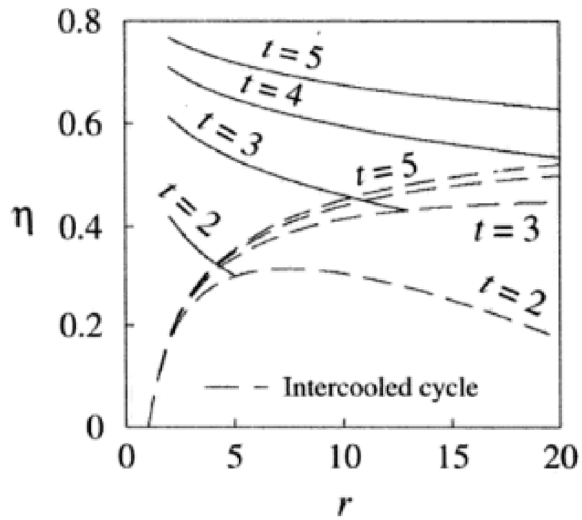


Fig. 5.30 r vs efficiency

$$W_{12} + W_{34} = C_p(T_4 - T_3) + C_p(T_2 - T_1) \tag{5.52}$$

Heat addition $[Q]$

$$Q_{75} = C_p(T_5 - T_7) \tag{5.53}$$

Turbine work $[W_T]$

$$W_{56} = C_p(T_5 - T_6) \tag{5.54}$$

Net work output $[W_N]$

$$= W_T - W_C$$

$$= C_p(T_5 - T_6) - C_p(T_2 - T_1) - C_p(T_4 - T_3)$$

The above expression is identical to equation in a simple cycle with intercooling, Hence, the maximum specific work output is given by

$$\frac{W_{max}}{C_pT_1} = t - \frac{t}{c} - 2\sqrt{c} + 2$$

$$\eta_{max} = \frac{W_{max}}{Q} = \frac{C_pT_1\left(t - \frac{t}{c} - 2\sqrt{c} + 2\right)}{C_p(T_5 - T_7)} \tag{5.55}$$

$$= \frac{t - \frac{t}{c} - 2\sqrt{c} + 2}{\frac{T_5}{T_1} - \frac{T_7}{T_1}}$$

Since $T_7 = T_6$

$$\eta_{max} = \frac{t - \frac{t}{c} - 2\sqrt{c} + 2}{\frac{T_5}{T_1} - \frac{T_6}{T_5}\frac{T_5}{T_1}} = \frac{t - \frac{t}{c} - 2\sqrt{c} + 2}{t - \frac{t}{c}}$$

$$= 1 - \frac{2(\sqrt{c} - 1)}{t(1 - \frac{1}{c})} \tag{5.56}$$

Figures 5.29 and 5.30 give the variation of specific power output and efficiency respectively with r for various t. As already discussed the specific power output increases over a simple cycle because of intercooling. Because of the addition of heat exchanger, substantial recovery of heat from the exhaust gas is possible which increases the efficiency also. It may be noted that the improvement in efficiency is substantial at lower pressure ratios and higher turbine inlet temperatures.

5.8 THE INTERCOOLED AND REHEAT CYCLE

We have seen – in the previous section – how to improve the efficiency of an intercooled cycle by adding an heat exchanger. We can further improve the specific work output of the intercooled cycle by adding reheat. The schematic diagram of the cycle is as shown in Fig. 5.31. The corresponding

p-V and T-s diagrams are as shown in Figs. 5.32 and 5.33 respectively. Figures 5.34 and 5.35 show the performance curves of the cycle. For a unit quantity of mass flow,

Compressor work $[W_C]$

$$W_{34} + W_{12} = C_p(T_4 - T_3) + C_p(T_2 - T_1) \tag{5.57}$$

Heat addition $[Q]$

$$Q_{45} + Q_{67} = C_p(T_5 - T_4) + C_p(T_7 - T_6) \tag{5.58}$$

Turbine work $[W_T]$

$$W_{56} + W_{78} = C_p(T_5 - T_6) + C_p(T_7 - T_8) \tag{5.59}$$

Net work output $[W_N]$

$$\begin{aligned} &= W_T - W_C \\ &= C_p(T_5 - T_6 + T_7 - T_8) - \\ &\quad C_p(T_4 - T_3 + T_2 - T_1) \qquad (5.60) \\ &= C_pT_1\left[\frac{T_5}{T_1} - \frac{T_6}{T_1} + \frac{T_7}{T_1} - \frac{T_8}{T_1} - \frac{T_4}{T_1} + \frac{T_3}{T_1} - \frac{T_2}{T_1} + 1\right] \end{aligned}$$

$$\frac{W_N}{C_pT_1} = \frac{T_5}{T_1} - \frac{T_6}{T_5}\frac{T_5}{T_1} + \frac{T_5}{T_1} - \frac{T_8}{T_7}\frac{T_5}{T_1} - \frac{T_4}{T_3} + \frac{T_1}{T_1} - \frac{T_2}{T_1} + 1$$

By following the usual procedure to get the maximum power output, we can show that

$$\begin{aligned} \frac{W\text{max}}{C_pT_1} &= t - \frac{t}{\sqrt{c}} + t - \frac{t}{\sqrt{c}} - \sqrt{c} + 1 - \sqrt{c} + 1 \\ &= 2t - \frac{2t}{\sqrt{c}} - 2\sqrt{c} + 2 \end{aligned}$$

$$\frac{W_{max}}{C_pT_1} = 2\left(t - \frac{t}{\sqrt{c}} - \sqrt{c} + 1\right) \tag{5.61}$$

$$\eta_{max} = \frac{W_{max}}{Q}$$

$$= \frac{2C_pT_1\left(t - \frac{t}{\sqrt{c}} - \sqrt{c} + 1\right)}{C_p(T_5 - T_4) + C_p(T_7 - T_6)} \qquad (5.62)$$

$$= \frac{2C_pT_1\left(t - \frac{t}{\sqrt{c}} - \sqrt{c} + 1\right)}{C_pT_1\left(\frac{T_5}{T_1} - \frac{T_4}{T_1} + \frac{T_7}{T_1} - \frac{T_6}{T_1}\right)}$$

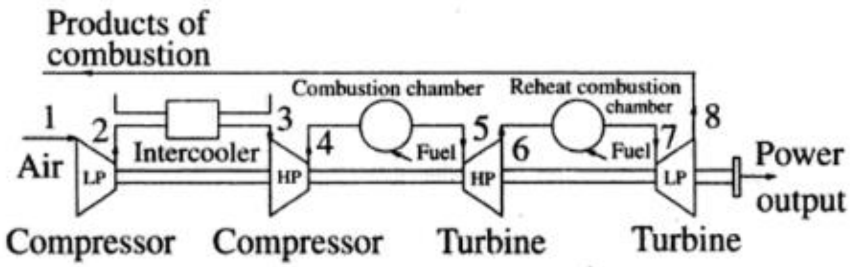


Fig. 5.31 Schematic arrangement of an intercooled and reheat cycle

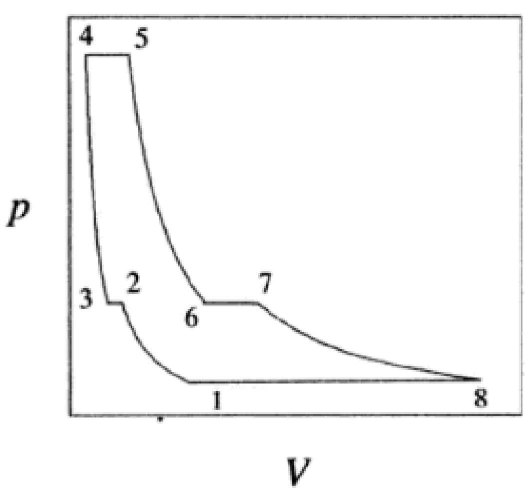


Fig. 5.32 p–V diagram

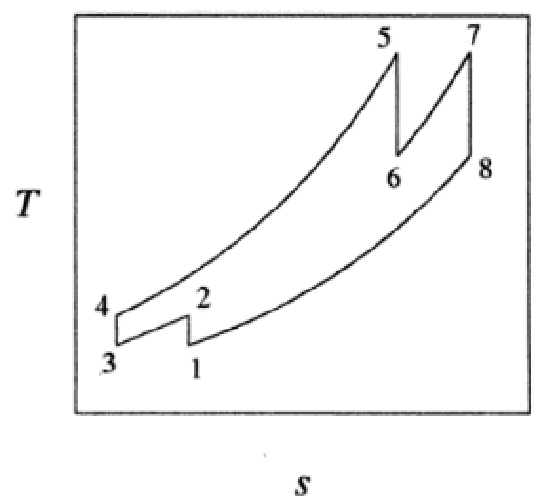


Fig. 5.33 T-s diagram

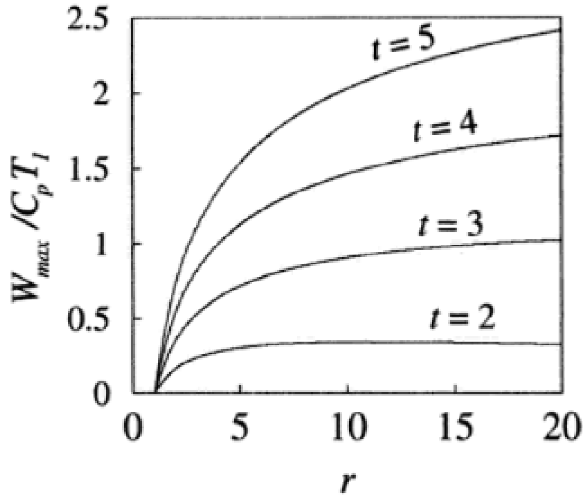


Fig. 5.34 r vs $\frac{W_{max}}{C_p\ T_1}$

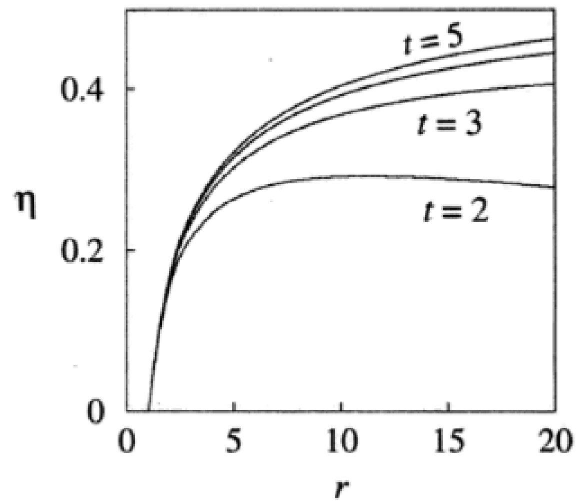


Fig. 5.35 r vs efficiency

$$= \frac{2\left(t - \frac{t}{\sqrt{c}} - \sqrt{c} + 1\right)}{\left(2t - \sqrt{c} - \frac{t}{\sqrt{c}}\right)}$$

$$= \frac{2(t - \sqrt{c}) - 2\left(\frac{t}{\sqrt{c}} - 1\right)}{\left(2t - \sqrt{c} - \frac{t}{\sqrt{c}}\right)} \tag{5.63}$$

$$= 1 - \frac{\frac{t}{\sqrt{c}} + \sqrt{c} - 2}{2t - \sqrt{c} - \frac{t}{\sqrt{c}}} \tag{5.64}$$

The specific work output and efficiency curves are shown in Figs. 5.34 and 5.35 respectively.

21.1 CLOSED CYCLE AND OPEN CYCLE PLANTS

The essential components of a gas turbine (GT) power plant are the compressor, combustion chamber and the turbine. The air standard cycle of a GT plant is the Brayton cycle.

A GT plant can either be open or closed. Figure 21.1 shows the arrangement of an open-cycle plant which is more common. The compressor takes in ambient air and raises its pressure. The temperature of air is increased when it flows through a combustion chamber where a fossil fuel is burned or a heat exchanger (nuclear fuel being used as a source of energy) is present. The high-pressure, high-temperature working fluid, mostly a gas, enters a turbine where it expands to a low-pressure (equal to or a little above the atmospheric pressure) fluid. In an open unit, the gas is released from the turbine to the surroundings and in a closed unit, the working fluid is cooled in a cooler after the exhaust from the turbine and is returned to the compressor (Fig. 21.2). A significant part of the power developed by the turbine is utilized to drive the compressor and any other auxiliaries, and the remainder is available as useful work.

21.2 ADVANTAGES OF A GT PLANT

The advantages of a GT plant for power generation are being enumerated below:

(1) *Warm-up Time* Once the turbine is brought up to the rated speed by the starting motor and the fuel is ignited, the GT will accelerate from cold start to full load without warm-up time.

(2) *Low Weight and Size* The weight of the plant per kW output is low, which is a favourable feature in all vehicles (land, air and sea). In utilities also, the foundation of the plant is lighter.

(3) *Fuel Flexibility* Any hydrocarbon fuel from high octane gasoline to heavy diesel oil and pulverized coal can be used effectively.

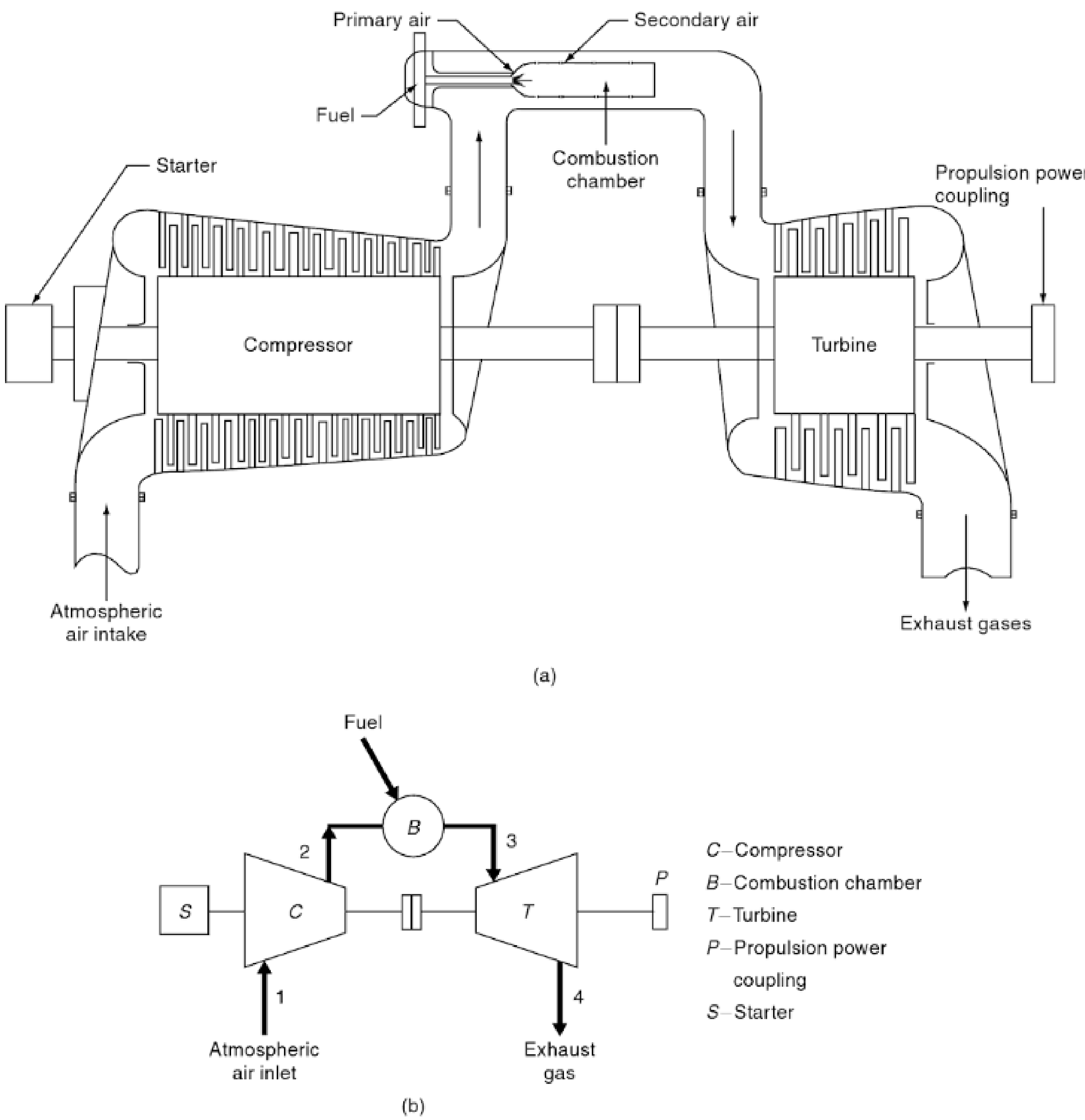


Fig. 21.1 (*a*) *Component parts of a simple open cycle (constant-pressure combustion) gas turbine,* (*b*) *Simple open cycle gas turbine*

(4) ***Floor Space*** Because of its smaller size, the floor space required for its installation is less.

(5) ***Start-up and Shut-down*** A GT plant can be started up as well as shut down quickly, like a diesel engine. Thus it is eminently suitable to meet the peak load demand of a certain region.

(6) ***High Efficiency*** Suitable blade cooling permits the use of high GT inlet temperature (as high as 1300°C) yielding a high thermal efficiency (on the order of 37%).

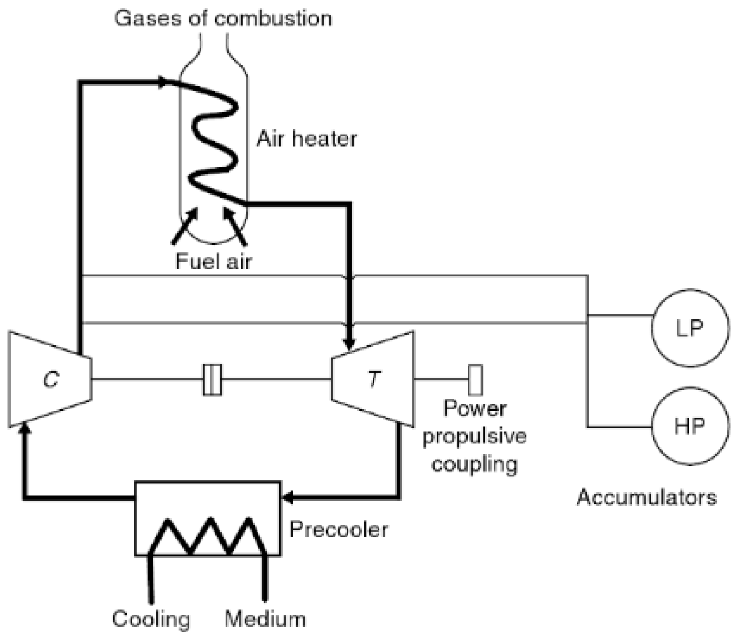


Fig. 21.2 *Simple closed-cycle GT plant*

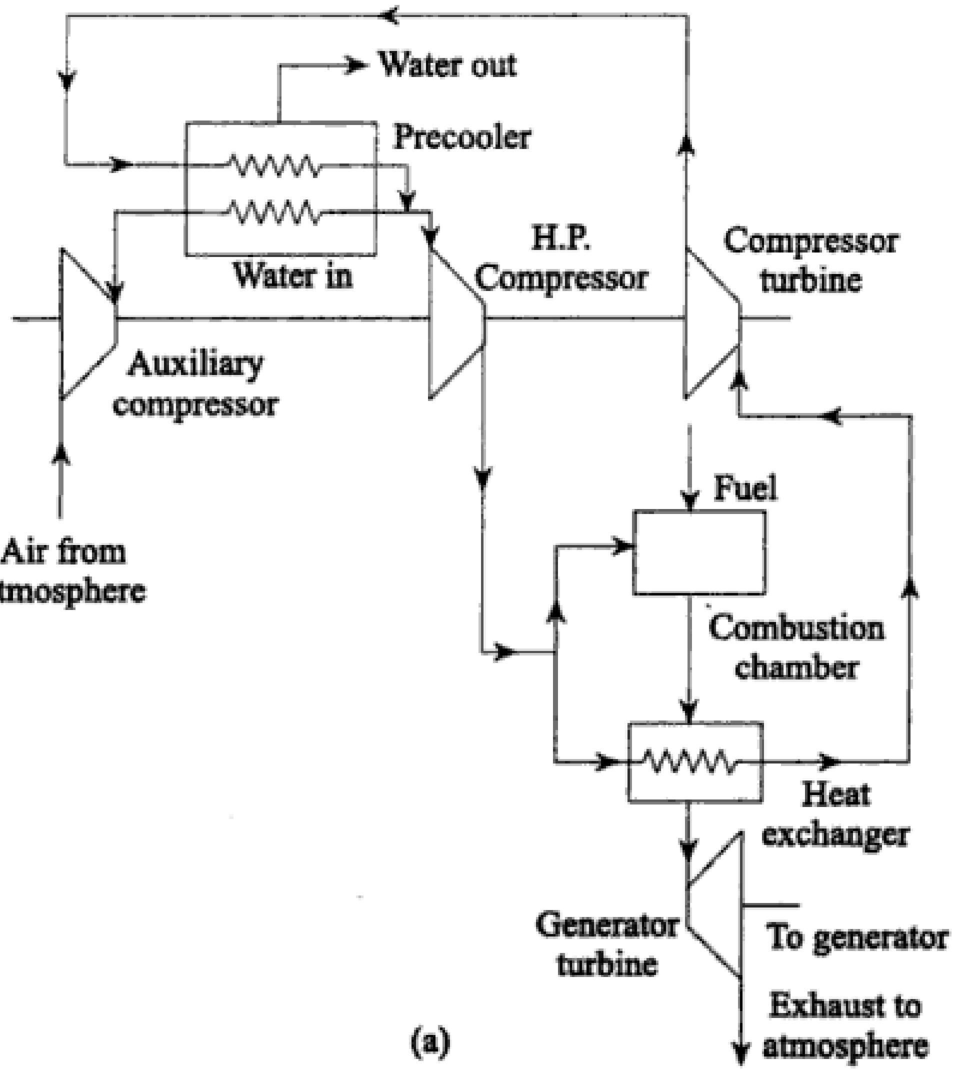
21.3 DISADVANTAGES OF A GT PLANT

These are enlisted below:

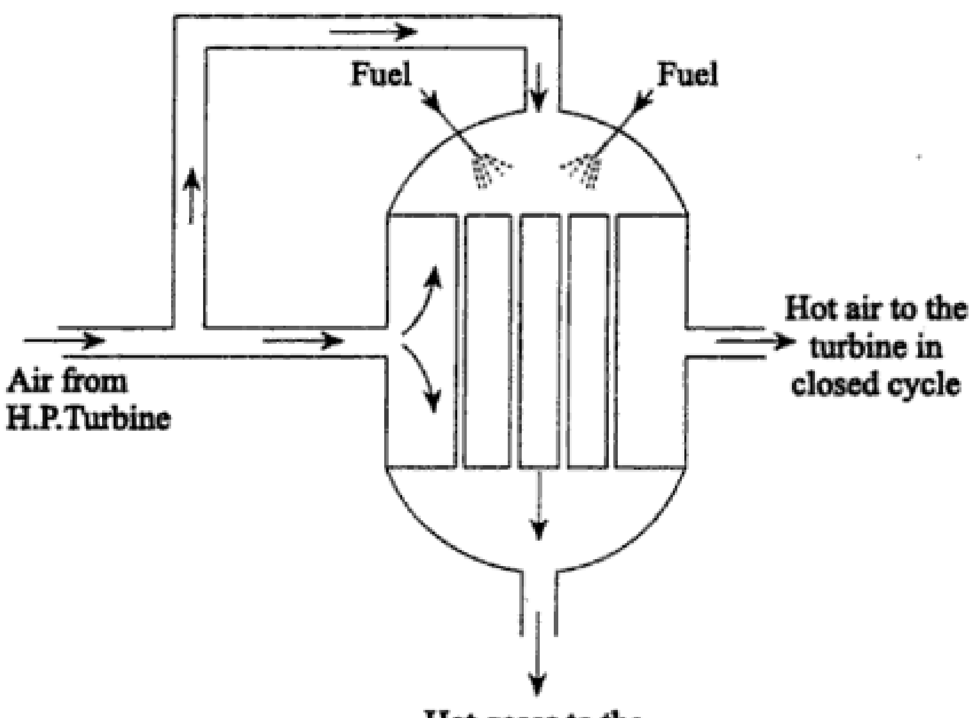
1. Part load efficiency is low
2. Highly sensitive to component efficiency like η_c and η_T
3. The efficiency depends on the ambient condition (p_a and T_a)
4. High air rate is required to limit the maximum GT inlet temperature, as a result of which the exhaust losses are high, unless the waste heat in it is utilized
5. Compressor work required is quite large, which tells upon the efficiency of the plant
6. Air and gas filters have to be of very high quality so that no dust enters to erode and corrode the turbine blades

21.6 SEMI-CLOSED CYCLE GT PLANT

The advantages of the open cycle plant, viz. quick and easy starting and the closed cycle plant, viz. constant efficiency at all loads and higher unit rating permitting the use of higher back pressure, are combined in a semi-closed cycle gas turbine power plant. Here, part of the compressed air is heated by the gases exiting the combustion chamber (CC) and then expanded in an air turbine which drives the compressor, thus operating in a closed cycle. The remaining air is used in the CC to burn fuel, and the combustion products after heating the air expand in a gas turbine to drive the generator before exhausting to the atmosphere (Fig. 21.15 a). Figure 21.15 (b) shows a combined combustion chamber and a heat exchanger, where hot gases of combustion leave to expand in the gas turbine in the open cycle and the heated air flows to the air turbine in the closed cycle.


Water out
Precooler
Water in
H.P.
Compressor
Compressor
turbine
Auxiliary
compressor
Air from
atmosphere
Fuel
Combustion
chamber
Heat
exchanger
Generator
turbine
To generator
Exhaust to
atmosphere

(a)



(b)

Fig. 11.47 (a) Semi-closed cycle gas turbine plant
(b) Combined combustion chamber and air heater

1. Centrifugal compressors
2. Axial flow compressors

These have been discussed in Chapter 19. The centrifugal compressor is comprised of two major parts, the impeller, or rotating component and the diffusor. The air enters the compressor at the hub and it then moves radially outward through the impeller and into the diffusor. The impeller converts the mechanical energy, available to the compressor, into kinetic energy, plus heat due to friction, in the working media. The diffusor then transforms the kinetic energy in the air into pressure energy in accordance with Bernoulli's principle. The flow through the diffusor is subject to frictional losses as well. Also, because the air leaves the impeller radially, it must normally be turned 90° to enter the combustion chamber or regenerator, involving more frictional losses. The choice of the blade shape (i.e., bent backward, forward or straight radialers) and the compressor rpm depend on stress limits and manufacturing costs.

In general, the *centrifugal compressor*, as compared to axial flow, is more rugged, simpler, relatively insensitive to surface deposits, has a wider stability range, is less expensive, and attains a higher pressure ratio per stage. However, the efficiency is lower, the diameter larger, and it is not readily adaptable to multi-staging. The single-stage compressors for use in industry may obtain efficiencies from 80 to 84% at pressure ratios between 2.5 and 3, while for aircraft use, pressure ratios are between 4 and 4.5 with efficiencies in the range of 76 to 81%.

The important characteristics of the *axial flow compressor* are its high peak efficiencies, adaptability to multistaging to obtain higher overall pressure ratios, high flow-rate capabilities, and relatively small diameter. However, the axial flow compressor is sensitive to changes in air flow and rpm, which result in a rapid drop off in efficiencies, i.e., the stability range of speeds for good efficiencies is small.

The axial flow compressor consists of a series of rotor-stator stages. The rotor comprises a series of blades that move relative to a series of stationary blades called the stator. The blades transmit the mechanical energy into kinetic energy in the air. Compression is accomplished in both the rotor and stator blades into pressure energy (i.e., continually diffusing it from a high velocity to a lower velocity with a corresponding rise in static pressure). The details of the flow diagram and the velocity triangles as well as the power input and efficiency of the compressor are given in Chapter 19.

A brief note on Gas Turbine Combustors

Over a period of five decades, the basic factors influencing the design of combustion systems for gas turbines have not changed, although recently some new requirements have evolved. The key issues may be summarized as follows.

• The temperature of the gases after combustion must be comparatively controlled to suit the highly stressed turbine materials. Development of improved materials and methods of blade cooling, however, has enabled permissible combustor outlet temperatures to rise from about 1100K to as much as 1850 K for aircraft applications.

• At the end of the combustion space the temperature distribution must be of known form if the turbine blades are not to suffer from local overheating. In practice, the temperature can increase with radius over the turbine annulus, because of the strong influence of temperature on allowable stress and the decrease of blade centrifugal stress from root to tip.

• Combustion must be maintained in a stream of air moving with a high velocity in the region of 30-60 m/s, and stable operation is required over a wide range of air/fuel ratio from full load to idling conditions. The air/fuel ratio might vary from about 60:1 to 120:1 for simple cycle gas turbines and from 100:1 to 200:1 if a heat-exchanger is used. Considering that the stoichiometric ratio is approximately 15:1, it is clear that a high dilution is required to maintain the temperature level dictated by turbine stresses

• The formation of carbon deposits ('coking') must be avoided, particularly the hard brittle variety. Small particles carried into the turbine in the high-velocity gas stream can erode the blades and block cooling air passages; furthermore, aerodynamically excited vibration in the combustion chamber might cause sizeable pieces of carbon to break free resulting in even worse damage to the turbine.

• In aircraft gas turbines, combustion must be stable over a wide range of chamber pressure because of the substantial change in this parameter with a altitude and forward speed. Another important requirement is the capability of relighting at high altitude in the event of an engine flame-out.

• Avoidance of smoke in the exhaust is of major importance for all types of gas turbine; early jet engines had very smoky exhausts, and this became a serious problem around airports when jet transport aircraft started to operate in large numbers. Smoke trails in flight were a problem for military aircraft, permitting them to be seen from a great distance. Stationary gas turbines are now found in urban locations, sometimes close to residential areas.

• Although gas turbine combustion systems operate at extremely high efficiencies, they produce pollutants such as oxides of nitrogen , carbon monoxide (CO) and unburned hydrocarbons (UHC) and these must be controlled to very low levels. Over the years, the performance of the gas turbine has been improved mainly by increasing the compressor pressure ratio and turbine inlet temperature (TIT). Unfortunately this results in increased production of . Ever more stringent emissions legislation has led to significant changes in combustor design to cope with the problem.

Probably the only feature of the gas turbine that eases the combustion designer's problem is the peculiar interdependence of compressor delivery air density and mass flow which leads to the velocity of the air at entry to the combustion system being reasonably constant over the operating range.

For aircraft applications there are the additional limitations of small space and low weight, which are, however, slightly offset by somewhat shorter endurance requirements. Aircraft engine combustion chambers are normally constructed of light-gauge, heat-resisting alloy sheet (approx. 0.8 mm thick), but are only expected to have a life of some 10000 hours. Combustion chambers for industrial gas turbine plant may be constructed on much sturdier lines but, on the other hand, a life of about 100000 hours is required. Refractory linings are sometimes used in heavy chambers, although the remarks made above regarding the effects of hard carbon deposits breaking free apply with even greater force to refractory material.

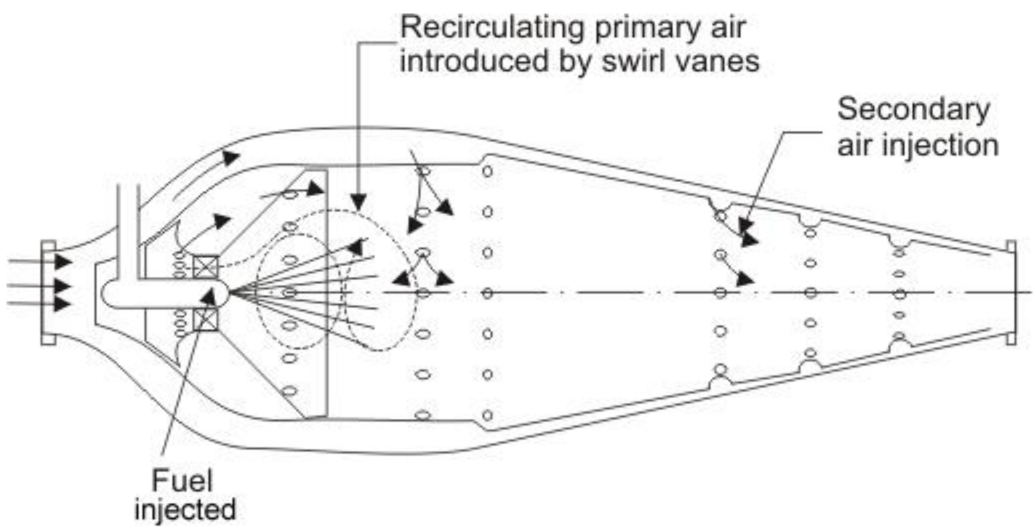


Figure 16.1 Combustion chamber with swirl vanes

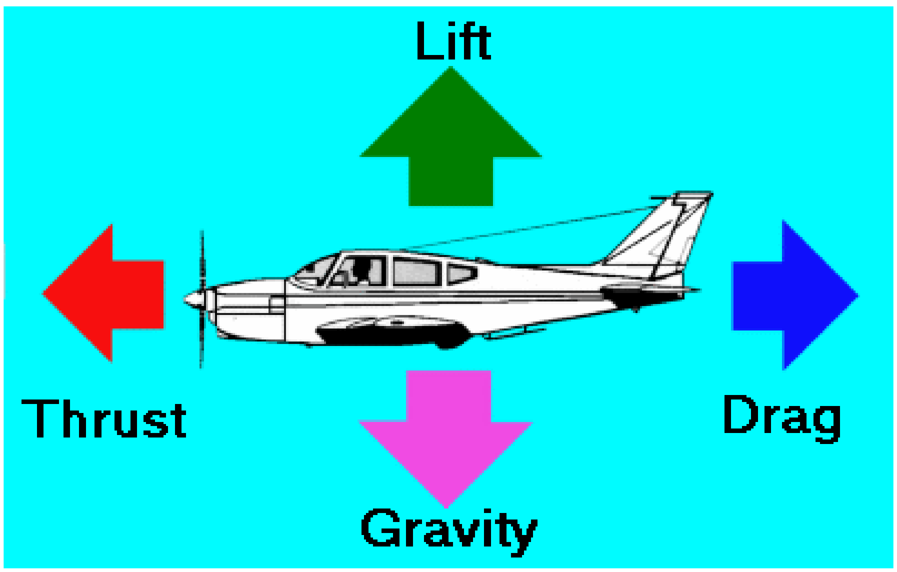
Figure 16.1 indicates the schematic of a combustion chamber. The primary air is introduced through twisted radial vanes known as 'swirl vanes', that results in a vortex motion with a low-pressure region along the axis of the chamber. The fuel is injected in the same direction of air. The vortex motion is some time enhanced by injecting the secondary air through short tangential chutes in the flame tube. The burning gases tends to flow towards the region of low pressure and some portion of them swept round towards the jet of fuel as indicated by the arrow. The objective is to obtain a stable flame.

V -UNIT

Jet Propulsion

A jet engine is a reaction engine that discharges a fast moving jet which generates thrust by jet propulsion in accordance with Newton's laws of motion. This broad definition of jet engines includes turbojets, turbofans, rockets, ramjets, and pulse jets. jet engine is nothing but a Gas Turbine.

HOW DOES AN AEROPLANE FLY



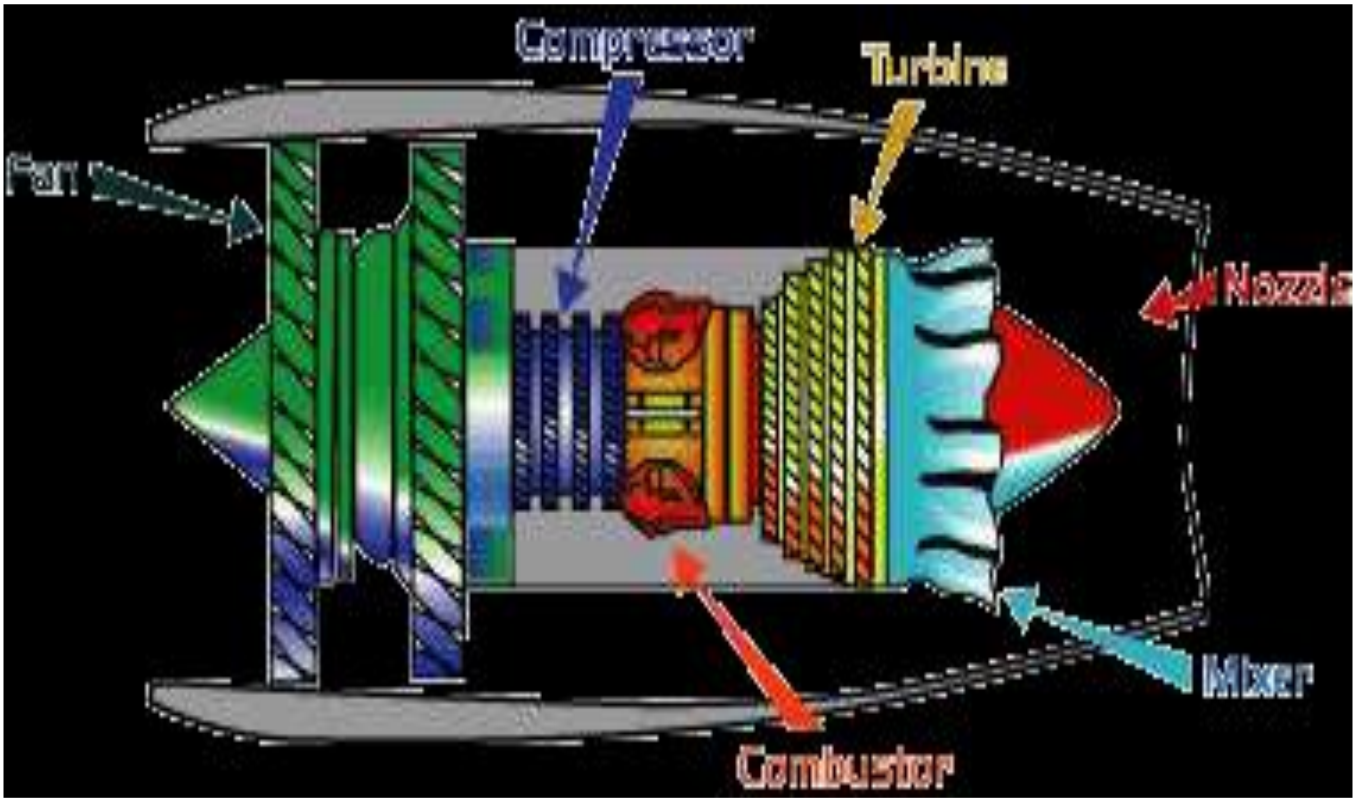
PRINCIPLE OF JET ENGINE :

• Principle of jet engine is based on Newton’s second and third law of motion.

• Second law states that the rate of change of momentum in any direction is proportional to the force acting in that direction.

• Third law states that for every action there is an equal and opposite reaction.

WORKING OF JET ENGINE



Sucks in air from front with fan

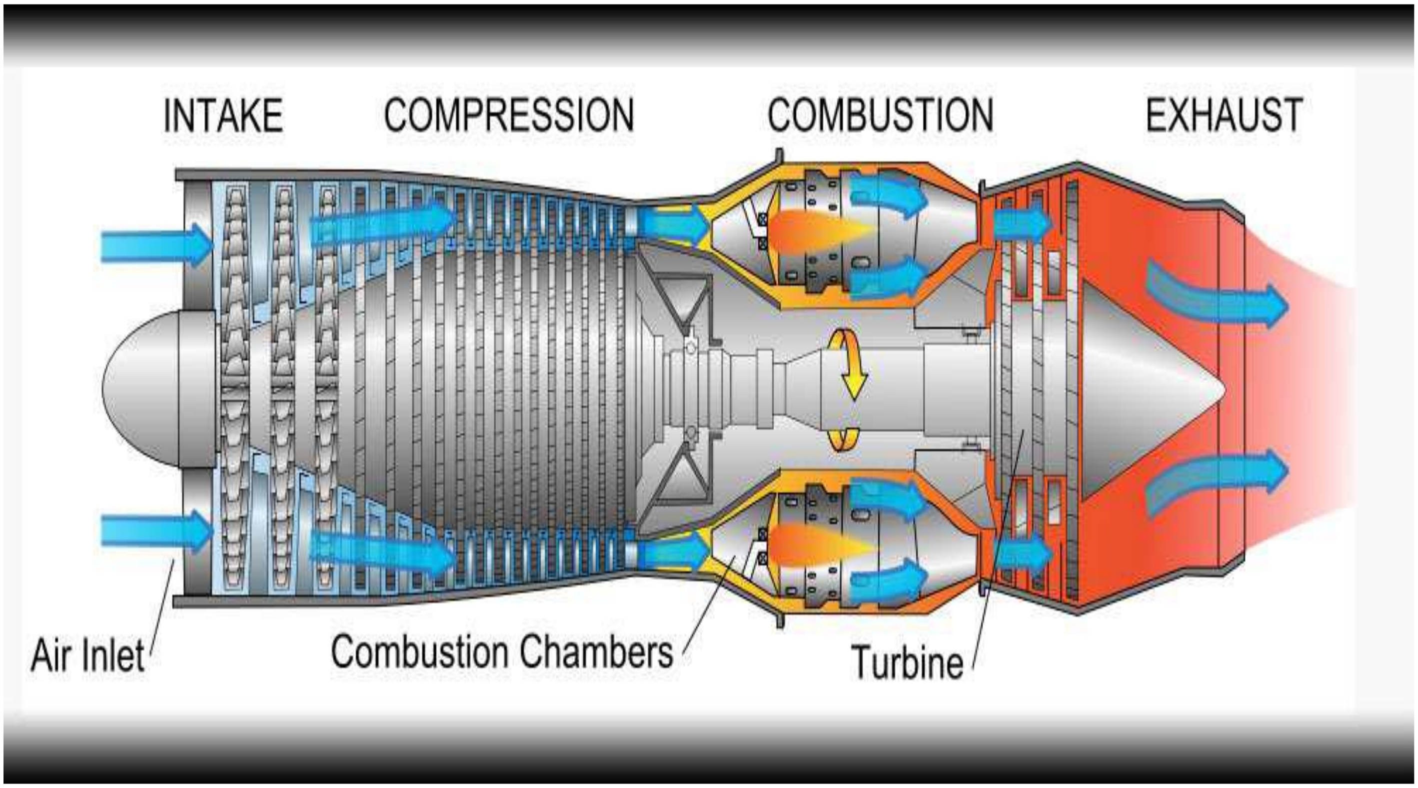
• A compressor raises the pressure of the air

• Then the compressed air is ignited

• Gas expends and comes out nozzle

• Engine/Aircraft thrusts forward

PARTS OF JET ENGINE



• Air intake (Fan)

• Compressor

• Combustion chamber

• Turbine

• Nozzle

Aircraft Engines and Propulsion System

The modern aircraft engine have the ability to actuate massive airstream and thus to produce high thrust. The engine airflow rate is perhaps 50 times the fuel flow rate, and the term air breathing engine is quite appropriate. Thus, a continuous stream of air flows through the air-breathing engine. The air is compressed, mixed with fuel, ignited, expanded through a turbine and then expelled as the exhaust gas.

The following four types of aircraft engines are generally used

• Turbojet Engine

• Turboprop Engine

• Turbofan Engine

• Ramjet Engine

At low speeds, propeller propulsion is more efficient than jet propulsion. Conventional propellers, however, become inefficient and noisy at flight speeds higher than 0.5 or 0.6 times the speed of sound. In contrast, turbojet and turbofan engines can function efficiently and quietly at flight speeds as high as 0.85 times the speed of sound. Turbojets can also operate at supersonic flight speeds. Ramjet, which is the simplest of all air-breathing engines can operate at a higher speed than turbojet engines and is mostly suitable for supersonic flight.

1. Turbojet Engine

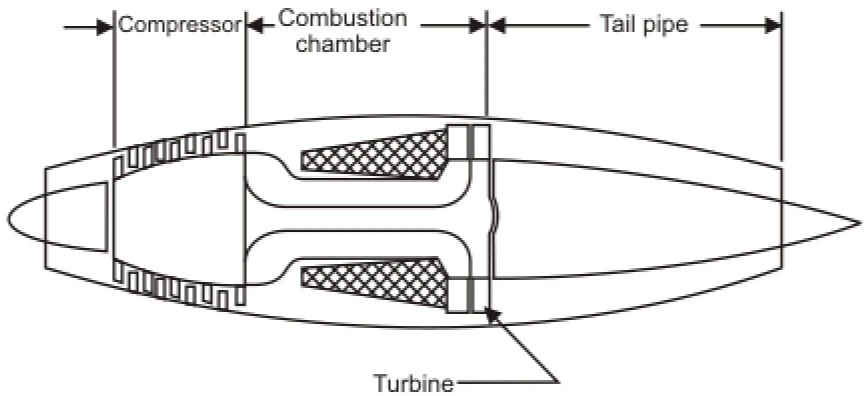


Figure 16.2 Turbojet

The turbojet engine consists of a gas turbine, the output of which is used solely to provide power to the compressor. The compressor and the turbine are normally mounted on common shaft. Air is taken into the engine through an approximate diffuser duct, passes through the compressor and enters the combustions chamber, where it is mixed and burned with fuel.

Most common fuels are hydrocarbons (Aviation kerosene). The ratio of fuel to air is determined by the maximum allowable gas temperature permitted by the turbine. Normally, a considerable excess air is used. The hot high pressure gases are then expanded through the turbine to a pressure which is higher than the ambient atmosphere, and yet sufficiently lower than the combustion chamber pressure, to produce just enough power in the turbine to run the compressor. After leaving the turbine, the gas is expanded to the ambient pressure through an appropriate nozzle. As this occurs, the gas is accelerated to a velocity, which is greater than the incoming velocity of the ingested air, and therefore produces a propulsive thrust.

3. Turbofan Engine

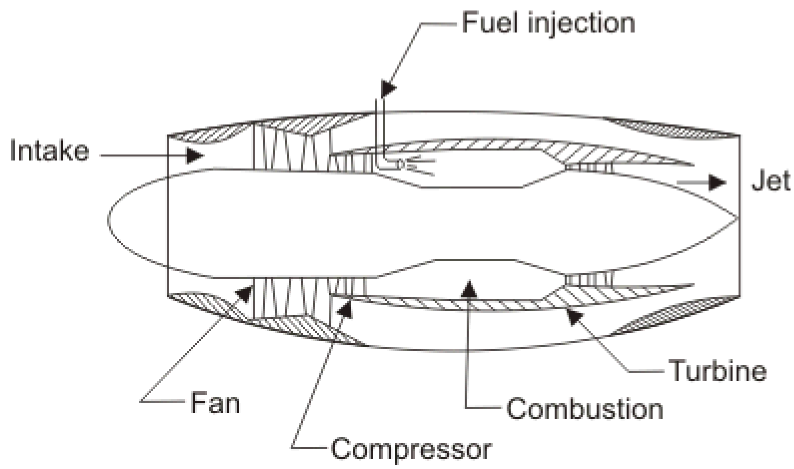


Figure 17.2 Turbofan

This is another variety of gas-turbine aircraft engine. This is very similar to the turboprop in principle, except that a fan is used instead of a propeller, and this fan is contained within a duct as shown in the above figure. Most airliners use modern turbofan engines because of their high thrust and good fuel efficiency. Figure below shows the picture of Boeing 747 aircraft that uses a turbofan engine. In a turbofan engine , the air is sucked by the engine inlet. Some of the incoming air passes through the fan and continues on into the core compressor and then the burner, where it is mixed with fuel and combustion occurs. The hot exhaust passes through the core and fan turbines and then out the nozzle, as in a basic turbojet. The rest of the incoming air passes through the fan and bypasses, or goes around the engine, just like the air through a propeller. The air that goes through the fan has a velocity that is slightly increased from free stream. So a turbofan gets some of its thrust from the core and some of its thrust from the fan. The ratio of the air that goes around the engine to the air that goes through the core is called the **bypass ratio**. Because the fuel flow rate for the core is changed only a small amount by the addition of the fan, a turbofan generates more thrust for nearly the same amount of fuel used by the core. This means that a turbofan is very fuel efficient. In fact, high bypass ratio turbofans are nearly as fuel efficient as turboprops.

Boeing 747:Typical example of a Turbofan Engine

4. Ramjet Engine

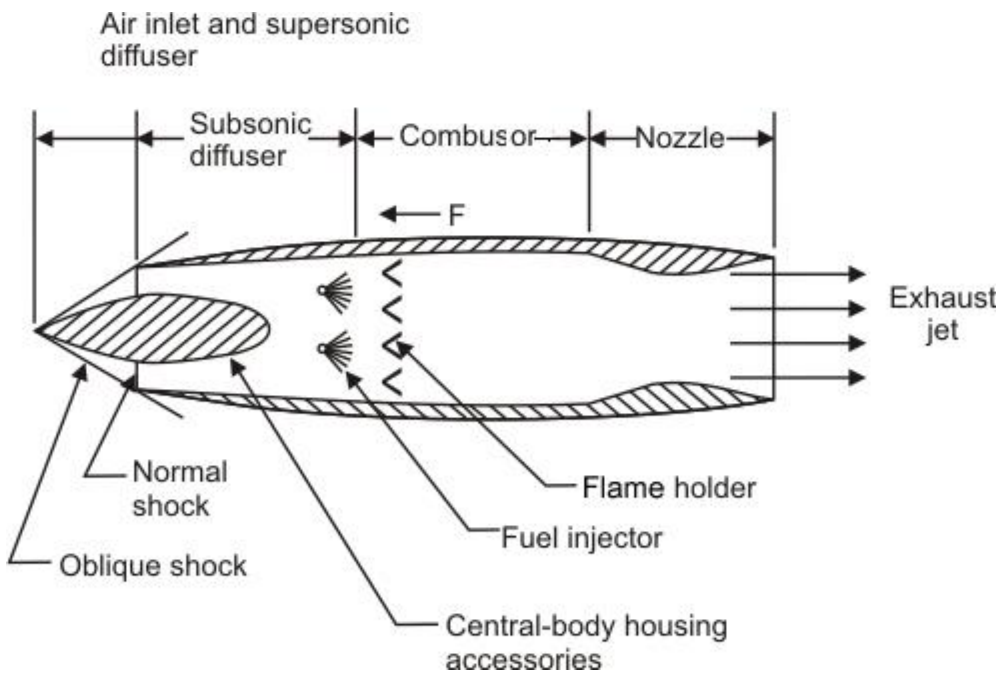


Figure 17.3 Ramjet

At higher forward speeds, the ram pressure of the air is already very large, and the necessity for a compressor tends to disappear. A turbojet engine minus the compressor and turbine, but with a combustion chamber, is known as a ramjet engine. Such engines simply consists of

1. A duct designed to diffuse the incoming air, slowing its velocity and raising its pressure
2. A combustor, designed to heat the air, normally by combustion with a liquid fuel
3. A nozzle, designed to expand and accelerate the heated gases rearwards

The ramjet engine does not accelerate itself from a standing start but requires some other form of propulsion, usually a rocket, to accelerate it to near its operating speed.

Rocket engine

A rocket engine is a type of jet engine that uses only stored rocket propellant mass for forming its high-speed propulsive jet. Rocket engines are reaction engines, obtaining thrust in accordance with Newton's third law. Most rocket engines are internal combustion engines, although non-combusting forms (such as cold gas thrusters) also exist. Vehicles propelled by rocket engines are commonly called rockets. Since they need no external material to form their jet, rocket engines can perform in a vacuum and thus can be used to propel spacecraft and ballistic missiles.

Compared to other types of jet engines, rocket engines are by far the lightest, and have the highest thrust, but are the least propellant-efficient (they have the lowest specific impulse). The ideal exhaust is hydrogen, the lightest of all gases, but chemical rockets produce a mix of heavier species, reducing the exhaust velocity. Rocket engines become more efficient at high velocities, due to greater propulsive efficiency and the Oberth effect. Since they do not require an atmosphere, they are well suited for uses at very high altitudes and in space.

Principle of operation

Rocket engines produce thrust by the expulsion of an exhaust fluid which has been accelerated to a high speed through a propelling nozzle. The fluid is usually a gas created by high pressure (10 to 200 bar)) combustion of solid or liquid propellants, consisting of fuel and oxidiser components, within a combustion chamber. The nozzle uses the heat energy released by expansion of the gas to accelerate the exhaust to very high (supersonic) speed, and the reaction to this pushes the engine in the opposite direction. Combustion is most frequently used for practical rockets, as high temperatures and pressures are desirable for the best performance, permitting a longer nozzle, giving higher exhaust speeds and better thermodynamic efficiency.

An alternative to combustion is the water rocket, which uses water pressurised by compressed air, carbon dioxide, nitrogen, or manual pumping, for model rocketry.

Propellant

Rocket propellant is mass that is stored, usually in some form of propellant tank, or within the combustion chamber itself, prior to being ejected from a rocket engine in the form of a fluid jet to produce thrust.

Chemical rocket propellants are most commonly used, which undergo exothermic chemical reactions which produce hot gas which is used by a rocket for propulsive purposes. Alternatively, a chemically inert reaction mass can be heated using a high-energy power source via a heat exchanger, and then no combustion chamber is used.

A solid rocket motor.

Solid rocket propellants are prepared as a mixture of fuel and oxidising components called 'grain' and the propellant storage casing effectively becomes the combustion chamber.

Propellant efficiency

For a rocket engine to be propellant efficient, it is important that the maximum pressures possible be created on the walls of the chamber and nozzle by a specific amount of propellant; as this is the source of the thrust. This can be achieved by all of: heating the propellant to as high a temperature as possible (using a high energy fuel, containing hydrogen and carbon and sometimes metals such as aluminium, or even using nuclear energy) using a low specific density gas (as hydrogen rich as possible) using propellants which are, or decompose to, simple molecules with few degrees of freedom to maximise translational velocity

Since all of these things minimise the mass of the propellant used, and since pressure is proportional to the mass of propellant present to be accelerated as it pushes on the engine, and since from Newton's third law the pressure that acts on the engine also reciprocally acts on the propellant, it turns out that for any given engine, the speed that the propellant leaves the chamber is unaffected by the chamber pressure (although the thrust is proportional). However, speed is significantly affected by all three of the above factors and the exhaust speed is an excellent measure of the engine propellant efficiency. This is termed exhaust velocity, and after allowance is made for factors that can reduce it, the effective exhaust velocity is one of the most important parameters of a rocket engine (although weight, cost, ease of manufacture etc. are usually also very important.

For aerodynamic reasons the flow goes sonic ("chokes") at the narrowest part of the nozzle, the 'throat'. Since the speed of sound in gases increases with the square root of temperature, the use of hot exhaust gas greatly improves performance. By comparison, at room temperature the speed of sound in air is about 340 m/s while the speed of sound in the hot gas of a rocket engine can be over 1700 m/s; much of this performance is due to the higher temperature, but additionally rocket propellants are chosen to be of low molecular mass, and this also gives a higher velocity compared to air.

Expansion in the rocket nozzle then further multiplies the speed, typically between 1.5 and 2 times, giving a highly collimated hypersonic exhaust jet. The speed increase of a rocket nozzle is mostly determined by its area expansion ratio—the ratio of the area of the throat to the area at the exit, but detailed properties of the gas are also important. Larger ratio nozzles are more massive but are able to extract more heat from the combustion gases, increasing the exhaust velocity.

Net thrust

Below is an approximate equation for calculating the net thrust of a rocket engine:

Since, unlike a jet engine, a conventional rocket motor lacks an air intake, there is no 'ram drag' to deduct from the gross thrust. Consequently, the net thrust of a rocket motor is equal to the gross thrust (apart from static back pressure).

At full throttle, the net thrust of a rocket motor improves slightly with increasing altitude, because as atmospheric pressure decreases with altitude, the pressure thrust term increases. At the surface of the Earth the pressure thrust may be reduced by up to 30%,depending on the engine design. This reduction drops roughly exponentially to zero with increasing altitude.

Rocket Propulsion:

Bernouilli's theorem can be applied to find out the velocity of exhaust gas emerging from a rocket and thrust exerted by the exhaust gas on the rocket. At the bottom of the fuel chamber of a rocket, there is an orifice (small opening) through which the exhaust gas leaves due to burning of fuel in the chamber.

Let P1, A1 and v1 be the pressure, area of cross section and velocity of burnt fuel gas inside the chamber and the corresponding values at the orifice is P2, A2 and v2.

According to Bernoullis theorem:

$$P_1 + \frac{1}{2}\rho v_1^2 + \rho g y_1 = P_2 + \frac{1}{2}\rho v_2^2 + \rho g y_2$$

.... (i)

In gas, the density is low and the pressure due to elevation, $\rho g(y_1 - y_2)$ would be negligible compared to other terms.

$$P_2 - P_1 = \frac{1}{2}\rho\left(v_1^2 - v_2^2\right)$$

$$v_2^2 = -\frac{2}{\rho}(P_2 - P_1) + v_1^2$$

$$v_2^2 = \frac{2}{\rho}(P - P_a) + v_1^2$$

where $P_1 = P$, the pressure inside the chamber, and $P_2 = P_a$ atmospheric pressure.

As per the continuity equation

$$A_1 v_1 = A_2 v_2$$

$$v_1 = \frac{A_2}{A_1} v_2$$

Since the area of cross section of orifice is very small compared to that of fuel chamber, $A_2 << A_1$, the velocity $v_1$would be negligible.

So, equation (ii) can be written as,

$$v_2 = \sqrt{\frac{2(P - P_a)}{\rho}}$$

Equation (iv) gives the speed of the efflux.

The thrust experienced by the rocket due to exhaust

$$= v_2 \frac{dM}{dt}$$

M = the mass of exhaust gas over the time interval dt seconds.

So, the thrust,

$$v_2 \frac{dM}{dt} = \rho A_2 v_2^2 = 2A_2 (P - P_a)$$

Thus the thrust experienced by the rocket can be calculated based on Bernoulli's theorem.

ROCKET PROPELLANTS

Introduction

Liquids

Solids

Hybrids

Propellant is the chemical mixture burned to produce thrust in rockets and consists of a fuel and an oxidizer. A fuel is a substance which burns when combined with oxygen producing gas for propulsion. An oxidizer is an agent that releases oxygen for combination with a fuel. Propellants are classified according to their state - liquid, solid, or hybrid.

The gauge for rating the efficiency of rocket propellants is specific impulse, stated in seconds. Specific impulse indicates how many pounds (or kilograms) of thrust are obtained by the consumption of one pound (or kilogram) of propellant in one second. Specific impulse is characteristic of the type of propellant, however, its exact value will vary to some extent with the operating conditions and design of the rocket engine.

Liquid Propellants

In a liquid propellant rocket, the fuel and oxidizer are stored in separate tanks, and are fed through a system of pipes, valves, and turbopumps to a combustion chamber where they are combined and burned to produce thrust. Liquid propellant engines are more complex then their solid propellant counterparts, however, they offer several advantages. By controlling the flow of propellant to the combustion chamber, the engine can be throttled, stopped, or restarted.

A good liquid propellant is one with a high specific impulse or, stated another way, one with a high speed of exhaust gas ejection. This implies a high combustion temperature and exhaust gases with small molecular weights. However, there is another important factor which must be taken into consideration: the density of the propellant. Using low density propellants means that larger storage tanks will be required, thus increasing the mass of the launch vehicle. Storage temperature is also important. A propellant with a low storage temperature, i.e. a cryogenic, will require thermal insulation, thus further increasing the mass of the launcher. The toxicity of the propellant is likewise important. Safety hazards exist when handling, transporting, and storing highly toxic compounds. Also, some propellants are very corrosive, however, materials that are resistant to certain propellants have been identified for use in rocket construction.

Liquid propellants used by NASA and in commercial launch vehicles can be classified into three types: petroleum, cryogenics, and hypergolics.

Petroleum fuels are those refined from crude oil and are a mixture of complex hydrocarbons, i.e. organic compounds containing only carbon and hydrogen. The petroleum used as rocket fuel is kerosene, or a type of highly refined kerosene called RP-1 (refined petroleum). It is used in combination with liquid oxygen as the oxidizer.

RP-1 and liquid oxygen are used as the propellant in the first-stage boosters of the Atlas/Centaur and Delta launch vehicles. It also powered the first-stages of the Saturn 1B and Saturn V rockets. RP-1 delivers a specific impulse considerably less than cryogenic fuels.

Cryogenic propellants are liquefied gases stored at very low temperatures, namely liquid hydrogen (LH2) as the fuel and liquid oxygen (LO2) as the oxidizer. LH2 remains liquid at temperatures of -423 degrees F (-253 degrees C) and LO2 remains in a liquid state at temperatures of -298 degrees F (-183 degrees C).

Because of the low temperatures of cryogenic propellants, they are difficult to store over long periods of time. For this reason, they are less desirable for use in military rockets which must be kept launch ready for months at a time. Also, liquid hydrogen has a very low density (0.59 pounds per gallon) and, therefore, requires a storage volume many times greater than other fuels. Despite these drawbacks, the high efficiency of liquid hydrogen/liquid oxygen makes these problems worth coping with when reaction time and storability are not too critical. Liquid hydrogen delivers a specific impulse about 40% higher than other rocket fuels.

Liquid hydrogen and liquid oxygen are used as the propellant in the high efficiency main engines of the space shuttle. LH2/LO2 also powered the upper stages of the Saturn V and Saturn lB rockets as well as the second stage of the Atlas/Centaur launch vehicle, the United States' first LH2/LO2 rocket

Hypergolic propellants are fuels and oxidizers which ignite spontaneously on contact with each other and require no ignition source. The easy start and restart capability of hypergolics make them ideal for spacecraft maneuvering systems. Also, since hypergolics remain liquid at normal temperatures, they do not pose the storage problems of cryogenic propellants. Hypergolics are highly toxic and must be handled with extreme care.

Hypergolic fuels commonly include hydrazine, monomethyl hydrazine (MMH) and unsymmetrical dimethyl hydrazine (UDMH). The oxidizer is typically nitrogen tetroxide (N2O4) or nitric acid (HNO3). UDMH is used in many Russian, European, and Chinese rockets while MMH is used in the orbital maneuvering system (OMS) and reaction control system (RCS) of the Space Shuttle orbiter. The Titan family of launch vehicles and the second stage of the Delta use a fuel called Aerozine 50, a mixture of 50% UDMH and 50% hydrazine.

Hydrazine is also frequently used as a mono-propellant in catalytic decomposition engines . In these engines, a liquid fuel decomposes into hot gas in the presence of a catalyst. The decomposition of hydrazine produces temperatures of about 1700 degrees F and a specific impulse of about 230 or 240 seconds.

Solid Propellants

Solid propellant motors are the simplest of all rocket designs. They consist of a casing, usually steel, filled with a mixture of solid compounds (fuel and oxidizer) which burn at a rapid rate, expelling hot gases from a nozzle to produce thrust. When ignited, a solid propellant burns from the center out towards the sides of the casing. The shape of the center channel determines the rate and pattern of the burn, thus providing a means to control thrust. Unlike liquid propellant engines, solid propellant motors can not be shut down. Once ignited, they will burn until all the propellant is exhausted.

There are two families of solids propellants: homogeneous and composite. Both types are dense, stable at ordinary temperatures, and easily storable.

Homogeneous propellants are either simple base or double base. A simple base propellant consists of a single compound, usually nitrocellulose, which has both an oxidation capacity and a reduction capacity. Double base propellants usually consist of nitrocellulose and nitroglycerine, to which a plasticiser is added. Homogeneous propellants do not usually have specific impulses greater than about 210 seconds under normal conditions. Their main asset is that they do not produce traceable fumes and are, therefore, commonly used in tactical weapons. They are also often used to perform subsidiary functions such as jettisoning spent parts or separating one stage from another.

Modern composite propellants are heterogeneous powders (mixtures) which use a crystallized or finely ground mineral salt as an oxidizer, often ammonium perchlorate, which constitutes between 60% and 90% of the mass of the propellant. The fuel itself is aluminum. The propellant is held together by a polymeric binder, usually polyurethane or polybutadienes. Additional compounds are sometimes included, such as a catalyst to help increase the burning rate, or other agents to make the powder easier to manufacture. The final product is rubberlike substance with the consistency of a hard rubber eraser.

Solid propellant motors have a variety of uses. Small solids often power the final stage of a launch vehicle, or attach to payloads to boost them to higher orbits. Medium solids such as the Payload Assist Module (PAM) and the Inertial Upper Stage (IUS) provide the added boost to place satellites into geosynchronous orbit or on planetary trajectories.

The Titan, Delta, and Space Shuttle launch vehicles use strap-on solid propellant rockets to provide added thrust at liftoff. The Space Shuttle uses the largest solid rocket motors ever built and flown. Each booster contains 1,100,000 pounds (499,000 kg) of propellant and can produce up to 3,300,000 pounds (14,680,000 Newtons) of thrust.

Hybrid Propellants

Hybrid propellant engines represent an intermediate group between solid and liquid propellant engines. One of the substances is solid, usually the fuel, while the other, usually the oxidizer, is liquid. The liquid is injected into the solid, whose fuel reservoir also serves as the combustion chamber. The main advantage of such engines is that they have high performance, similar to that of solid propellants, but the combustion can be moderated, stopped, or even restarted. It is difficult to make use of this concept for vary large thrusts, and thus, hybrid propellant engines are rarely built.