22
Cycles for Steam Power Plants

22.1. Rankine cycle and its Analysis. 22.2. Reheat cycle. 22.3. Regenerative cycle. 22.4. Reheat-Regenerative cycle. 22.5. Binary vapour cycle. 22.6. Superposed or topping cycle.

22.1. RANKINE CYCLE AND ITS ANALYSIS

The line diagram of the power plant working on Rankine cycle is shown in Fig. 22.1 (*a*). The Rankine cycle used for steam power plant is shown in Fig. 22.1 (*b*) and Fig. 22.1 (*c*) on *p-v* and *T-s* diagrams.

The different processes of the Rankine cycle are described below :

1. The point '*d*' represents the water at condenser pressure p_2 and corresponding saturation temperature T_2. The process '*de*' represents the adiabatic compression of water by the pump from pressure p_2 (condenser pressure) to pressure p_1 (boiler pressure). There is slight rise in temperature of water during the compression process '*de*'.

2. During the processes '*ea*' and '*ab*', heat is supplied by the boiler to the water to convert into steam. The process '*ea*' represents the supply of heat at constant pressure till the saturation temperature of water

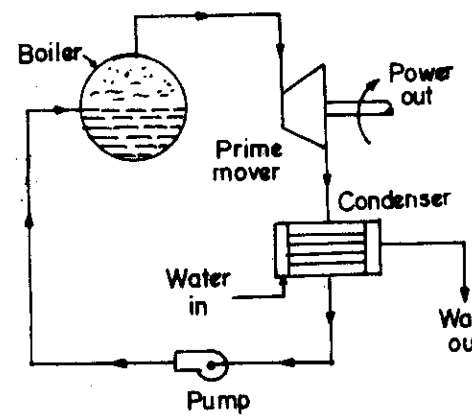


Fig. 22.1 (*a*) The arrangement of the components used for steam power plant working on Rankine cycle.

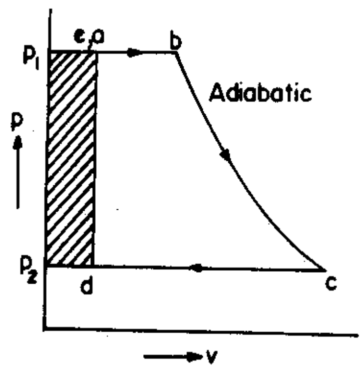


Fig. 22.1 (*b*) Rankine cycle on *p-v* diagram.

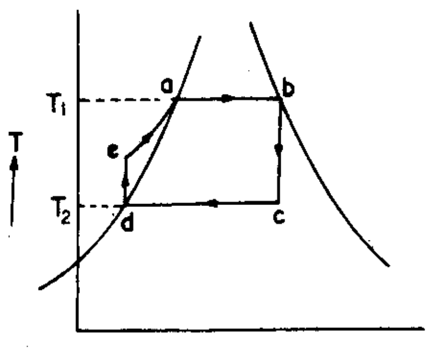


Fig. 22.1 (*c*) Rankine cycle on *T-s* diagram.

is reached corresponding to boiler pressure p_1. The points *e* and *a* are same on *pv* diagram as increase in volume of water during this heating process is negligible. The process '*ab*' represents the addition of heat to the water at constant pressure till the water completely converts into steam. The final condition of steam (*b*) may be wet, dry saturated or superheated depending upon the quantity of heat supplied by the boiler.

3. The process '*bc*' represents the isentropic expansion of steam in the prime mover as shown in Fig. 22.1 (*c*). During this expansion, external work is developed and the pressure of steam falls from p_1 to p_2 and its temperature will be T_2.

4. The process '*cd*' represents the condensation of steam coming out from the prime mover in the condenser. During the condensation of steam, the pressure is constant and there is only change of phase

from steam to water as the latent heat of steam ($x_2\ h_{fg2}$) is carried by circulating water in the condenser.

5. Again the process '*de*' represents the adiabatic compression of water by the pump from the pressure p_2 to p_1 and the cycle is repeated.

Let h_b = Enthalpy of steam per kg at point '*b*'

h_e = Enthalpy of steam per kg at point '*c*'

v_w = Specific volume of water at point *d* or *e* as there is much change in specific volume during this process

h_{fe} = Enthalpy of water per kg at point '*e*'

h_{fa} = Enthalpy of water per kg at point '*a*'

h_{fd} = Enthalpy of water per kg at point '*d*'

Total heat supplied by the boiler per kg of steam generated

$$= h_b - h_{fe} = h_b - (h_{fd} + W_p)$$

where W_p is the work done by the pump per kg of water supplied.

Work done per kg of steam in the prime mover

$$= h_b - h_c$$

Work done by the pump per kg of water supplied to the boiler

$$W_p = [v_{f2}\ (p_1 - p_2)] \text{ J/kg where } p \text{ is in N/m}^2$$

where v_{f2} is the specific volume of saturated water at pressure p_2.

$\therefore$ Net work available per kg of water

$$= (h_b - h_c) - v_{fa}\frac{(p_1 - p_2)}{1000}$$

$$= (h_b - h_c) - W_p.$$

The Rankine efficiency of the cycle is given by

$$\eta_p = \frac{\text{Net work available}}{\text{Heat supplied}}$$

$$= \frac{(h_b - h_c) - W_p}{h_b - (h_{fd} + W_p)} \qquad ...(22.1)$$

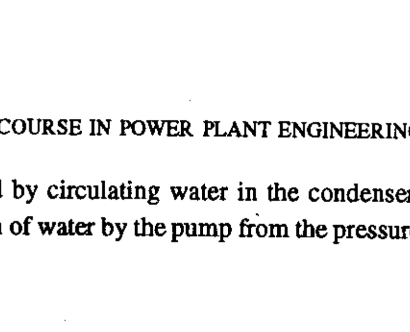


Fig. 22.2. Rankine cycle neglecting pump work.

The pump work, 'W_p' is always neglected for all practical purposes as it is very small compared with other heat quantities.

$$\eta_r = \frac{h_b - h_c}{h_b - h_{fd}} = \frac{h_1 - h_2}{h_1 - h_{f2}} \text{ as } W_p \approx 0 \qquad ...(22.2)$$

where h_1 = enthalpy of steam of point '*b*' at pressure p_1.

h_2 = enthalpy of steam at point '*c*' at pressure p_2.

h_{f2} = enthalpy of water at point '*d*' at pressure p_2.

The Rankine cycle neglecting pump work is represented on *T-s* diagram as shown in Fig. 22.2.

22.2. REHEAT CYCLE

The efficiency of the ordinary Rankine cycle can be improved by increasing the pressure and temperature of the steam entering into the turbine. As the initial pressure increases, the expansion ratio in the turbine also increases and the steam becomes quite wet at the end of expansion. This is not desirable because the increased moisture content of the steam causes corrosion of the turbine blades and increases the losses. This reduces the nozzle and blade efficiency.

In reheat cycle, the steam is extracted from a suitable point in the turbine and is reheated with the help of the flue gases in the boiler furnace as shown in Fig. 22.3. The main purpose of reheating is to increase the dryness friction of steam passing through the lower stages of the turbine. The dryness fraction of steam coming out of turbine should not fall below 0.88. By using the reheat cycle, the specific steam consumption decreases and thermal efficiency also increases. The increase in thermal efficiency due to reheat depends upon the ratio of reheat pressure to original pressure of steam. The reheat pressure is generally kept within 20% of the initial pressure of the steam. The efficiency of the reheat cycle may be less than the Rankine efficiency if the reheat is used at low pressure.

The main advantage of reheat cycle is to reduce the specific steam consumption and consequently reduce the size of the boiler and auxiliaries for the same output.

The reheat cycle is only preferred for high capacity plants (above 100 MW and when pressure of the steam is as high as 100 bar. Only one stage reheating is generally used in practice.

It is not preferred for low capacity plants as the cost of the reheater is not justified.

Referring to Figs. 22.3 and 22.4, the total heat supplied and work done are given by

$$h_s = (h_1 - h_{f6}) + (h_3 - h_2)$$

and

$$W = (h_1 - h_2) + (h_3 - h_4) - (h_{f6} - h_{f5})$$

where $(h_{f6} - h_{f5})$ is the pump work.

Therefore, the efficiency of the reheat cycle is given by,

$$(\eta_{reheat}) = \frac{W}{h_s} = \frac{(h_1 - h_2) + (h_3 - h_4) - (h_{f6} - h_{f5})}{(h_1 - h_{f6}) + (h_3 - h_2)} \quad ...(22.3)$$

If the pump work is neglected,

$$(\eta_{reheat}) = \frac{(h_1 - h_2) + (h_3 - h_4)}{(h_1 - h_{f5}) + (h_3 - h_2)} \text{ as } h_{f6} = h_{f5} \quad ...(22.4)$$

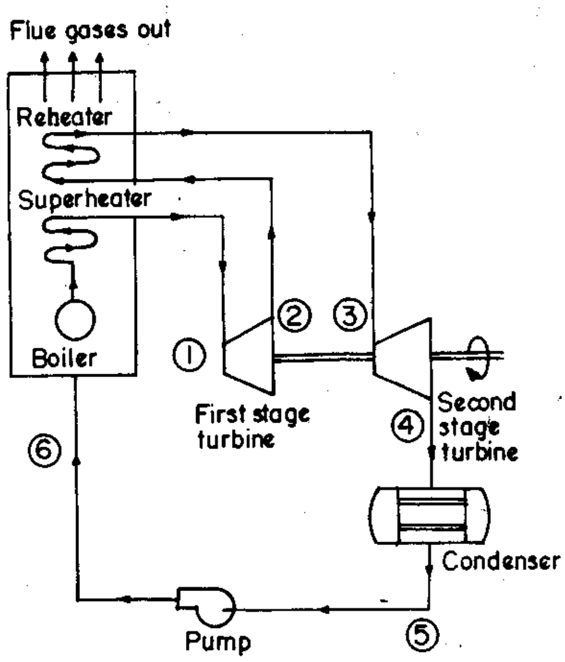


Fig. 22.3.

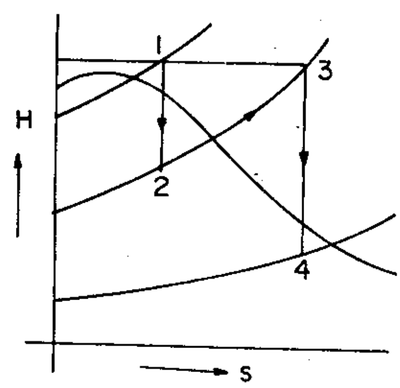


Fig. 22.4. Reheat Cycle.

Advantages and Disadvantages of Reheat Cycle. The advantages of reheat cycle over simple Rankine-cycle for turbine-generator having the same rating and the same steam condition at stop-valve are listed below.

Advantages : 1. There is a limit to the degree of superheat due to metallurgical conditions, therefore, it is not possible to get all superheat in one stage. The inevitable effect of use of higher pressure in modern power plants is that, the saturation line is reached earlier during isentropic expansion as shown in Fig. 22.5, and, therefore, most of the turbine stages operate in saturated region which is highly undesirable. There is heavy blade erosion due to the impact of water particles carried with the steam. Therefore, the reheating is essential in high pressure modern power plants to increase the life of the plant.

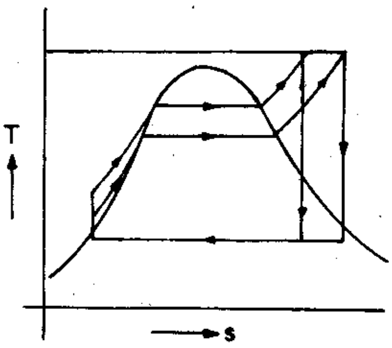


Fig. 22.5. Rankine cycle with increasing boiler pressure.

2. The reheating reduces 4 to 5% fuel consumption with a corresponding reduction in fuel handling.

3. The reheat cycle reduces the steam flow of 15 to 20% with corresponding reductions in boiler, turbine and feed heating equipments capacities. This also reduces the pumping power in that proportion.

4. The wetness of the exhaust steam with reheat cycle is reduced to 50% of Rankine cycle with a corresponding reduction in exhaust blade erosion.

5. Lower steam pressures and temperatures and less costly materials can be used to obtain the required thermal performance.

6. A reduction in steam volume and heat to the condenser is reduced by 7 to 8%. Therefore, the condenser size and cooling water requirement are also reduced by the same percentages.

7. The size of the L.P. turbine blades is reduced because specific volume is reduced by 7 to 8%.

The advantages claimed for the reheat cycle are higher thermal efficiency, reduced feed pump power, smaller condenser, smaller boiler, long life of turbine and less handling of fuel and firing equipments.

Disadvantages : 1. The cost of extra pipes and equipments and controls makes this cycle more expensive than ordinary Rankine cycle. Therefore, the minimum capacity of the plant must be 10 MW for the adoption of reheat cycle.

2. The greater floor space is required to accommodate the longer turbine and reheat piping.

3. The turbine blades of the second stage immediately after the reheater is considerably longer than the corresponding pressure stages of non-reheat set. Therefore, additional design problems are involved.

4. The complexity of operation and control increases with the adoption of reheat cycle.

5. At light loads, the steam passing through the last blade rows to the condenser are seriously superheated if the same reheat is maintained. Feed water is sometimes sprayed into the low pressure cylinders as low steam flows as a precaution against overheating of blades.

The above disadvantages are compensated in new installations by the saving in cost of the equipments for coal storage, coal handling, fuel burning, fans and dust collectors, ash disposal, boiler house structure and auxiliary electrical equipments.

Methods of Reheating. The various methods of reheating used in practice are discussed below :

(1) **Gas Reheating.** The gas reheating system is shown in Fig. 22.6 (*a*).

The steam extracted from H.P. turbine is piped back to reheater, arranged in a boiler as shown in figure. The steam is normally reheated to its initial throttle temperature.

Dr. De Smaele gives general rule to extract the steam at a pressure to reheat which gives best results. The rule states, "For one gas reheating to the throttle temperature, the best thermal efficiency of the cycle is obtained by reheating at a pressure attained when the temperature of the steam has become one-half of the throttle temperature provided the steam at the beginning of reheating is still superheated by about 5°C".

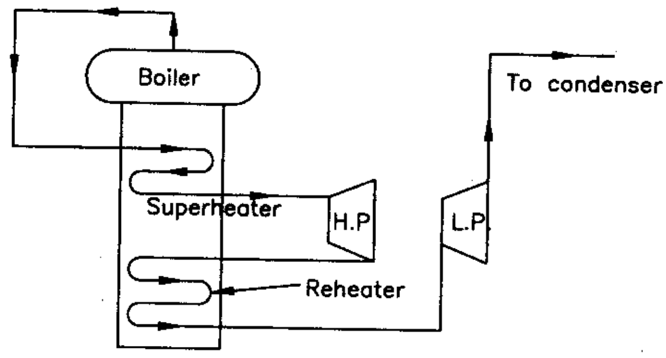


Fig. 22.6. (*a*) Steam reheating with flue gases.

The reheater is always placed behind the high pressure super-heater so that the former receives flue gases which have passed through the latter. Since, it is not possible to increase the gas temperature at the high pressure of the superheater at will, therefore, the reheater must operate at a much smaller temperature difference between the gas and steam. This is why reheating normally employs counterflow heat exchanger.

In this arrangement of reheating, the steam can be reheated to initial throttle temperature but it suffers from the following disadvantages :

(1) This requires long and large pipe connections. Therefore, the cost is more as well as pressure drop is higher. This may be as high as 10%.

(2) A provision for the expansion and contraction of the piping should be made.

(3) The amount of steam stored within the piping and reheater is sufficient to cause a considerable rise in turbine speed in case of the failure of emergency control.

(2) **Live-Steam Reheating.** The live-steam reheating system is shown in Fig. 22.6 (*b*). The high pressure steam from the boiler is used for reheating the steam coming out from H.P. turbine in a specially designed heat-exchanger.

This type of reheating offers the following advantages over gas heating :

(1) It is simple in operation.

(2) The reheater can be placed near the turbine thus avoiding the use of large piping.

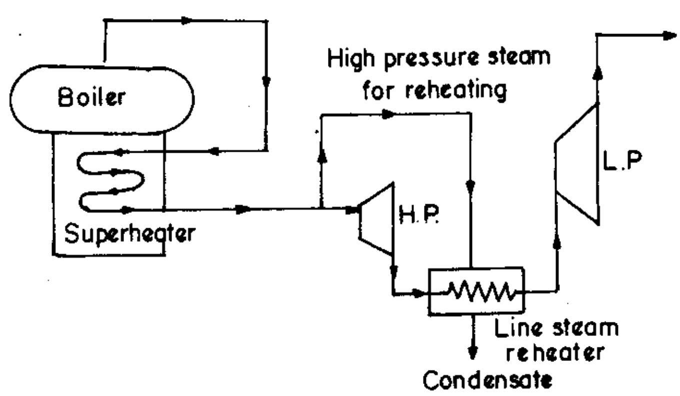


Fig. 22.6. (*b*) Steam reheating with live steam.

(3) The temperature control is simple as the change in combustion condition does not effect the live steam reheater performance.

(4) Wet steam can also be reheated.

(5) More than one reheating can be used as the piping requirements are low.

De-Smaele's curves are generally used to find out the proper reheat pressure so as to attain highest thermal efficiency of the turbine having live steam reheater. These curves are based on the assumption that the steam is reheated to saturation temperature corresponding to the throttle pressure and that the reheated steam suffers 3% drop in pressure.

(3) **Combined Gas and Live Steam Reheater.** The arrangement of combined heating system is shown in Fig. 22.6 (*c*).

Live steam reheating suffers from the disadvantage that the steam cannot be reheated to its initial throttle temperature. This difficulty is avoided with the use of combined heating system. The live steam heating system is placed in series with the gas reheater. The steam coming out from the H.P. turbine is first passed through the live steam reheater and then to gas reheater as shown in figure. The supply of live steam to first reheater is thermostatically controlled so as to maintain a constant final temperature.

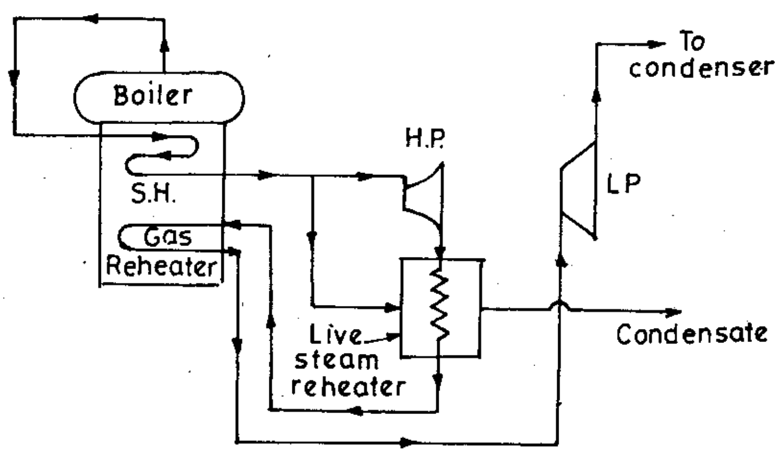


Fig. 22.6. (*c*) Combined live steam and gas reheating system.

There is optimum reheat pressure which gives best thermal efficiency of the plant for the given initial pressure and temperature of the steam. The effect of ratio of reheat pressure to initial pressure on the reheat cycle efficiency taking initial pressure as parameter is shown in Fig. 22.7.

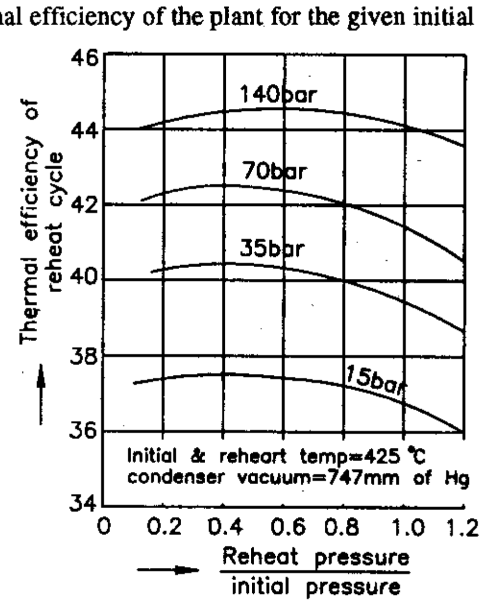


Fig. 22.7. Effect of ratio of reheat pressure to initial pressure on the thermal efficiency of the cycle.

Design Considerations of Reheat Cycle

The reheat cycle is rarely used for power plant of below 10 MW capacity and is not used with less than 30 bar pressure and 400°C temperature. All the plants above 120 MW capacity prefer the reheat cycle.

(1) **Steam Conditions.** The initial pressure of 100 bar to 150 bar and initial temperature of 500°C plant always prefer reheat cycle.

(2) **Turbine Arrangement for Reheat.** Normally tandeem compounding as shown in Fig. 22.8 (*a*) and cross compounding as shown in Fig. 22.8 (*b*) are used when reheating is employed and generators run at 3600 RPM.

(3) Where both initial and reheated steam are admitted to the same casing then opposed flow arrangement is used. The system concentrates highest temperatures in a small area

of high pressure casing so that temperature differentials and resulting tendencies to distortion are minimised. In addition to this, thrust is practically balanced.

(4) Ferritic materials are used for shells, pipes and valve bodies at 550°C in place of austenitic materials used earlier at this temperature. Austenitic material is used only for valves, valve steams and bushing.

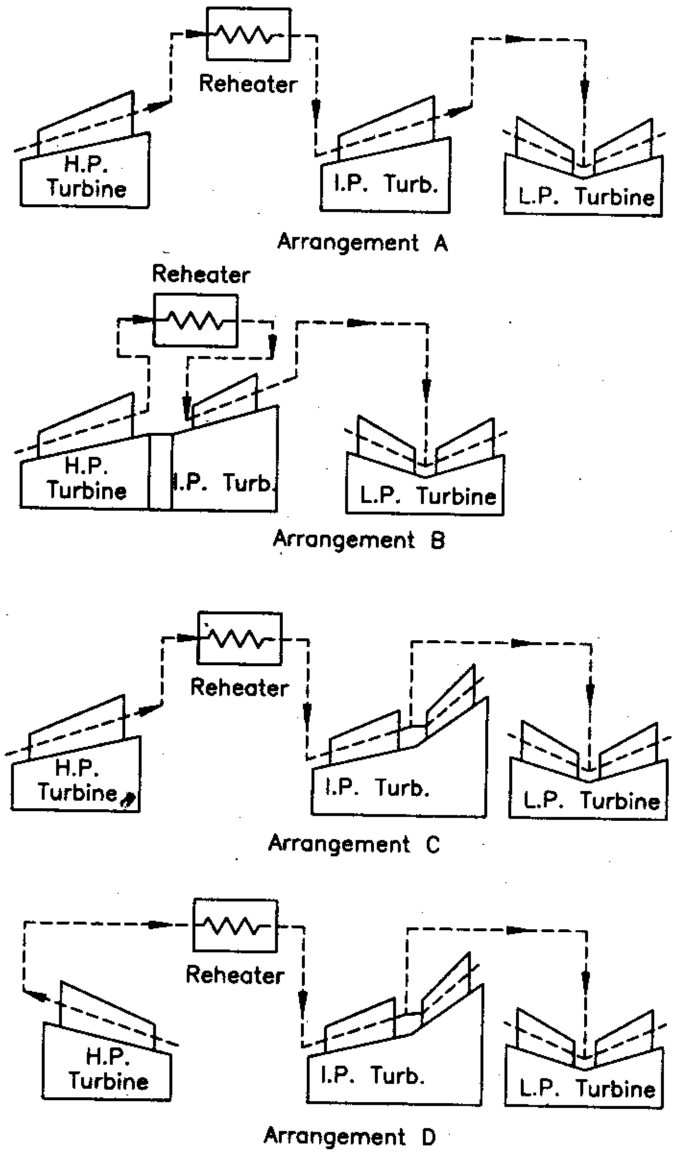


Fig. 22.8. (*a*) Arrangement of Tandem-Compound Reheat Turbines.

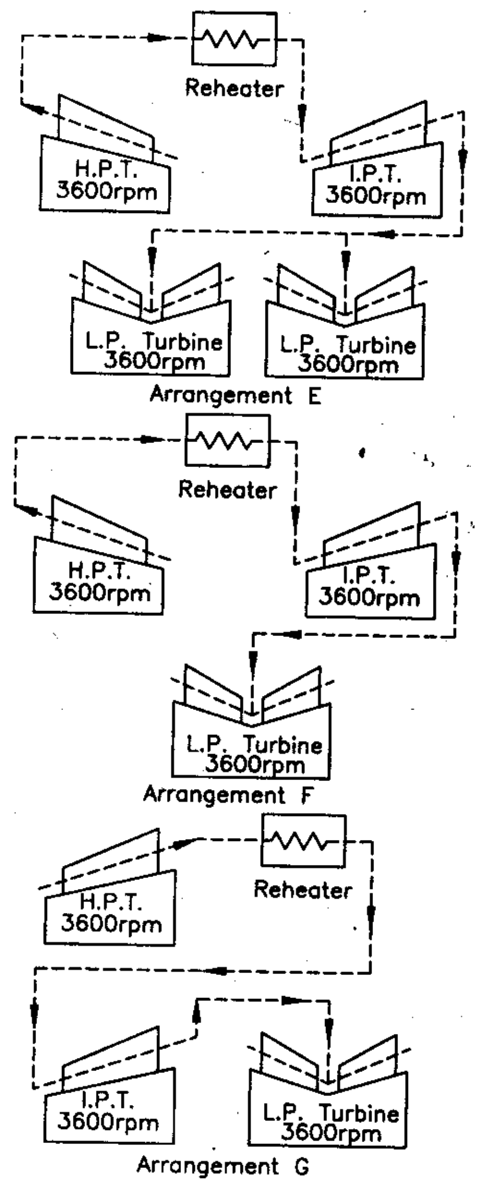


Fig. 22.8 (*b*) Arrangement of Cross-Compound Reheat Turbines.

Economics of reheat cycle. The economic justification of the reheat cycle is very difficult to make as the factors involved vary considerably from installation to installation. However, E.E. Harries and A.O. While have shown that for reheat pressures below 35 bar ; the percentage gain in economy of reheat over non-reheat is 5%, the reduction in steam flow is 16.5% at full load and heat load on condenser is reduced by 8.7%, all other conditions remain same except for reheat.

It is justified that t. . use of reheat cycle is more economical when the capacity of the plant is more than 10 MW and load factor is more than 50% and fuel prices are increasing rapidly.

The future of reheat cycle is very bright as reheat cycle is commonly used in U.S.A. and European countries above the 10 MW capacity of the units. Four units at Chandrapura power station of D.V.C. of 140 MW each are using reheat cycle. A unit of 150 MW capacity at Trombay also uses a reheat cycle.

The use of reheat cycles in Indian power-plants has opened a new epoch. The introduction of reheat-cycle in Indian power plants is justified on the assumption that the load factors will increase and fuel prices will also go on increasing. The trend is quite apparent and, therefore, the attractive methods by which operating costs are reduced definitely encourage for more and more use of reheat cycle for future units.

The number of reheats used also depends upon the economical justification which requires optimization of the system to decide the number of reheats to produce the power at cheapest rate.

22.3. REGENERATIVE CYCLE

The Rankine efficiency of a steam cycle is less than Carnot cycle as all the heat is not supplied as shown in Fig. 22.9 at the highest temperature. The heating of feed water from 3 to 4 takes place at lower temperature. The Rankine cycle will be as efficient as Carnot if the temperature of feed water is raised to boiler temperature by ***reversible interchange*** of heat before it enters the boiler.

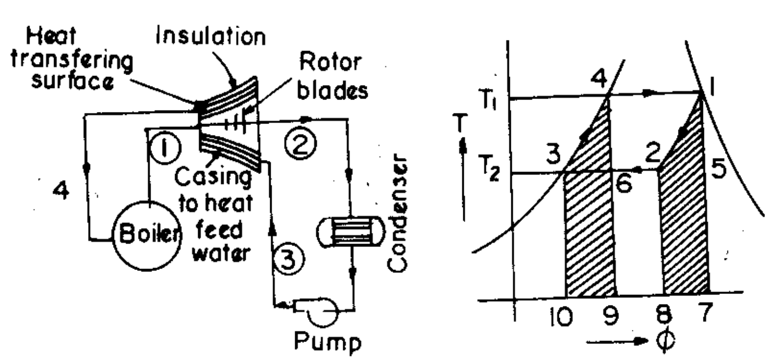
An arrangement for reversible heat transfer to feed water before entering into the boiler is shown in Fig. 22.9 (*a*) and its corresponding $T - s$ diagram is shown in Fig. 22.9 (*b*).

The dry-saturated steam coming out from boiler enters into the turbine at temperature T_1 and it expands adiabatically to temperature T_2. The condensate coming out from condenser is pumped back through the annular space of the turbine as shown in figure. The feed water is heated with the help of steam in a reversible manner, the temperature of steam and water is same at any section.

Such type of heating is known as ***Regenerative Heating*** as the steam is used to heat itself (feed water). Under ideal regeneration condition, the feed water enters at temperature T_1 and represented by the point 4. The temperature of steam entering the turbine and temperature of water leaving the turbine are same.

If the system is considered as isolated adiabatic system, the heat lost by the steam must be equal to heat gained by the water.

$\therefore$ Area "3 – 4 – 9 – 10 – 3" = Area "1 – 7 – 8 – 2 – 1".



(*a*) (*b*)

Fig. 22.9.

This type of heating arrangement (non-extraction feed heating) gives the efficiency equivalent to Carnot cycle efficiency. This type of arrangement cannot be used in practice because steam becomes too wet in the latter stages of the turbine. In actual practice, the advantage of regenerative heating principle is taken by "bleeding" a part of steam from the turbine at certain stages of expansion and it is used for heating the feed water in the separate feed heaters. This arrangement does not reduce the dryness fraction of remaining steam passing through the turbine.

If there were an infinite number of these extraction feed heating stages, the resulting cycle would be thermodynamically equivalent to the non-extraction feed heating cycle. The use of regenerative feed heating enabled the Rankine cycle to be improved upon and a nearer approach made to Carnot cycle.

To heat the condensate from the hot-well temperature to a temperature corresponding to the saturation temperature of steam at the stop valve requires continuous heat transfer from the heating medium with smallest temperature difference. This can be achieved only with infinite number of feed heating stages. But, in practice, it is not possible to have infinite heaters.

The number of feed heaters used in the system is largely a problem of economic evaluation, in which the annual saving in fuel costs is balanced against annual fixed charges on heaters investment to determine how many heaters are required in a given installation for minimum generating costs.

Five extraction points are generally used in practice. Some plants (using critical pressure) use as many as nine feed heaters.

The effect of number of heaters used on the cycle efficiency taking supply-pressure as parameter is shown in Fig. 22.10 (*a*).

The rate of efficiency improvement against the feed temperature is approximately shown in Fig. 22.10 (*b*). It is obvious from the figure that efficiency increases with an increase in number of feed heaters. The optimum feed temperature rises with the increase in number of feed heaters, but the final feed temperature should be kept considerably below the boiler saturation temperature to guard against evaporation in the feed lines.

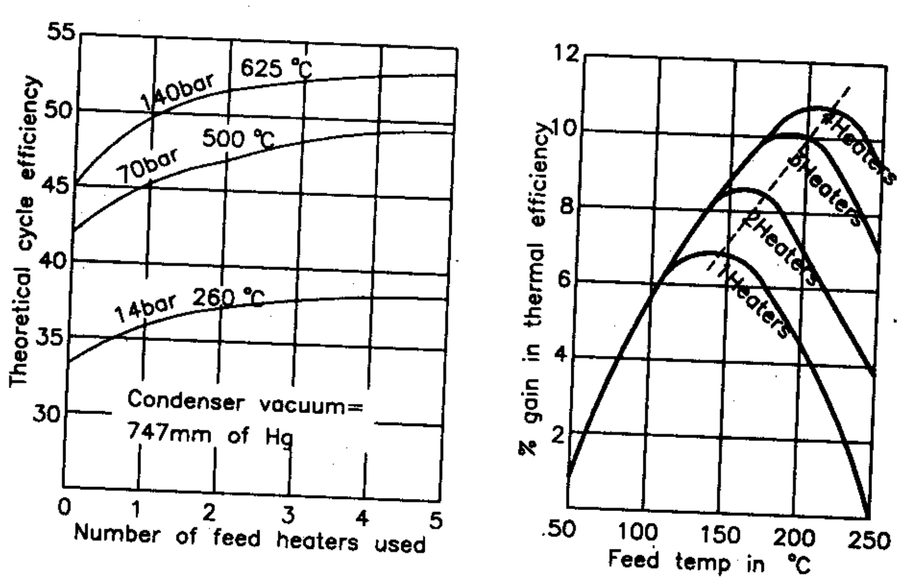


Fig. 22.10. (*a*) Effect of number of heaters on thermal efficiency of the plant.

Fig. 22.10 (*b*) Gains in thermal efficiency due to feed heaters.

Disposal of Bled Steam Condensation. The use of bled steam for heating the feed water and methods of disposal of bled steam condensate are discussed below.

(*a*) **Direct Contact Heaters.** The steam bled from the different points of turbine is mixed directly with the feed water to increase the temperature of feed water and the steam mixed with feed water is extracted with the help of pump as shown in Fig. 22.11 and supplied to the boiler.

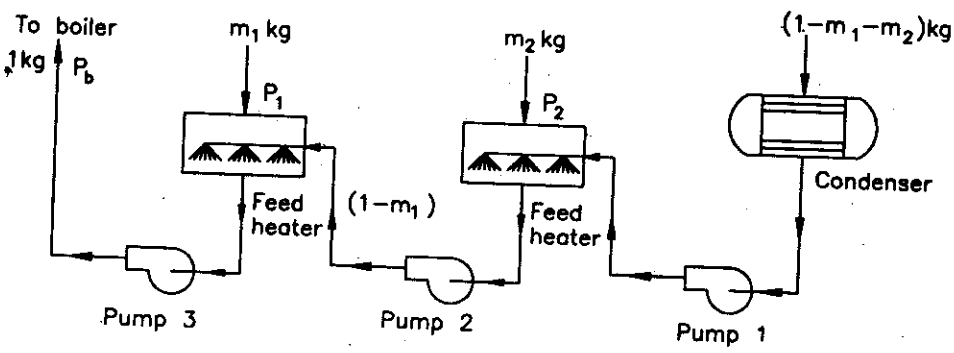


Fig. 22.11.

The main disadvantage of this system is the pump has to work with hot feed. Therefore, this system is rarely used in practice.

(*b*) **Drain Pump Method.** In this method, the feed water is heated with indirect contact of bled steam in the heat exchangers as shown in Fig. 22.12.

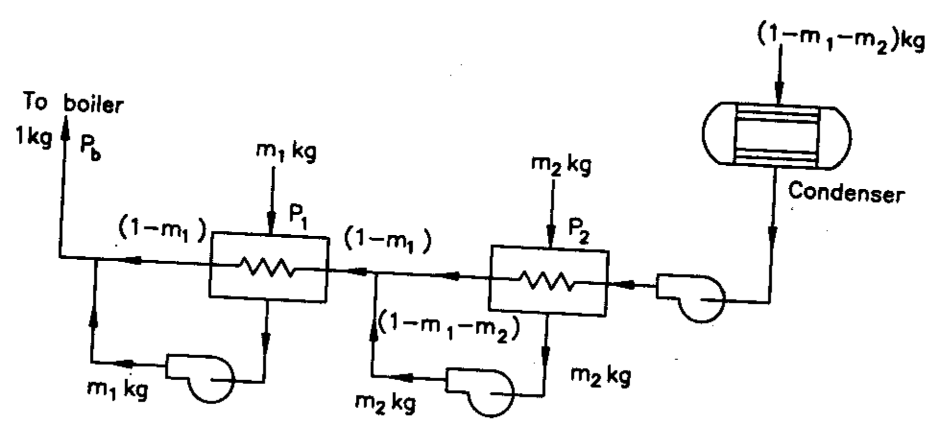


Fig. 22.12.

The bled condensate is extracted by the drain pump and discharges into the feed pipe line as shown in Fig. 22.12. This method also suffers from the same disadvantage as mentioned in the previous arrangement.

(*c*) **All Drains to Hot Well.** In this method, the condensate of the bled steam coming out from the indirect heat exchangers is fed to the hot well as shown in Fig. 22.13. The total condensate (condensate coming from condenser + condensate from bled steam) from the hot well is pumped to the boiler through regenerative feed heaters as shown in Fig. 22.13.

(*d*) **Cascade System.** In this method, the condensate of bled steam coming out from the first heat

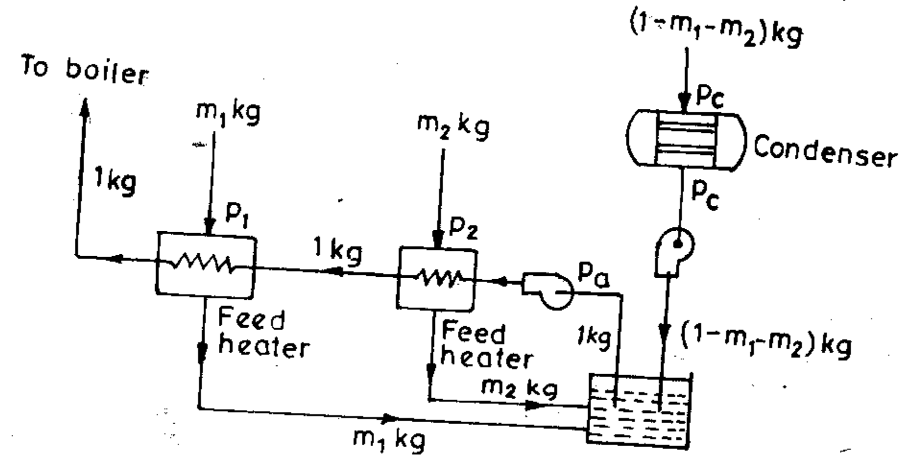


Fig. 22.13.

exchanger is passed through the second heat exchanger and lastly to the hot well as shown in Fig. 22.14.

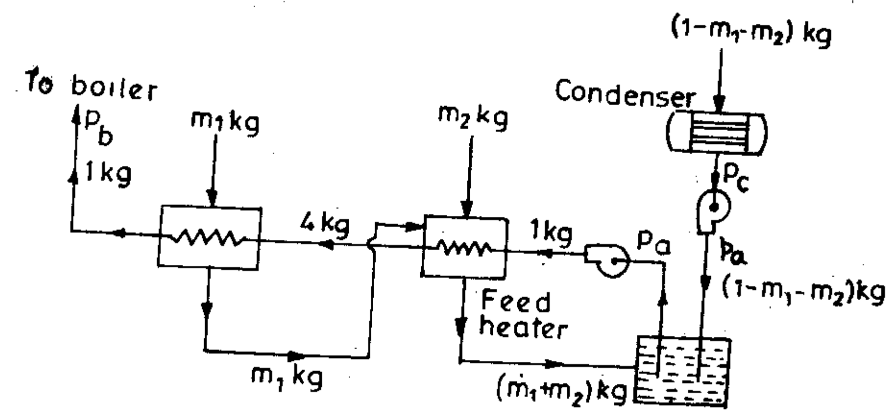


Fig. 22.14.

The total condensate (condensate from condenser and condensate of bled steam) from the hot well is pumped to the boiler through the regenerative heat exchangers.

This type of feed-heating is often used to save the cost of pump. The heaters used may be either of closed type in which the water and bled steam do not mix with each other or of open type in which wateer and bled steam mix with each other.

Efficiency of Regenerative Cycle. For calculating the efficiency of the regenerative cycle, the following assumptions are made :

(1) The bled steam is condensed and cooled to the temperature of the entering feed water at the stage considered and the feed water is heated to the saturation temperature at the pressure of the bled steam.

(2) The bled steam gives up its superheat if any and all its latent heat to the water to be heated. The condensed steam leaves the heater at the saturation temperature corresponding to the bleeding pressure.

(*a*) **Efficiency of Direct Contact Heating Cycle.** The arrangement of the system is shown in Fig. 22.15 (*a*) and corresponding *T-s* diagram is shown in Fig. 22.15 (*b*).

The point '*b*' represents the condition of steam entering the turbine and point '*c*' represents the condition of steam entering into the condenser.

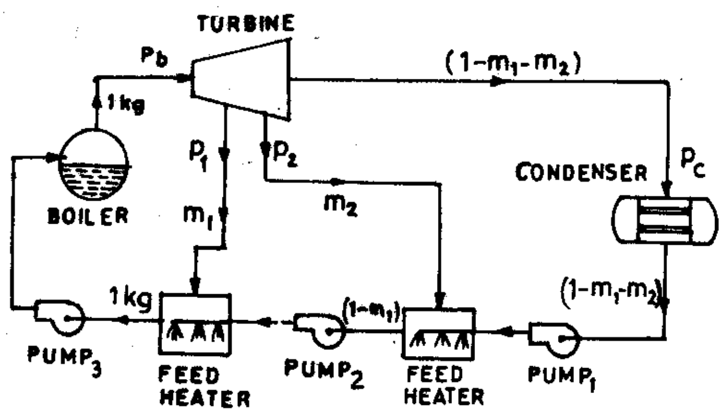


Fig. 22.15 (*a*).

The processes 3 – 4, 5 – 6, 7 – 8 represent the pump processes and the processes 4 – 5, 6 – 7 represent the heating into the feed heaters. The process 8 – 9 – 10 – *b* represents the boiler process where the heat is added to generate the steam whose quality is represented by the point '*b*' when the feed water is supplied to the boiler at condition '8'.

Work drawn per kg of steam supplied to the turbine is given by

$W = (h_b - h_1) + (1 - m_1)\ (h_1 - h_2) + (1 - m_1 - m_2)\ (h_2 - h_c)$ ...(22.5)

Total pump work is given by

$W_p = W_{p1} + W_{p2} + W_{p3}$

$= v_w\ (1 - m_1 - m_2)\ (p_2 - p_c) + v_w\ (1 - m_1)\ (p_1 - p_2) + v_w\ (p_b - p_1)$ ...(22.6)

In the above expression the specific volume of feed water is taken same at all temperatures. In calculation the specific volume at mean temperature should be taken.

$\therefore$ Net work done in the system is given by

$W_n = W - W_p$ ...(22.7)

The heat supplied by the boiler per kg of steam generated is given by

$h_s = h_b - h_{f8}$ ...(22.8)

Therefore, the efficiency of the regenerative cycle is given by

$$\eta_{regenerative} = \frac{W_n}{h_s} = \frac{W - W_p}{h_b - h_{f8}} \quad ...(22.9)$$

Neglecting the heat losses and considering the heat balance at feed heaters, we can find out the values of m_1 and m_2 required to be bled from steam turbine.

Heat lost by the steam = Heat gained by the water.

$\therefore\ m_2\ (h_2 - h_{f5})$

$= (1 - m_1 - m_2)\ (h_{f5} - h_{f3})$ ...(22.10)

and $m_1\ (h_1 - h_{f7})$

$= (1 - m_1)\ (h_{f7} - h_{f5})$ ...(22.11)

The rise in temperature of water due to pump action is neglected.

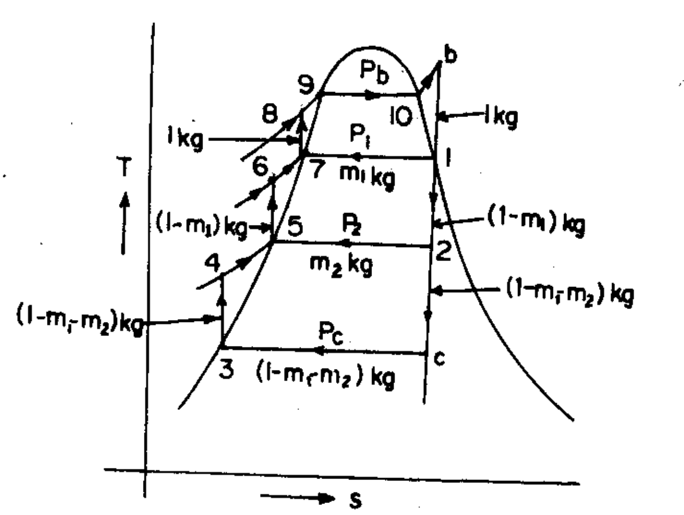


Fig. 22.15 (*b*).

From the above two equations, we can find the values of m_1 and m_2 if the pressures p_1 and p_2 are known.

(*b*) **Efficiency of indirect contact heating cycle.** The arrangement of the system is shown in Fig. 22.16 (*a*) and corresponding $T-s$ diagram is shown in Fig. 22.16 (*b*).

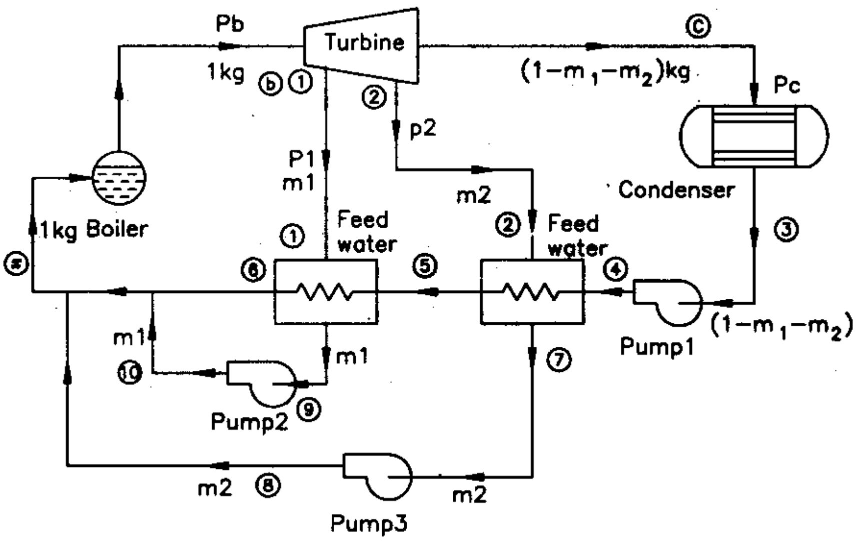


Fig. 22.16 (*a*).

The condition of the fluid at different stages is represented on $T-s$ diagram and corresponding numbers representing the condition of steam for easy understanding to the students.

Work done per kg of steam supplied to the turbine is given by,

$$W = (h_b - h_1) + (1 - m_1)(h_1 - h_2) + (1 - m_1 - m_2)(h_2 - h_c) \quad ...(22.12)$$

Total pump work is given by

$$W_p = W_{p1} + W_{p2} + W_{p3}$$
$$= v_w (1 - m_1 - m_2)(p_b - p_c) + v_w (m_2)(p_b - p_2) + v_w (m_1)(p_b - p_1) \quad ...(22.13)$$

The pressure losses in the system are neglected and specific volume of the feed water is considered constant.

$\therefore$ Net work done is given by,

$$W_n = W - W_p.$$

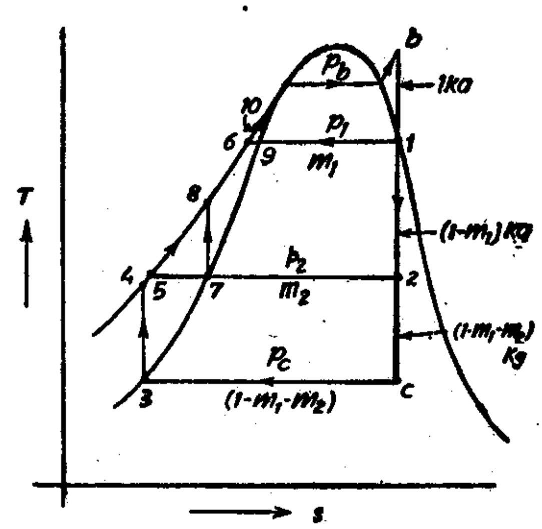


Fig. 22.16 (*b*).

The heat supplied by the boiler per kg of steam generated is given by,

$$h_s = h_b - h_{fx} \quad ...(22.14)$$

where h_{fx} is the enthalpy of feed water entering into the boiler.

Therefore, the efficiency of regenerative cycle is given by,

$$\eta_{regenerative.} = \frac{W_n}{h_s} = \frac{W - W_p}{h_b - h_{fx}} \quad ...(22.15)$$

Neglecting the heat losses and considering the heat balance at feed heaters ; we can find the values of m_1 and m_2 required to be bled from steam turbine.

$$m_2 (h_2 - h_{f7}) = (1 - m_1 - m_2) (h_{f7} - h_{f3}) \quad ...(22.16)$$

and

$$m_1 (h_1 - h_{f9}) = (1 - m_1 - m_2) (h_{f9} - h_{f7}) \quad ...(22.17)$$

In the above two expressions, the rise in temperature of water due to pump actions is neglected as it is very small and it is considered that the temperature of feed water going out of heat exchanger is equal to the saturation temperature of steam entering into the heat exchanger.

The condition of the feed water (h_x) going to boiler is given by the following expression :

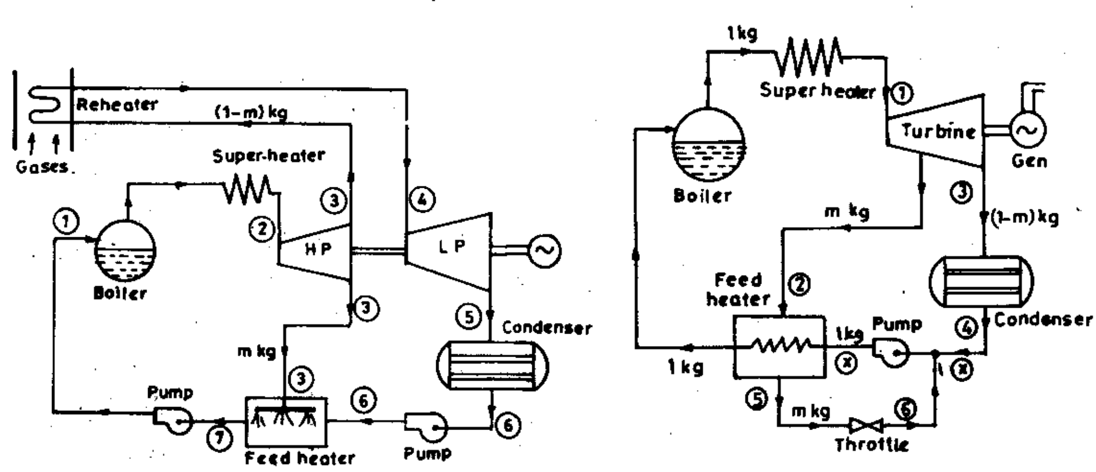
$$(1 - m_1 - m_2) h_{f9} + m_2 h_{f7} + m_1 h_{f9} = 1 \times h_x$$

$$\therefore \quad h_x = (1 - m_2) h_{f9} + m_2 h_{f7} \quad ...(22.18)$$

In the above expression also, the rise in temperature due to pump action is neglected.

In actual thermal power plants, the reheat of steam with one or two heaters, and regeneration with one or two feed heaters are generally used to increase the overall efficiency of the thermal plant.

Comparison between Closed and Open Type Feed Heaters



(*a*) Open Type Feed Heaters (*b*) Closed Type Feed Heaters

Fig. 22.17.

The open type and closed type feed heaters are shown in Fig. 22.17. The open types are more simple and have good heat transfer characteristics. In this type, one pump is required with each feed heater as shown in Fig. 22.17 (*a*).

The closed types are more complex in construction as piping is concerned. Its heat transfer characters are poor compared with open type. The only advantage is, it requires only one pump but pump work is more than the pump-works of the open type as high pressure feed is throttled to low pressure first and then pressurised.

In modern power plants, to achieve optimum economy (additional capital cost of feed heaters and increased efficiency) about 8-feed-heaters are used. It is common practice to include one open heater at moderate pressure, which incorporates a de-aerator to remove the dissolved air which reduces heat transfer rate in condenser and maintaining the high vacuum in condenser becomes more difficult. The remaining heaters are of closed type.

22.4. REHEAT-REGENERATIVE CYCLE

In actual thermal power plant with high steam pressure (above 100 bar), the reheat regenerative cycle is used to increase the overall efficiency of the cycle. The use of reheat for the plant using steam below 100 bar is not economically justified.

The arrangement of this cycle is shown in Fig. 22.18.

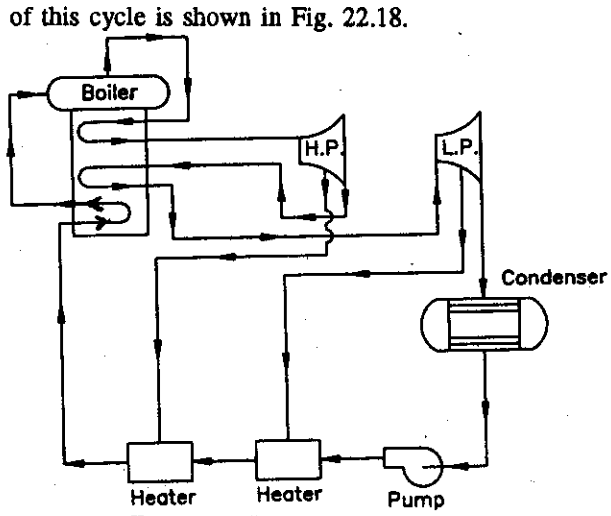


Fig. 22.18. Reheat Regenerative cycle.

The thermal efficiency of the reheat-regenerative cycle is higher than only reheat or only regenerative cycle.

22.5. BINARY VAPOUR CYCLE

The Carnot cycle-efficiency will increase with an increase in initial or supply temperature or with the decrease in exit temperature. It has been already seen that the exit temperature of the steam is limited by the available technique to maintain the vacuum in the condenser and temperature of cooling water. Using steam as working fluid, the rise in temperature is accompanied by rise in pressure (critical temperature is 374.15°C at critical pressure of 225.65 bar). High pressure of the steam creates many difficulties in design, operation and control.

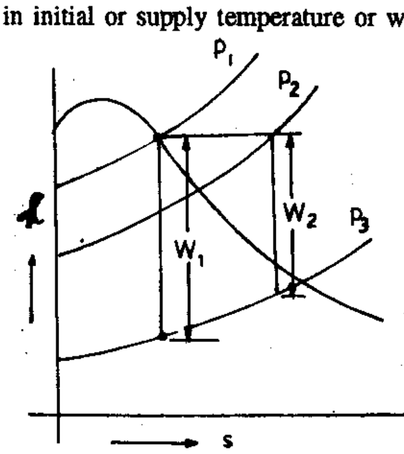


Fig. 22.19.

It would be desirable to use another fluid than steam which has more desirable thermodynamic properties than water has. This possibility has been considered by many scientists and engineers and extensive research has been made in last five decades.

It would seem at first sight that, by superheating, high steam temperature could be obtained without the necessity of high pressures. However, it is found that the thermal efficiency of the cycle depends upon the saturation temperature corresponding to supply pressure rather than on the superheat temperature. This point is clear from Fig. 22.19 as $W_2 < W_1$.

In order to increase the overall efficiency of the plant, the initial temperature of the steam should be as high as possible. Very high pressures are required with moderate temperatures if water is used as working fluid, therefore, very rugged equipment and special types of boilers are required.

It is possible to use the fluid which will increase the temperature range of the complete cycle and yet avoid the difficulties connected with high pressures. The overall economy of the cycle will be increased if the fluid used has desirable thermodynamic properties.

It is obvious from the above discussion that we should use the working fluid other than steam which has low saturation pressure at high temperature. Mercury is the only working fluid which has been successfully used in practice. It has high critical temperature (588.4°C) at correspondingly low critical pressure (21 bar).

The mercury alone cannot be used as the saturation temperature of Hg at atmospheric pressure is sufficiently high. Hence binary vapour cycle is generally used to increase the overall efficiency of the plant. Two fluids (mercury and water) are used in the binary cycle for the production of power.

The few more properties required for an ideal binary fluid used in higher temperature limit are listed below.

1. It should have high critical temperature at reasonably low pressure.
2. It should have high heat of vaporization to keep the weight of fluid in the cycle to minimum.
3. Freezing temperature should be below room temperature.
4. It should have chemical stability through the working cycle.
5. It must be non-corrosive to the metals normally used in power plants.
6. It must have an ability to wet the metal surfaces to promote the heat transfer.
7. The vapour pressure at a desirable condensation temperature should be nearly atmospheric which will eliminate the requirement of power for the maintenance of vacuum in the condenser.
8. After expansion through the primemover, the vapour should be nearly saturated so that a desirable heat transfer coefficient can be obtained which will reduce the size of the condenser required.
9. It must be available in large quantities at reasonable cost.
10. It should not be toxic and, therefore, dangerous to human life.

Although mercury does not have all the required properties, it is more favourable than any other fluids investigated. It is most stable under all operating conditions.

Although, Hg does not cause any corrosion to the metals, but it is extremely dangerous to human life, therefore, elaborate precautions must be taken to prevent the escape of vapour. The major disadvantage associated with mercury is that it does not wet the surface of the metal and forms a serious resistance for heat flow. This difficulty can be considerably reduced by adding magnesium and titanium (2 parts in 100000 parts) in mercury.

Theremal Properties of Mercury. Mercury fulfills practically all the desirable thermodynamic properties stated above.

1. Its freezing point is – 40°C and boiling point is 355°C at atmospheric pressure.
2. The pressure required when the temperature of vapour is 430°C is only 32 bar and, therefore, heavy construction is not required to get high initial temperature.
3. Its liquid specific heat is only 0.08 kJ/kg so that its liquid saturation curve is very steep, approaching the isentropic compression of the Carnot cycle.
4. It has no corrosive or erosive effects upon metals commonly used in practice.
5. Its critical temperature is so far removed from any possible upper temperature limit with existing metals as to cause no trouble.

Some undesirable properties of Hg are listed below.

1. The latent heat of Hg is only 262.5 kJ/kg over a wide range of desirable condensation temperatures, therefore, several kg of Hg must be circulated per kg of water evaporated in binary cycle.
2. The cost is a considerable item as the quantity required is 8 to 10 times the quantity of water circulated in binary system.
3. Mercury vapour in large quantities is poisonous, therefore, the system must be perfect and tight.

Figure 22.20 (*a*) shows the schematic line diagram of binary vapour cycle using mercury and water as working fluids. The processes are represented on *T-s* diagram as shown in Fig. 22.20 (*b*).

The steam cannot be superheated in mercury condenser (or steam boiler) as some temperature gradient is required for the flow of heat from mercury vapour to water.

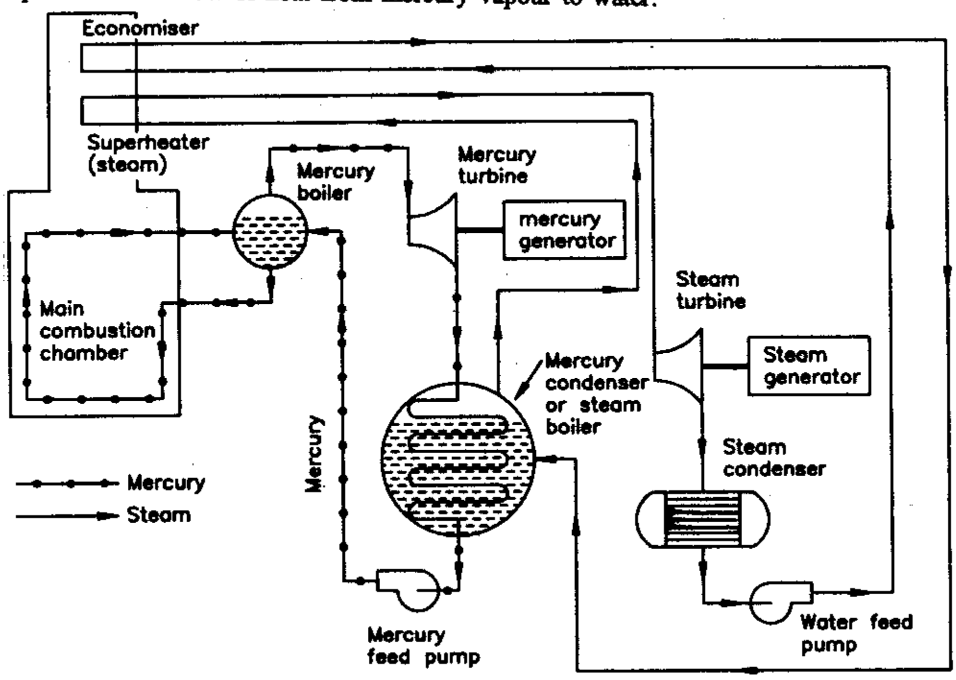


Fig. 22.20 (*a*). Line diagram of binary vapour cycle.

Analysis of binary vapour cycle

h_{hg1} = Heat supplied per kg of mercury vapour formed in the mercury boiler.

h_{hg2} = Heat lost by one kg of mercury vapour in the mercury condenser.

h_s = Heat given per kg of steam generated in the mercury condenser or steam boiler.

W_{hg} = Work done per kg of mercury in the cycle.

W_s = Work done per kg of steam in the steam cycle.

η_s = Thermal efficiency of the steam cycle.

η_{hg} = Thermal efficiency of the mercury cycle.

m = Mass of mercury in the mercury cycle per kg of steam circulated in the steam cycle.

In the following analysis, the heat losses to the surroundings are neglected and steam generated is considered one kilogram and mercury in the circuit is m kg per kg of water in the steam cycle.

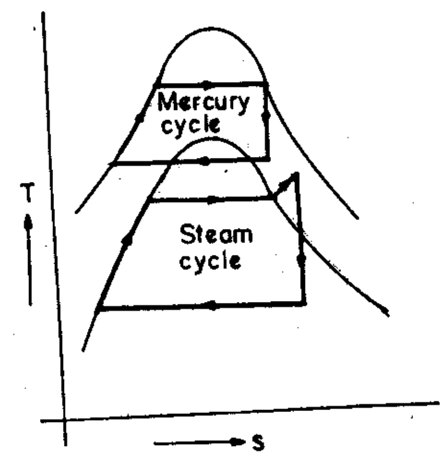


Fig. 22.20 (*b*).

Heat supplied in the mercury boiler,

$$h_t = m \times h_{hg1} \quad \text{...(22.19)}$$

Work done in the mercury cycle

$$= m \,.\, W_{hg} \quad \text{...(22.20)}$$

Work done in the steam cycle

$$= 1 \times W_s \quad \text{...(22.21)}$$

Total work done in the binary cycle is given by

$$W_t = mW_{hg} + W_s \quad ...(22.22)$$

$\therefore$ Overall efficiency of the binary cycle is given by

$$\eta = \frac{\text{Work done}}{\text{Heat supplied}} = \frac{W_t}{h_t} = \frac{mW_{hg} + W_s}{mh_{hg1}} \quad ...(22.23)$$

Thermal efficiency of the mercury cycle is given by

$$\eta_{hg} = \frac{mW_{hg}}{mh_{hg1}}$$

$$= \frac{W_{hg}}{h_{hg1}} = \frac{h_{hg1} - h_{hg2}}{h_{hg1}} = 1 - \frac{h_{hg2}}{h_{hg1}} \quad ...(22.24)$$

$$= \frac{mh_{hg1} - h_s}{m.h_{hg1}} = 1 - \frac{1}{m} \cdot \frac{h_s}{h_{hg1}} \quad ...(22.25)$$

Heat lost by mercury vapour = Heat gained by steam

$$\therefore \quad mh_{hg2} = 1 \times h_s \quad ...(22.26)$$

Substituting the value of m from equation (22.26) into equation (22.25), we get

$$\eta_{hg} = 1 - \frac{h_{hg2}}{h_{hg1}} \quad ...(22.27)$$

The thermal efficiency of the steam cycle is given by

$$\eta_s = \frac{W_s}{h_s} = \frac{h_{s1} - h_{s2}}{h_{s1}} = \frac{h_{s1} - h_2}{m.h_{hg2}} \quad ...(22.28)$$

From the equations (22.23), (22.25), (22.26), (22.27) and (22.28), we get

$$\eta = \eta_{hg}(1 - \eta_s) + \eta_s \quad ...(22.29)$$

The equations (22.24), (22.28) and (22.29) are used for solving the problems.

Another important problem in the design of binary cycle is the limit of exhaust pressure of the mercury (location of optimum exhaust pressure) which will provide maximum work per kg of Hg circulated in the system and high thermal efficiency of the cycle. It is not easy to decide as number of controlling factors are many.

Mercury-steam power plants. The first experimental mercury turbine of 1800 kW capacity was installed at Dutch-point Conn in 1923. Number of difficulties were experienced during its operation. A larger commercial unit was installed in 1928 at Hartford by General Electric Co. co-operating with Harford Electric Co. After sufficient experimental data were obtained on this installation, a 10 MW capacity plant was installed in 1930 at South Meadow station of the Hartford Electric Company. A binary power plant of 50 MW capacity (20 MW capacity of mercury turbine and 30 MW capacity of steam turbine) was installed at Kearny in U.S.A. The weight of the mercury used in this plant was 196 tons and the overall efficiency of the plant was 37.2%. The recent mercury steam installation is Schiller station of 40 MW capacity at New Hampshire installed in 1949.

The other fluids considered in binary system are Diphenyl $(C_6H_5)_2$, Diphemyloxide $(C_6H_5)_2O$. Aluminium bromide Al_2Br_6 and Zinc ammonium chloride $Zn(NH_3)_2Cl$. The readers interested in more details concerning the applications of these fluids in binary cycle systems are advised to refer to chapter No. 24 from "steam power stations" by Graffert.

It is necessary to consider the capital cost involved in the binary system (cost of mercury) in addition to the fuel saving due to increase in thermal efficiency to obtain a true picture of the overall economics of the mercury steam cycle.

An experience with operating plants has proved that the binary cycle has real economic merits over ordinary steam cycle. The main thing lacking in the development of such plant is the knowledge of the design and operation.

The mercury plants are preferred where acute shortage of important facilities exists such as deficiency of condenser cooling water, insufficient steam-boilers and demand for electrical generating capacity.

With an advancement of critical and supercritical boilers, the binary-power plants are rarely used now-a-days as modern thermal power plants give equally high thermal efficiencies.

22.6. SUPERPOSED OR TOPPING CYCLE

The power demand by public utility section is seldom static. Therefore, whenever the demand increasees, the capacity of the existing thermal power plant may be expanded either by increasing the capacity of existing plant or by purchasing additional equipment similar to that already installed or by superposing a high pressure non-condensing steam plant on the existing plant. The second method used to cope up with an increasing demand for power is known as topping cycle. Where an existing plant is becoming inadequate chiefly due to growth of load rather than by natural depreciation, superposed power is more economical method of increasing the capacity of existing plant.

The capacity of existing plant working with moderate pressure and operating at good efficiency can be increased with an inclusion of a high pressure non-condensing steam plant.

The arrangement of the topping cycle is shown in Fig. 22.21.

It is possible to supply sufficient steam by the superposed unit into the original plant header. In this way, not only the excellent qualities of existing turbines are retained, but it is also possible to obtain the benefits of the progress that occurred in high pressure steam generation. Many plant extensions have been made with the help of topping cycle and have displayed remarkable economics of operation when compared with original units.

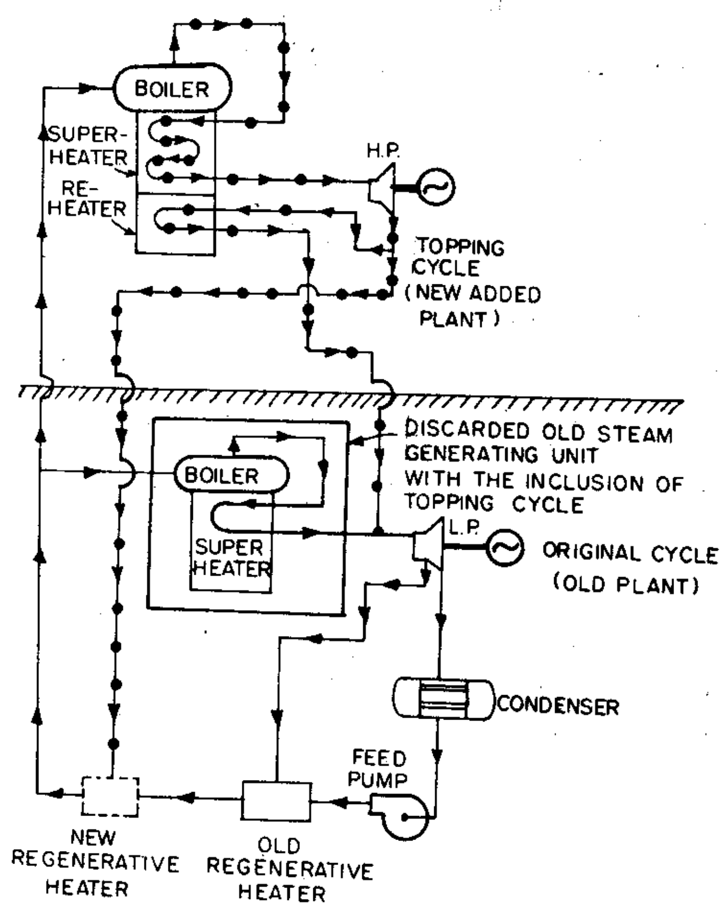


Fig. 22.21. Topping cycle.

SOLVED PROBLEMS

Problem 22.1. *A power plant using steam as working fluid operates on a Rankine cycle. The boiler and condenser pressures are 30 bar and 1 bar. The condition of steam entering the prime-mover is dry and saturated. Find the thermal efficiency of the cycle (a) neglecting the feed pump work and (b) considering the feed pump work.*

Solution. The components of the cycle are shown in Fig. Prob. 22.1 (*a*) and cycle is represented in Fig. Prob. 22.1 (*b*) and Fig. Prob. 22.1 (*c*) neglecting and considering the feed pump work.

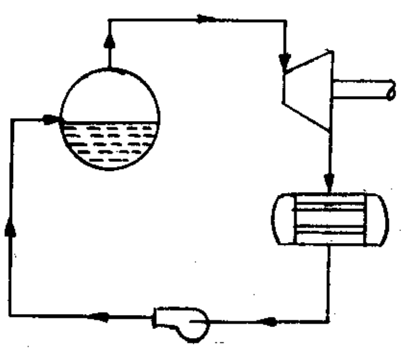


Fig. Prob. 22.1. (*a*) Components of power plant working on Rankine-cycle.

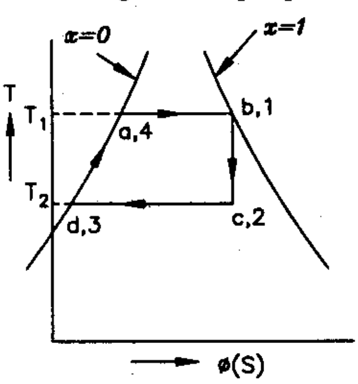


Fig. Prob. 22.1. (*b*) Rankine cycle without pump-work.

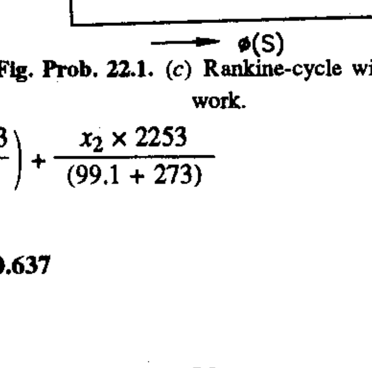


Fig. Prob. 22.1. (*c*) Rankine-cycle with pump work.

Total heat at the point '*b*' at pressure 30 bar is taken from the steam table

$$h_1 = 2796 \text{ kJ/kg}.$$

For finding the dryness-fraction of steam at the point '*c*', we can equate the entropies.

At pressure 30 bar At pressure 1 bar

$$\therefore \quad \log_e\left(\frac{T_{s1}}{273}\right) + \frac{h_{fg1}}{T_{s1}} = \log_e\left(\frac{T_{s2}}{273}\right) + \frac{x_2 h_{fg2}}{T_{s2}}$$

Taking the values T_{s1}, T_{s2}, h_{fg1} and h_{fg2} from steam tables and substituting in the above equation, we get

$$\log_e\left(\frac{232.8 + 273}{273}\right) + \frac{1797}{(232.8 + 273)} = \log_e\left(\frac{99.1 + 273}{273}\right) + \frac{x_2 \times 2253}{(99.1 + 273)}$$

$$\therefore \quad 0.62 + 3.55 = 0.307 + 6.06\, x_2$$

$$\therefore \quad x_2 = \frac{4.17 - 0.307}{6.06} = \mathbf{0.637}$$

$\therefore$ Total heat at point '*C*' or 2 is given by

$$h_2 = h_{f2} + x_2 h_{fg2}$$
$$= 414.6 + 0.637 \times 2253 = \mathbf{1849.8\ kJ/kg}$$

$$\eta_r = \frac{(h_1 - h_2)}{(h_1 - h_{f2})}$$

where h_{f2} is the feed water enthalpy at 1 bar

$$\therefore \quad \eta_r = \frac{2796 - 1849.8}{2796 - 414.6} \times 100 = \mathbf{39.55\%}.$$

The suffix '*f*' represents liquid water.

(*b*) When the pump work is considered, the Rankine efficiency is given by,

$$\eta_r = \frac{(h_1 - h_2) - W_p}{h_1 - (h_{f2} + W_p)}$$

where W_p (pump work) is given by,

$$W_p = \frac{v_{f2}\,(p_1 - p_2)}{J}$$

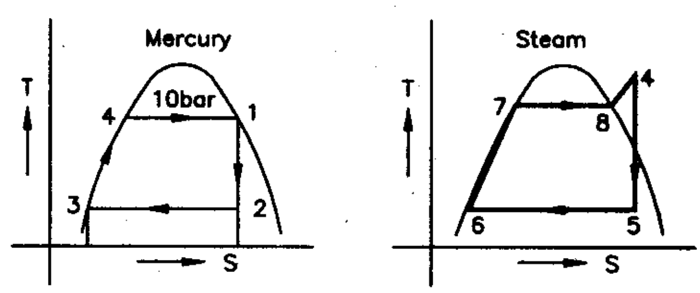
$$v_{w2} = 0.001043 \text{ m}^3/\text{kg (from steam table)}$$

$$\therefore \quad W_p = \frac{0.001043 \times 1^5\,(30 - 1)}{1000 \times 1} = \mathbf{3.02\ kJ}$$

$$\eta_r = \frac{(2796 - 1849.8) - 3.02}{(2796 - 414.6) + 3.02} \times 100 = \mathbf{39.57\%}.$$

Problem 22.2. *The mercury cycle of a binary vapour plant works between pressure of 10 bar and 0.08 bar and the mercury leaves the boiler as saturated vapour. In the steam cycle, the feed water is heated in an economiser and is evaporated in the mercury condenser with a temperature differential of 16°C to provide for heat transfer. The steam is externally superheated to 450°C and expanded to a saturation pressure corresponding to 33°C. Determine the cycle efficiency. Neglect pump work.*

Solution.



(*a*) (*b*)

Fig. Prob. 22.2.

From tables for mercury the following enthalpy values and entropy values are taken :

$h_1 = 359.11$, $s_{f2} = 0.087$,

$h_{f2} = 33.21$, $h_{fg2} = 294.7$,

$s_1 = 0.5089$, $s_{fg2} = 0.5721$

$s_1 = s_2$

$\therefore \quad 0.5089 = 0.087 + x_2 \times 0.5721$

$\therefore \quad x_2 = 0.7375$, and $h_2 = 33.21 + 0.7375 \times 294.7 = \mathbf{250.55}$.

From steam tables and chart

$h_4 = 3348.6,\quad h_5 = 2183,\quad h_6 = 138,\quad h_7 = 972,\quad h_8 = 2803.$

In the heat exchanger, mercury condensing from condition 2 to 3, evaporates the water from 7 to 8.

$$\therefore \quad \text{By heat balance : } \frac{m_{hg}}{m_{H2O}} = \frac{h_8 - h_7}{h_2 - h_3} = \frac{2803 - 972}{250.55 - 33.21} = 8.42.$$

For each kg of steam generated, 8.42 kg of mercury is to be used.

The heat added from external sources is :

(1) to evaporate mercury $= m_{Hg}\,(h_1 - h_3)$

(2) to preheat water to saturation condition = 1 $(h_7 - h_6)$

(3) to superheat the steam from 8 to 4 = 1 $(h_4 - h_8)$.

The works done are :

(*i*) $m_{Hg}\,(h_1 - h_2)$

(*ii*) 1 $(h_4 - h_5)$.

$$\therefore \quad \text{Cycle efficiency} = \frac{8.42\ (369.11 - 250.55) + (3348.6 - 2183)}{8.42\ (359.1 - 33.21) + (972 - 138) + (3348.6 - 2803)} = \mathbf{0.504.}$$

Problem 22.3. *In a nuclear power plant, saturated steam at 30 bar is expanded till the dryness is 0.841 and then the steam is passed through a moisture separator where all liquid particles are removed. The dry steam at this pressure is then expanded to 0.04 bar. The condensates are separated by pumps through different pumps to the boiler. Determine the cycle efficiency, dryness at exit and compare them with a unit without moisture separator.*

Solution.

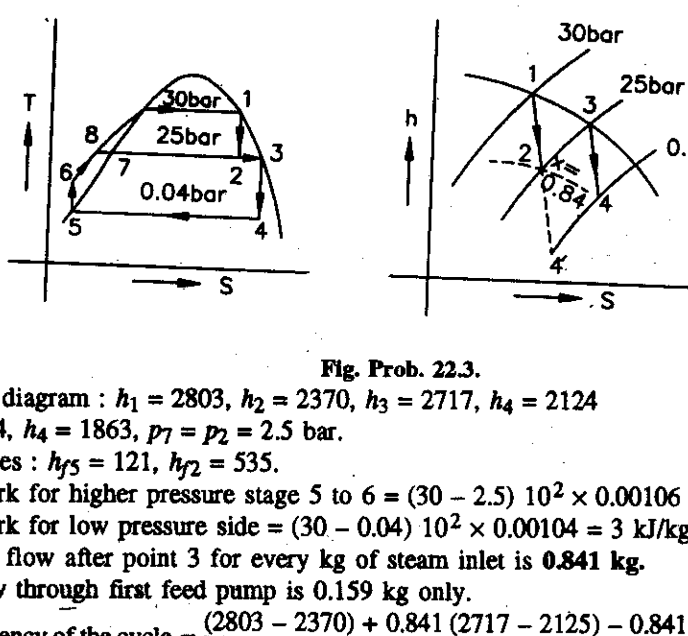


Fig. Prob. 22.3.

From *h-s* diagram : $h_1 = 2803$, $h_2 = 2370$, $h_3 = 2717$, $h_4 = 2124$

$x_2 = 0.824$, $h_4 = 1863$, $p_7 = p_2 = 2.5$ bar.

From tables : $h_{f5} = 121$, $h_{f2} = 535$.

Pump work for higher pressure stage 5 to 6 = $(30 - 2.5)\ 10^2 \times 0.00106$ = **2.9 kJ/kg**

Pump work for low pressure side = $(30 - 0.04)\ 10^2 \times 0.00104 = 3$ kJ/kg.

The mass flow after point 3 for every kg of steam inlet is **0.841 kg.**

Mass flow through first feed pump is 0.159 kg only.

$$\therefore \quad \text{Efficiency of the cycle} = \frac{(2803 - 2370) + 0.841\ (2717 - 2125) - 0.841 \times 3 - 0.159 \times 2.9}{0.841\ (2803 - 124) + 0.159\ (2803 - 537.9)} = \mathbf{0.355}$$

To determine the efficiency without separation :

Pump work $= (30 - 0.04)\ 10^2 \times 0.00104 = $ **3 kJ/kg.**

$$\text{Cycle efficiency} = \frac{(2803 - 1863) - 3}{2803 - 12} = \mathbf{0.35}$$

Dryness $x_4 = $ **0.717.**

Separation does not affect the efficiency very much. But if the moisture is separated, the dryness of exit steam is very low and the turbine blades will wear out very rapidly.

Problem 22.4. *The steam at 90 bar and 480°C is supplied to a steam turbine. The steam is reheated to its original temperature passing the steam through reheater at 12 bar. The expansion after the reheating takes place to condenser pressure of 0.07 bar. Find the efficiency of reheat cycle and work output if the flow of steam is 1 kg/sec. Neglect the pressure loss in the system and assume the expansion through the turbine is isentropic. Do not neglect the pump work.*

Solution. The cycle of operation is represented on $T - s$ diagram as shown in Fig. Prob. 22.4 (*a*) and the reheating and expansion processes are represented on $H - s$ diagram as shown in Fig. Prob. 22.4 (*b*).

We can locate the points 1, 2, 3 and 4 on $h - s$ chart as conditions are known and we can find the enthalpies at different points from $h - s$ chart.

$\therefore$ $h_1 = 3333.5$ kJ/kg $\quad h_2 = 2815$ kJ/kg

$h_3 = 3425.5$ kJ/kg $\quad h_4 = 2364$ kJ/kg

h_{f5} (saturated liquid heat at 0.07 bar) = 161.8 kJ/kg (from steam table).

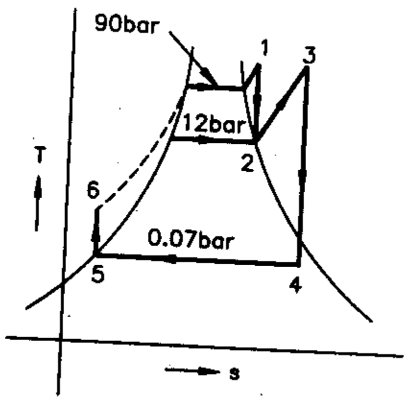


Fig. Prob. 22.4 (*a*).

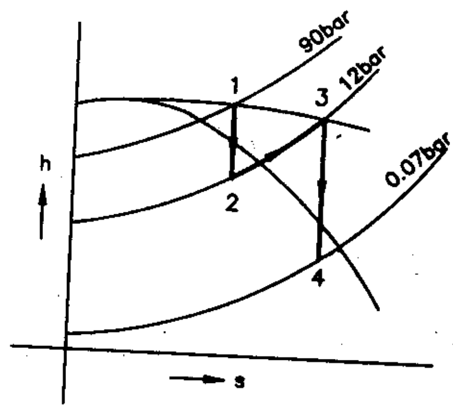


Fig. Prob. 22.4 (*b*).

Enthalpy at point '6' is given by $h_6 = h_5 +$ work done by pump

$$= h_{f5} + \frac{v_{sw}\,(p_1 - p_3)}{J}$$

where $p_1 = 90$ bar, $p_3 = 0.07$ bar

and v_{sw} (specific volume of saturated water at 0.07 bar) = 0.001013 m^3/kg

$$\therefore \quad h_6 = h_5 + \frac{0.001013\,(90 - 0.07) \times 10^5}{1000 \times 1}$$

$$= 161.8 + 9.1 = \mathbf{171\ kJ/kg.}$$

Net work done per kg of steam $= (h_1 - h_2) + (h_3 - h_4) - W_p$

where W_p is the work done by the pump per kg of steam flow.

$\therefore$ W_n (net work done) $= (3333.5 - 2815) + (3425.5 - 2364) - 9.1$

$= \mathbf{1570\ kJ/kg}$

Power generating capacity of the plant

$= 1570 \times 1 = \mathbf{1570\ kW}$

Heat supplied per kg of steam is given by

$$H_s = (h_1 - h_{f6}) + (h_3 - h_2)$$

$$= (3333.5 - 171) + (3425.5 - 2815)$$

$$= 3162.5 + 610.5 = \mathbf{3773\ kJ/kg}$$

$\therefore$ Efficiency of the cycle $= \dfrac{W_n}{H_s} = \dfrac{1570}{3773} = 0.416 = \mathbf{41.6\%}.$

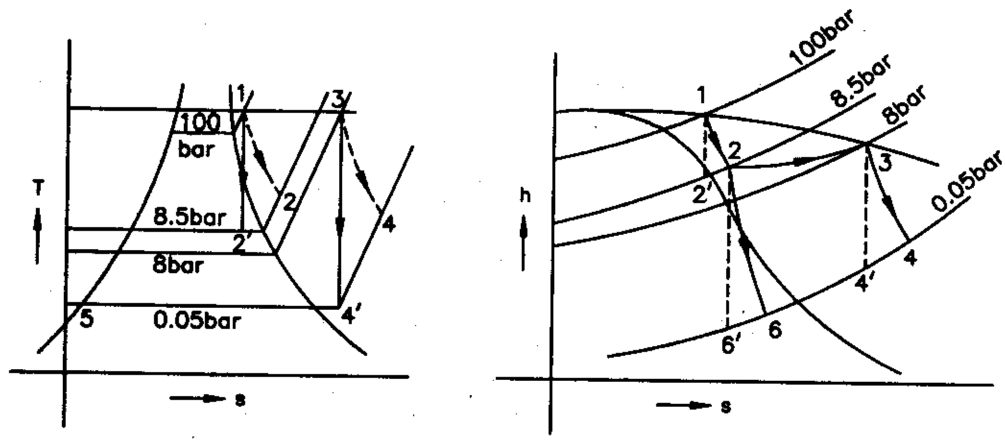
Problem 22.5. *The steam at 100 bar and 500°C expands in the turbine upto 8.5 bar with an isentropic efficiency of 80%. The steam is then reheated to original temperature and then it expands in the lower stage of the turbine upto the condenser pressure of 0.05 bar. The isentropic efficiency of the lower stage of the turbine is 85%. Find the thermal efficiency of the cycle assuming the pressure loss in the reheater of 0.5 bar.*

If the expansion of the steam is allowed to continue in the lower part of the turbine with an isentropic efficiency of 75% without reheating, then find the thermal efficiency of the cycle.

Neglect the pump work in both cases.

Solution. The cycle of operation is represented on $T-s$ diagram as shown in Fig. Prob. 22.5 (*a*) and the reheating and expansion processes are represented on $h-s$ chart as shown in Fig. Prob. 22.5 (*b*).

We can locate the points 1, 2', 3, 4' as the conditions at these points are known and we read the enthalpies from $h-s$ chart.



(*a*) (*b*)

Fig. Prob. 22.5.

$\therefore \quad h_1 = 3377$ kJ/kg $\qquad h_2' = 2750$ kJ/kg

$h_3 = 3478$ kJ/kg $\qquad h_4' = 2738$ kJ/kg

The isentropic efficiency of the expansion 1-2 is 80% as given in problem.

$$\therefore \quad \frac{h_1 - h_2}{h_1 - h_2'} = 0.8$$

$$\therefore \quad \frac{3377 - h_2}{3377 - 2750} = 0.8$$

$$\therefore \quad h_2 = \mathbf{2875\ kJ/kg}$$

The isentropic efficiency of the expansion 3-4 is 85% as given in problem.

$$\therefore \quad \frac{h_3 - h_4}{h_3 - h_4'} = 0.85.$$

$$\therefore \quad \frac{3478 - h_4}{3478 - 2738} = 0.85$$

$$\therefore \quad h_4 = \mathbf{2849\ kJ/kg}$$

The efficiency of the cycle
$$= \frac{(h_1 - h_2) + (h_3 - h_4)}{(h_1 - h_{f5}) + (h_3 - h_2)} = \frac{(3377 - 2875) + (2478 - 2849)}{(3377 - 137) + (3478 - 2875)}$$

$$= \frac{1131}{3843} = 0.29 = \mathbf{29\%}$$

If there is no reheating then the process of expansion in the second stage of the turbine is shown in Fig. Prob. 22.5 (*b*) by the process 2-6.

We can read h_6' from the $h - s$ diagram.

$\therefore$ $h_6' = \mathbf{2305}$ **kJ/kg.**

The isentropic efficiency of the process 2-6 is 75% as given in problem.

$$\therefore \quad \frac{h_2 - h_6}{h_2 - h_6'} = 0.75$$

$$\therefore \quad \frac{2875 - h_6}{2875 - 2306} = 0.75$$

$$\therefore \quad h_6 = 2452 \text{ kJ/kg}$$

The thermal efficiency of the cycle without reheating

$$= \frac{(h_1 - h_2) + (h_2 - h_6)}{(h_1 - h_{f5})} = \frac{(3377 - 2875) + (2875 - 2452)}{3377 - 137} = 0.285 = \mathbf{28.5\%}.$$

Problem 22.6. *The steam at 215 bar and 500°C is supplied to a steam turbine. It expands to 40 bar and 280°C. It is then reheated to 490°C and expands to 8 bar and 270°C. The steam coming out from the second expansion of the turbine is further reheated to 490°C and expands to condenser pressure of 0.07 bar. The steam entering into the condenser is dry and saturated. Assuming the pressure loss in first heater of 1 bar and in second heater 0.5 bar, find (a) the thermal efficiency of the cycle and power developed if the flow of steam is 10 kg/sec through the turbine, (b) the isentropic efficiencies of the expansion stages.*

Neglect the pump work.

Solution. The expansion and reheating processes of steam are represented on $h - s$ diagram as shwn in Fig. Prob. 22.6.

We can locate the points 1, 2, 3, 4, 5 and 6 on $h - s$ diagram as the conditions are known. We can also locate the points 2′, 4′ and 6′ as the processes 1-2′, 3-4′ and 5-6′ are isentropic.

We can read the enthalpies at all points from $h - s$ diagram.

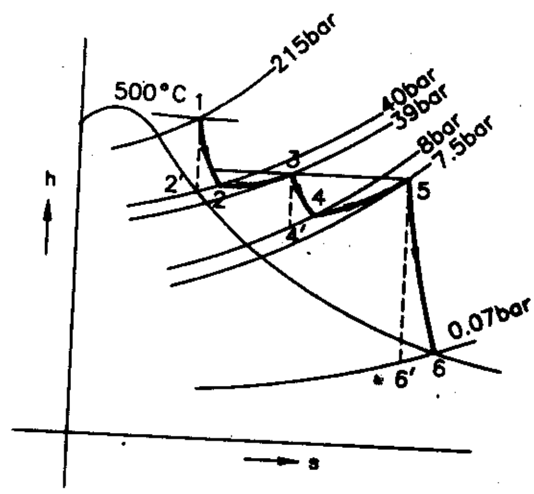


Fig. Prob. 22.6.

$\therefore$ $h_1 = 3234$ kJ/kg $\quad h_2' = 2822$ kJ/kg

$h_2 = 2910$ kJ/kg $\quad h_3 = 3435$ kJ/kg

$h_4' = 2977$ kJ/kg $\quad h_4 = 2998$ kJ/kg

$h_5 = 3473$ kJ/kg $\quad h_6' = 2444$ kJ/kg

$h_6 = 2578$ kJ/kg

h_{f7} (saturated liquid heat at 0.07 bar)
= 162 kJ/kg (from steam table).

Work done per kg of steam

$$= (h_1 - h_2) + (h_3 - h_4) + (h_5 - h_6) = (3234 - 2910) + (3435 - 2998) + (3473 - 2578)$$
$$= 324 + 437 + 895 = \mathbf{1656\ kJ/kg}$$

Heat supplied per kg of steam

$$= (h_1 - h_7) + (h_3 - h_2) + (h_5 - h_4) = (3234 - 162) + (3435 - 2910) + (3473 - 2998)$$
$$= 3072 + 525 + 475 = \mathbf{4072\ kJ/kg}$$

$\therefore$ Thermal efficiency of the cycle

$$= \frac{1656}{4072} = 0.407 = \mathbf{40.7\%}$$

Power developed by the plant

$$= \frac{1656 \times 10}{1} = \mathbf{16560\ kW}$$

(*a*) Isentropic efficiency of first stage

$$= \frac{h_1 - h_2}{h_1 - h_2'} = \frac{3234 - 2910}{3234 - 2822} = \frac{324}{412} = \mathbf{78.6\%}$$

(*b*) Isentropic efficiency of second stage

$$= \frac{h_3 - h_4}{h_3 - h_4'} = \frac{3435 - 2998}{3435 - 2977} = \frac{437}{458} = \mathbf{95.4\%}$$

(*c*) Isentropic efficiency of third stage

$$= \frac{h_5 - h_6}{h_5 - h_6'} = \frac{3473 - 2578}{3473 - 2444} = \frac{895}{1029} = \mathbf{87\%}.$$

Problem 22.7. *The steam at 100 bar and 400°C is supplied to a steam turbine. The steam expands to 20 bar and then the steam is withdrawn from the turbine and reheated to 380°C in the reheater. It further expands to condenser pressure of 0.05 bar. The pressure loss in the reheater and piping is 1 bar and the isentropic efficiency of both the expansions is 80%. Assuming the transmission efficiency from turbine to generator 98% and the generator efficiency of 95%, find the quantity of steam circulated per hour through the system if the generator output is 60 MW.*

Neglect the pump work.

Solution. The expansion and reheating processes of steam are represented on $h - s$ diagram as shown in Fig. Prob. 22.7.

We can locate the points 1 and 3 as the conditions at these points are known. The points 2′ and 4′ can also be located as the processes 1-2′ and 3-4′ are isentropic.

We can find the enthalpies at different points directly from $h - s$ diagram.

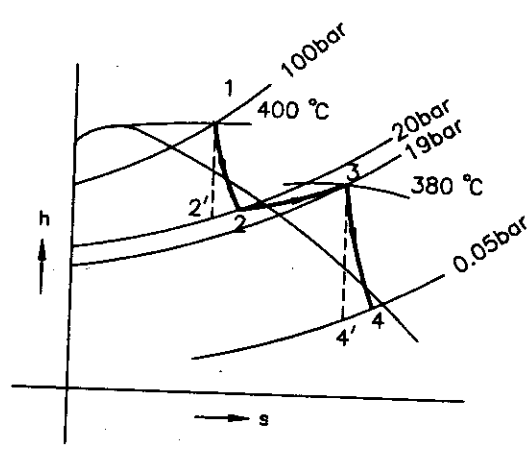


Fig. Prob. 22.7.

$\therefore \quad h_1 = 3093$ kJ/kg $\qquad h_2' = 2734$ kJ/kg

$h_3 = 3202$ kJ/kg $\qquad h_4' = 2157$ kJ/kg.

The isentropic efficiency of the expansions 1-2 and 3-4 is 80%.

$$\therefore \qquad \frac{h_1 - h_2}{h_1 - h_2'} = \frac{h_3 - h_4}{h_3 - h_4'} = 0.8.$$

$$\therefore \qquad \frac{3093 - h_2}{3093 - 2734} = \frac{3202 - h_4}{3202 - 2157} = 0.8.$$

$$\therefore \qquad h_2 = 2805 \text{ kJ/kg and } h_4 = 2366 \text{ kJ/kg.}$$

Work done per kg of steam $= (h_1 - h_2) + (h_3 - h_4)$

$= (288) + (836) =$ **1124 kJ/kg.**

If the mass of steam passing through the turbine per sec is m_s kg then,

$$[(m_s) \times 1124]\, \eta_t \times \eta_g = 60 \times 1000$$

$$\therefore \qquad m_s = \frac{60 \times 1000}{1124 \times 0.98 \times 0.95} = \mathbf{57.3 \text{ kg/sec}}$$

$$= \frac{57.3 \times 3600}{1000} = \mathbf{206.4 \text{ tons/hr.}}$$

Problem 22.8. *The steam at 50 bar and 400°C is supplied to a two stage steam turbine. The steam is first expanded in H.P. turbine to 5 bar. The steam is reheated at 5 bar to 250°C using the 0.96 dry steam from the boiler at 50 bar. The reheated steam is passed through L.P. stage. The steam from L.P. turbine is exhausted to a condenser at 0.03 bar. The isentropic efficiency of both the stages is 80%. Assuming the mechanical efficiency of 99% and generator efficiency of 96%, find the mass of steam generated by the boiler per kW-hour. Assume that the heating steam in the reheater gives up heat by condensing only. Neglect the pump work and other losses.*

Solution. The arrangement of the components is shown in Fig. Prob. 22.8 (*a*).

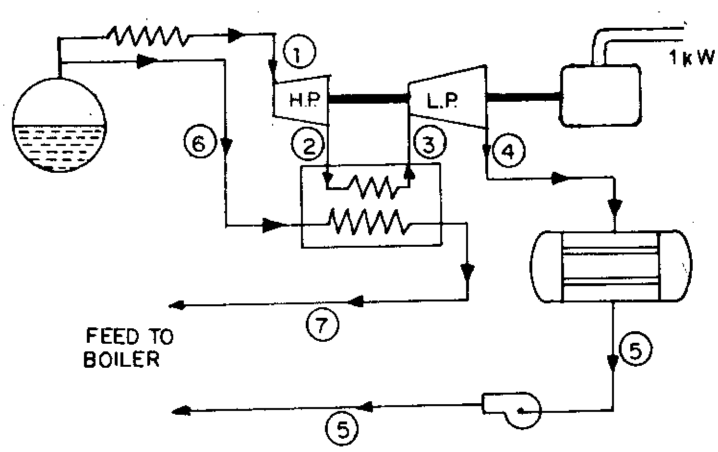


Fig. Prob. 22.8 (*a*).

The conditions of steam passing through different components are represented on $h - s$ diagram as shown in Fig. Prob. 22.8 (*b*).

The points '1' and '3' are located on $h - s$ chart as conditions are known. The points 2′ and 4′ can also be located as the processes 1-2′ and 3-4′ are isentropic. The enthalpies at different points can be taken from $h - s$ chart directly.

$\therefore \quad h_1 = 3198$ kJ/kg $\quad h_2' = 2675$ kJ/kg

$h_3 = 2955$ kJ/kg $\quad h_4' = 2153$ kJ/kg.

$h_1 - h_2 = 0.8\,(h_1 - h_2')$

$= 0.8\,(3198 - 2675) = 418.4$ kJ/kg

$\therefore \quad h_2 = 3198 - 418.4 = \mathbf{2779.6}$ **kJ/kg**

$h_3 - h_4 = 0.8\,(h_3 - h_4')$

$= 0.8\,(2955 - 2153) = 0.8 \times 802$

$= 641.6$ kJ/kg

Work done per kg of steam,

$= 418.4 + 641.6 = \mathbf{1060}$ **kJ/kg**

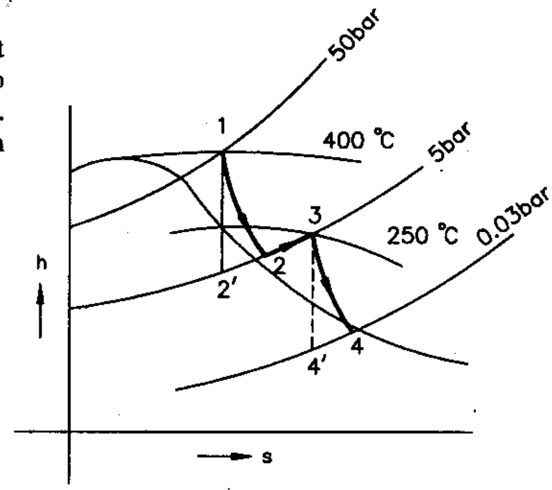


Fig. Prob. 22.8 (*b*).

The work used out of 1060 kJ for the generation of electricity

$= 1060 \times 0.99 \times 0.96 = \mathbf{1007}$ **kJ/kg.**

Steam generated in the boiler per 1 kW power generation

$$= \frac{1000}{1007} \times 3.6 = \mathbf{3.58 \text{ kg/kW-hr.}}$$

The latent heat of steam lost for reheating the steam in reheater per kg

$= xh_{fg} = 0.96 \times 1643.5 = \mathbf{1578}$ **kJ/kg.**

The steam used in the reheater for reheating the steam per kW-hr energy generated

$$= \frac{3.58\,(h_3 - h_2)}{xh_{fg}} = \frac{3.58\,(2955 - 2779.6)}{1578} = \mathbf{0.398 \text{ kg.}}$$

$\therefore$ Steam generated by the boiler per kW-hr output from the generator

$= 3.58 + 0.398 = \mathbf{3.978}$ **kg.**

Problem 22.9. *In a steam plant operating between 100 bar and 0.035 bar, the steam is initially superheated to 500°C. The steam after expansion to saturation conditions at an intermediate pressure, is reheated to the initial temperature and then expanded to the condenser pressure. Determine the final dryness and efficiency of the cycle and compare them with simple Rankine cycle.*

Solution.

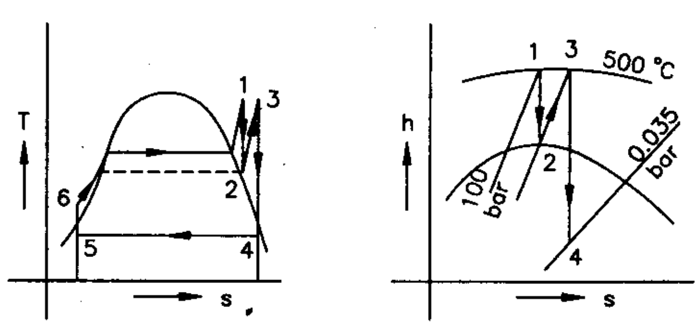


Fig. Prob. 22.9.

From tables : Use of the *h-s* (Mollier) chart is recommended for solving the problem as otherwise a lot of interpolation to determine the condition 2 is neceessary.

From the chart : $h_1 = 3373$, $h_2 = 2778$, $h_3 = 3478$,
$h_4 = 2322$, $x_4 = 0.907$, $h_{f5} = 112$, (from steam tables)

and pump work as calculated $= \mathbf{10\ kJ/kg}$

$$\therefore \text{ Cycle efficiency} = \frac{(h_1 - h_2) + (h_3 - h_4) - W_p}{(h_1 - h_{f6}) + (h_3 - h_2)} = \frac{(3373 - 2778) + (3478 - 2322) - 10}{(3373 - 112) + (3478 - 2778)} = \mathbf{0.44}.$$

There is a marginal increase in efficiency from 42.8% to 44% but a considerable increase in dryness is obtained, from 76.3% to 90.7%. This shows that reheating is mainly for the reduction of moisture at the last stages of the turbine. Work done per kg is also improved from 1390.3 to 1441 kJ/kg.

Problem 22.10. *In a steam power plant, steam is supplied at 100 bar and 500°C and condenser pressure is 0.05 bar. The steam is reheated after passing through I-stage turbine to its original temperature and then expanded to condenser pressure. The reheating is carried out when the steam becomes just dry-saturated. Assume isentropic efficiency of each stage expansion as 85%. Find the efficiency of the plant and the steam required per hour if the capacity of the plant is 100 MW. Neglect the feed pump work.*

(b) If there is no reheat and expansion is in one stage with 80% isentropic efficiency then find efficiency and steam consumption per hour.

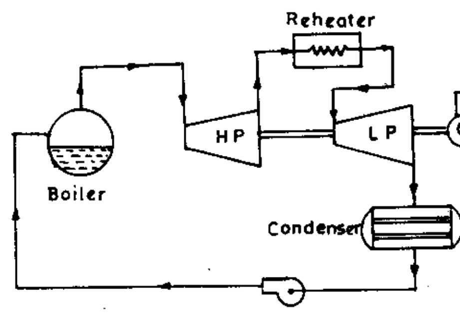


Fig. Prob. 22.10 (a).

Solution. The arrangement of the components and their respective processes are shown in Fig. Prob. 22.10 (*a*), Fig. Prob. 22.10 (*b*) and Fig. Prob. 22.10 (*c*). Now locate the point '2' as pressure (100 bar) and temperature (500°C) are given. Draw a vertical line through 2 till it cuts saturation line at 3. Then locate 3′ as

2-3′ = 0.85 of 2-3 (given)

Then draw a horizontal line through 3′ which cuts the pressure line (11.5 bar) at 3″.

Locate the point 4 as it is the cross-point of 500°C temp. line and 11.5 bar pressure line. Then draw a vertical line through point 4 till it cuts 0.05 bar pressure line and locate the point 5. Then locate 5′ as

4-5′ = 0.85 of 4-5 (given)

Draw horizontal line through 5′ which cuts the 0.05 bar pressure line and locate 5″.

The 2-3″ and 4-5″ represents the conditions of the steam passing through H.P. and L.P. turbines.

After marking the points, the enthalpies are noted from *h-s* chart at all points as listed below

$h_2 = 3370$, $h_3'' = 2860$, $h_4 = 3500$, $h_5'' = 2530$

and $h_{f6} = 137.8$ (from steam tables)

All values are in kJ/kg

W_{t1} (H.P. turbine) $= h_2 - h_3'' = 3370 - 2860 = 510$ kJ/kg

W_{t2} (L.P. turbine) $= h_4 - h_5'' = 3500 - 2530 = 970$ kJ/kg

Q_b (heat supplied in the boiler) $= h_2 - h_{f6} = 3370 - 137.8 = 3232.2$ kJ/kg

Q_r (heat supplied in the reheater) $= h_4 - h_3'' = 3500 - 2860 = 640$ kJ/kg

$$\eta = \frac{W_{t1} + W_{t2}}{Q_b + Q_r} = \frac{510 + 970}{3232.2 + 640} = \frac{1480}{3872.2} = \mathbf{0.382 = 38.2\%}$$

If m_s is the mass flow of the steam per second then the power developed

$= m_s\,(W_{t1} + W_{t2})$ kJ/sec $= m_s\,(W_{t1} + W_{t2})$ kW

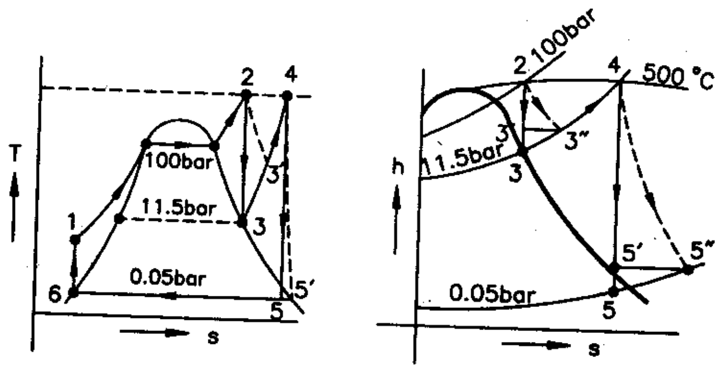


Fig. Prob. 22.10 (b).

$$\therefore \quad m_s (W_{t1} + W_{t2}) = 1000 \times 100$$

$$m_s = \frac{1000 \times 100}{510 + 970} = \frac{1000 \times 100}{1480} = 67.57 \text{ kg/sec}$$

$$= \frac{67.57 \times 3600}{1000} \text{ tons/hr}$$

$$= \mathbf{243.25 \text{ tons/hr}}$$

(*b*) In this case the processes are shown in Fig. Prob. 22.10 (*c*).

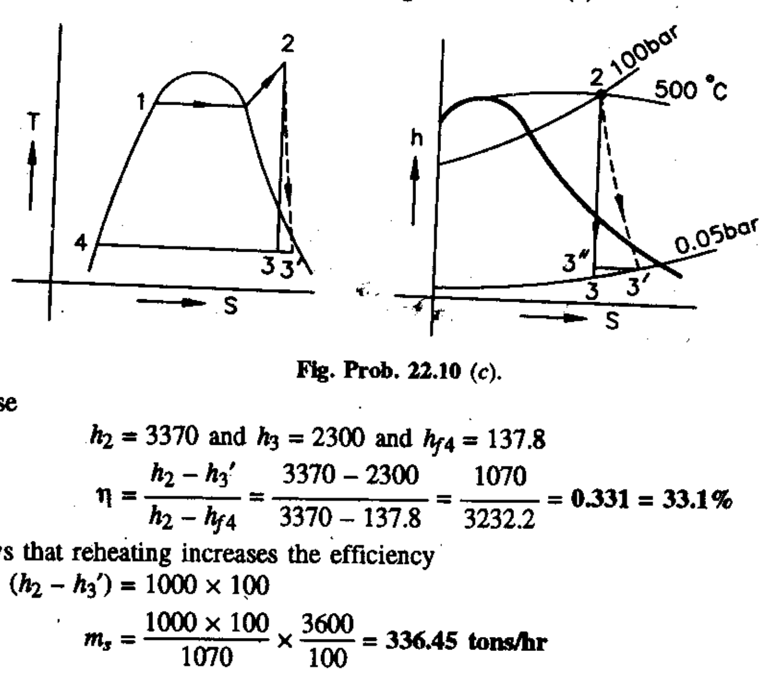


Fig. Prob. 22.10 (c).

In this case

$$h_2 = 3370 \text{ and } h_3 = 2300 \text{ and } h_{f4} = 137.8$$

$$\therefore \quad \eta = \frac{h_2 - h_3'}{h_2 - h_{f4}} = \frac{3370 - 2300}{3370 - 137.8} = \frac{1070}{3232.2} = \mathbf{0.331 = 33.1\%}$$

This shows that reheating increases the efficiency

$$m_s (h_2 - h_3') = 1000 \times 100$$

$$\therefore \quad m_s = \frac{1000 \times 100}{1070} \times \frac{3600}{100} = \mathbf{336.45 \text{ tons/hr}}$$

This also shows that steam consumption also increases if reheating is not used.

Problem 22.11. *A Regenerative steam power plant generates 27000 kW energy from the electric generator directly coupled to steam turbine. The steam at 60 bar and 450°C is supplied to the steam turbine. The condenser vacuum is 707.5 mm of Hg. The steam is bled from the steam turbine at 3 bar. The heating of the feed water is done in the direct contact heater. Assuming the turbine efficiency of each portion of expansion is 87%, find (a) the steam bled per kg of steam supplied to the turbine. (b) The steam generated per hour if the 10% of the generator output is used to run the pumps, (c) the overall efficiency of the plant* *if the boiler efficiency is 90% and alternator efficiency 95% and mechanical efficiency from turbine to generator is 98%. Neglect the pump work in calculating the input to the boiler.*

Solution. The arrangement of the components is shown in Fig. Prob. 22.11 (*a*). The conditions of the fluid passing through the components are represented on $T-s$ and $h-s$ diagrams as shown in Fig. Prob. 22.11 (*b*) and Fig. Prob. 22.11 (*c*).

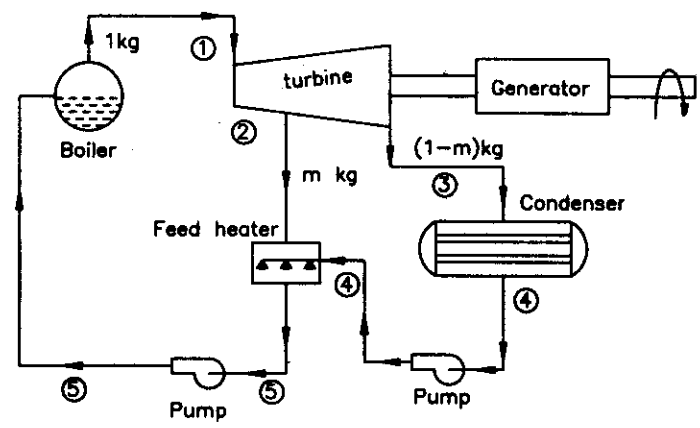
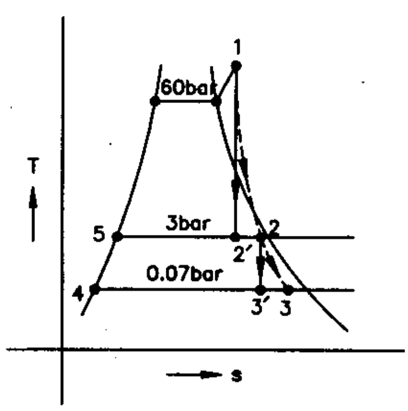


Fig. Prob. 22.11 (*a*).



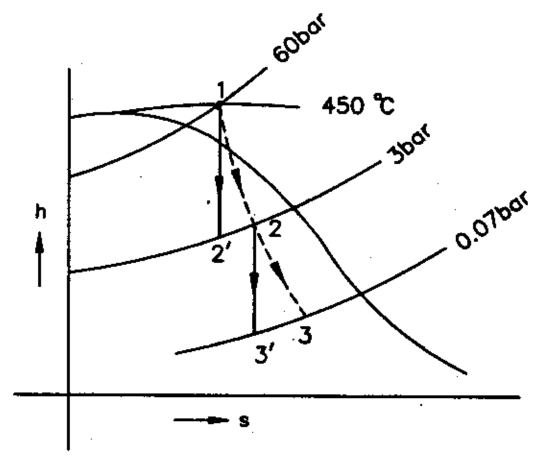


Fig. Prob. 22.11 (*b*). **Fig. Prob. 22.11** (*c*).

The conditions of the fluid entering and leaving the pump are shown by the same point as the rise in temperature due to pump work is neglected.

$$\text{The condenser pressure} = \frac{(760 - 707.5)}{760} \times 1.013 = 0.07 \text{ bar.}$$

We can locate the point 1 as the pressure and temperature of the steam are known.

$$\therefore \qquad h_1 = 3296 \text{ kJ/kg from } h-s \text{ chart.}$$

Draw the vertical line through point '1' till it cuts the 3 bar pressure line, then locate the point 2'.

$\therefore \quad h_2' = 2606$ kJ/kg

Now $\quad \dfrac{h_1 - h_2}{h_1 - h_2'} = 0.87$ as given in the problem.

$$\therefore \quad \frac{3296 - h_2}{3296 - 2606} = 0.87$$

$\therefore \quad h_2 = 2695$ kJ/kg.

Locate the point '2' on $h - s$ chart as pressure and enthalpy are known and then draw a vertical line through point '2' till it cuts the 0.07 bar pressure line and then locate the point 3.

$\therefore \quad h_3' = 2163$ kJ/kg

$$\frac{h_2 - h_3}{h_2 - h_3'} = 0.87$$

$$\therefore \quad \frac{2695 - h_3}{2695 - 2163} = 0.87$$

$\therefore \quad h_3 = 2232$ kJ/kg

We can find the saturated liquid heats at points 4 and 5 from steam tables as the pressures are known.

$\therefore \quad h_{f4} = 162$ kJ/kg and $h_{f5} = 558$ kJ/kg

For finding the value of m (bled steam) per kg we can use the following equation :

$$m\,(h_2 - h_{f5}) = (1 - m)\,(h_{f5} - h_{f4})$$

$$\therefore \quad m\,(2695 - 558) = (1 - m)\,(558 - 162)$$

$\therefore \quad m = 0.156$ kg/kg of steam generated

Work developed per kg of steam in the turbine

$= (h_1 - h_2) + (1 - m)\,(h_2 - h_3)$

$= (3296 - 2695) + (1 - 0.156)\,(2695 - 2232) =$ **991.8 kJ/kg**

Actual work developed by the trubine

$= \dfrac{27000}{0.95 \times 0.98} =$ **29001 kW**

$\therefore$ Steam generated per hour $= \dfrac{29001}{991.8} = 29.24$ kg/sec $= \dfrac{29.24 \times 3600}{1000} =$ **105.3 ton/hr**

(*c*) Net power available deducting pump power

$= 27000 \times (1 - 0.1) =$ **24300 kW**

$\therefore$ Heat supplied in the boilet $= \dfrac{105.3 \times 1000\,(h_1 - h_{f5})}{0.9}$

$= \dfrac{105.3 \times 1000}{0.9}\,(3296 - 588) =$ **88010 kW**

The overall efficiency of the plant $= \dfrac{\text{Net power available}}{\text{Heat supplied in the boiler}}$

$= \dfrac{24300}{88010} = 0.276 =$ **27.6%.**

Problem 22.12. *The steam at 35 bar and 300°C is supplied to a steam turbine. The exhaust pressure of the turbine is 0.1 bar. A single bleed is taken between the high pressure and low pressure of turbine at 1.5 bar for regenerative feed heating. The isentropic efficiency for both sections of the turbine is 80%. The temperature of the bleed condensate coming out of heat exchanger is 10°C lower than the temperature of the bleed steam. Find (a) bleed steam per kg of steam supplied to the steam turbine and (b) the thermal efficiency of the plant. Neglect the losses and pump work. The condensate coming out from heat exchanger and condenser are led to the hot well.*

Solution. The arrangement of the components is shown in Fig. Prob. 22.12 (*a*).

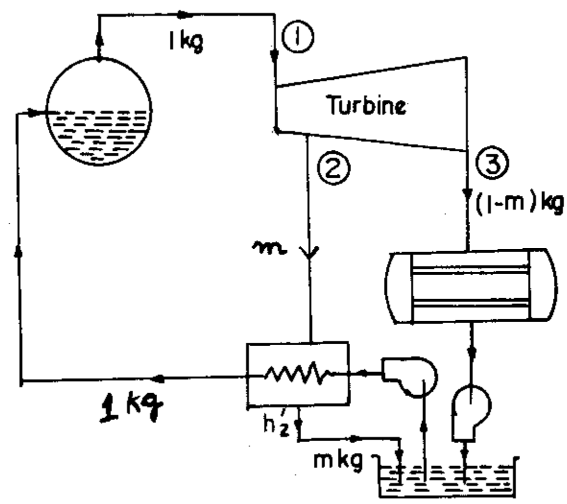


Fig. Prob. 22.12 (*a*).

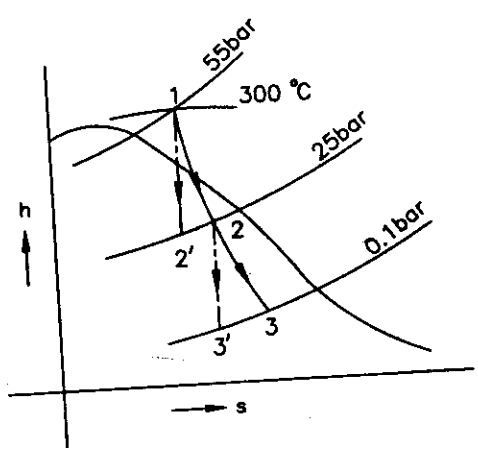


Fig. Prob. 22.12 (*b*).

The expansion processes of steam passing through the turbine are represented on *h* – *s* diagram as shown in Fig. Prob. 22.12 (*b*).

We can mark the point 1 and 2′ as the condition at point '1' is known and process 1-2′ is isentropic. The point 2 can also be marked at $(h_1 - h_2)$ is 80% of $(h_1 - h_2')$. Similarly, we can locate the point 3. The enthalpies at different points can be directly taken from $h - s$ diagram.

$\therefore \quad h_1 = 2970$ kJ/kg, $h_2 = 2504$ kJ/kg, $h_3 = 2197$ kJ/kg.

h_{f2} (saturated liquid enthalpy at 1.5 bar)
$= 264$ kJ/kg (from steam table)

h_{f2}' (actual enthalpy of liquid coming out of heat exchanger)
$= 264 - 42 = 422$ kJ/kg as the temperature of coming out condensate is 10°C less than the temperature of bleed steam.

h_{f3} (saturated liquid enthalpy at 0.1 bar) = 190 kJ/kg (from steam tables).

Using the heat balance at hot well

$$(1 - m)\, h_{f3} + m h_{f2}' = 1 \times h_{f4} \qquad \text{...(1)}$$

where h_{fh} is the enthalpy of condensate at hot well.

Using the heat balance at heat exchanger

$$m\,(h_2 - h_{f2}') = (h_{f2}' - h_{f4}) \qquad \text{...(2)}$$

The temperature of heated condensate (or enthalpy) is same as the temperature of condensate formed from bleed steam for ideal heat exchanger.

Adding the equations (1) and (2), we get

$$(1 - m)\, h_{f3} + m h_{f2}' + m\,(h_2 - h_{f2}') = h_{f2}'.$$

$$\therefore \quad m\,(h_2 - h_{f3}) = h_{f2}' - h_{f3}$$

$$\therefore \quad m = \frac{h_{f2}' - h_{f3}}{h_2 - h_{f3}} = \frac{422 - 190}{2504 - 190} = \frac{232}{2314} \approx \mathbf{0.1\ kg.}$$

Work done per kg of steam supplied to the turbine

$$= (h_1 - h_2) + (1 - m)\,(h_2 - h_3)$$
$$= (2970 - 2504) + (1 - 0.1)\,(2504 - 2197)$$
$$= 466 + 0.9 \times 307 = 111.5 + 66.15 = \mathbf{742\ kJ/kg.}$$

Heat supplied per kg of steam $= h_1 - h_{f2}' = 2970 - 422 = \mathbf{2548\ kJ/kg.}$

$\therefore$ Thermal efficiency of the plant $= \dfrac{742}{2548} = \mathbf{29.2\%}.$

Problem 22.13. *The steam at 40 bar and 300°C is supplied to a steam turbine. The steam is bled for feed heating at 14 bar and 3.4 bar. The condenser pressure is 0.07 bar. Find (a) the optimum mass of bled steam at each stage per kg of steam supplieed to the H.P. turbine and (b) the cycle efficiency.*

Assume the turbine efficiency of each portion of the expansion is 80% and feed water leaving each heater is raised to the steam temperature entering the heater. The heaters used are indirect heaters and the condensate coming out from first heater is passed through second heater and the condensate coming out from second heater is passed to the condenser outlet.

Neglect the pump work.

Solution. The arrangement of the components is shown in Fig. Prob. 22.13 (*a*) and the conditions of the fluid passing through the components are shown on $T - s$ and $h - s$ diagrams as shown in Fig. Prob. 22.13 (*b*) and Fig. Prob. 22.13 (*c*).

We can locate the point '1' as pressure and temperature of the steam are known, we can also find out the values of h_1 and h_2' from $h - s$ chart directly as $1 - 2'$ is isentropic expansion.

$\therefore \quad h_1 = 2953$ kJ/kg, $h_2' = 2738$ kJ/kg.

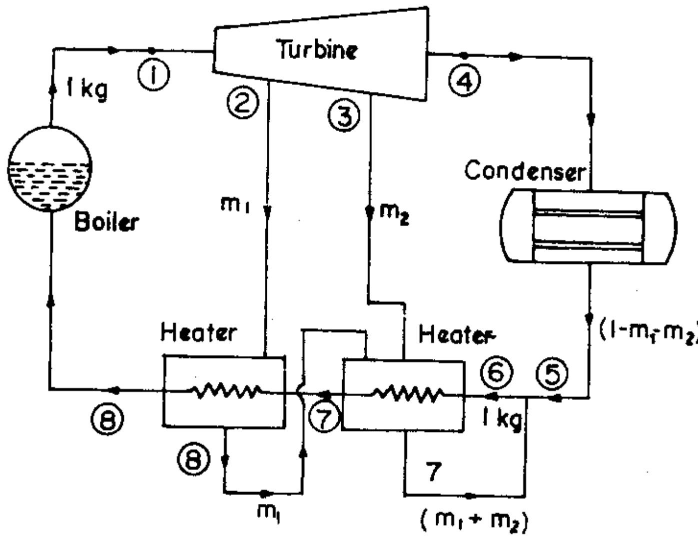


Fig. Prob. 22.13 (*a*).

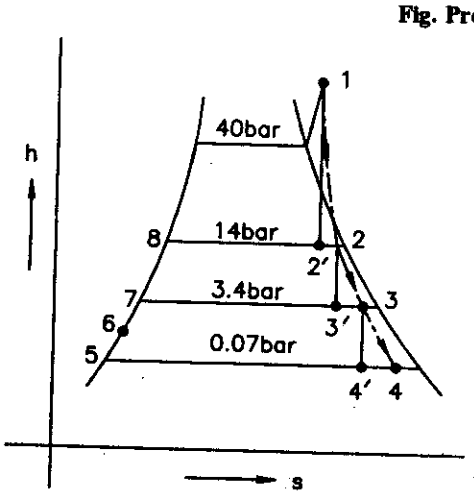


Fig. Prob. 22.13 (*b*).

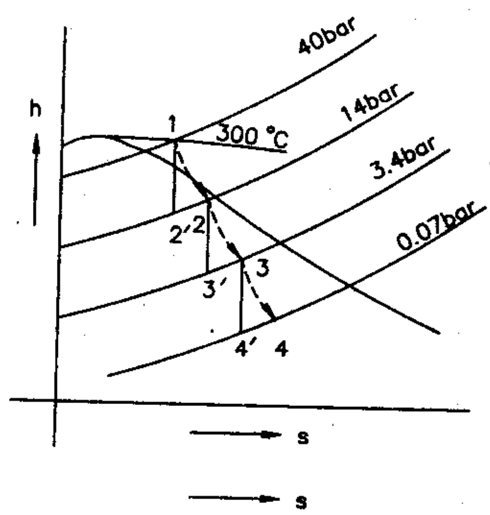


Fig. Prob. 22.13 (*c*).

The efficiency of expansion for the first portion is given by

$$\frac{h_1 - h_2}{h_1 - h_2'} = 0.8 \text{ as given in problem.}$$

$$\therefore \quad \frac{2953 - h_2}{2953 - 2738} = 0.8$$

$$\therefore \quad \mathbf{h_2 = 2738\ kJ/kg.}$$

Again we can locate the point '2' as pressure and enthalpy are known and we can find out the value of h_3' from $h - s$ chart drawing vertical line through point '2' till it cuts the pressure line of 3.4 bar.

$$\therefore \quad h_3' = 2529 \text{ kJ/kg}$$

The expansion efficiency of the second portion is given by

$$\frac{h_2 - h_3}{h_2 - h_3'} = 0.8$$

$$\therefore \quad \frac{2738 - h_3}{2738 - 2529} = 0.8$$

$$\therefore \quad h_3 = 2579 \text{ kJ/kg.}$$

We can locate the point '3' similar to point '2' and then we can find out the value of h_4' from $h - s$ chart.

$$\therefore \quad h_4' = 2040 \text{ kJ/kg}$$

Again
$$\frac{h_3 - h_4}{h_3 - h_4'} = 0.8$$

$$\therefore \quad \frac{2579 - h_4}{2579 - 2040} = 0.8$$

$$\therefore \quad h_4 = \mathbf{2148.5 \text{ kJ/kg.}}$$

We can find out the saturated liquid heats at points 5, 7 and 8 from the steam table as the pressures are known.

$\therefore$ $h_{f5} = 162$ kJ/kg, $h_{f7} = 575$ kJ/kg, $h_{f8} = 825$ kJ/kg.

For finding the values of m_1 and m_2, we can use the following equations :

$$m_1 (h_2 - h_{f8}) = 1 (h_{f8} - h_{f7})$$

$$\therefore \quad m_1 = \frac{h_{f8} - h_{f7}}{h_2 - h_{f8}} = \frac{825 - 575}{2738 - 825} = \mathbf{0.1307 \text{ kJ/kg of steam}}$$

Similarly, $m_2 (h_3 - h_{f7}) = 1 (h_{f7} - h_{f6})$

and $(m_1 + m_2) h_{f7} + (1 - m_1 - m_2) h_{f5} = 1.h_{f6}$

Substituting all the values in the above equations,

$$m_2 (2579 - 575) = (575 - h_{f6})$$

$$2004 \, m_2 = 575 - h_{f6} \quad \text{...(a)}$$

and
$$(0.1307 + m_2) \times 575 + (1 - 0.1307 - m_2) \times 162 = h_{f6}$$

$$\therefore \quad 413 \, m_2 = h_{f6} - 215.8 \quad \text{...(b)}$$

Solving the above two equations, we get

m_2 = **0.15 kJ/kg** of steam supplied to turbine and h_{f6} = **216 kJ/kg**

(*b*) The efficiency of the cycle is given by

$$\eta = \frac{(h_1 - h_2) + (1 - m_1)(h_2 - h_3) + (1 - m_1 - m_2)(h_3 - h_4)}{(h_1 - h_{f8})}$$

$$= \frac{(2953 - 2738) + (1 - 0.1307)(2738 - 2579) + (1 - 0.1307 - 0.15) \times (2579 - 2148.5)}{(2953 - 825)}$$

$$= \frac{215 + 140 + 314}{2128} = \frac{669}{2189} = 0.305 = \mathbf{30.5\%}.$$

Problem 22.14. *The steam at 30 bar and at 1 bar 350°C is supplied to a steam turbine. Two regenerative heaters are used by extracting the steam at 6 bar. The steam exhaust pressure is 0.07 bar. The feeds coming out from first and second heaters are saturated liquids at corresponding pressures. The feed drain from the first heater is passed into the second heater through a steam trap and the combined drains from the second heater are pumped by a drain pump into the feed pipe after the second heater.*

If the internal power developed by the turbine is 10 MW, find the quantity of steam extracted at each point of the turbine per minute.

Take the isentropic efficiency of each stage 80%.

Neglect the pump works.

Solution. The arrangement of the components is shown in Fig. Prob. 22.14 (*a*) and the expansion processes are represented on $h - s$ diagram as shown in Fig. Prob. 22.14 (*b*).

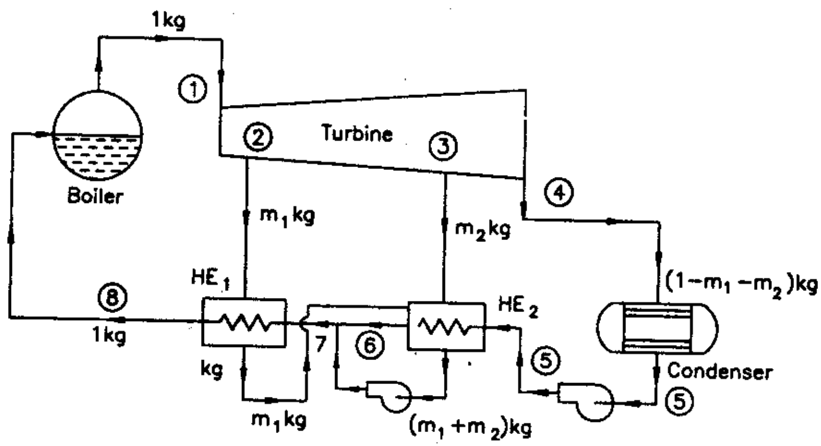


Fig. Prob. 22.14 (*a*).

First we will find the value of m_1 and m_2 per kg of steam supplied to the turbine.

We can mark the points 1 and 2′ as condition at point 1 is known and process 1-2′ is isentropic. We can mark the point 2 as the $(h_1 - h_2)$ is 80% of $(h_1 - h_{2'})$ and similarly we can mark the points 3 and 4 also. We can also find the total heats at points 1, 2, 3 and 4 from the $h - s$ chart directly.

$\therefore\quad h_1 = 3106$ kJ/kg $\quad h_2 = 2811$ kJ/kg

$h_3 = 2560$ kJ/kg $\quad h_4 = 2259$ kJ/kg

Using the heat balance equation at HE_1

$$m_1\,(h_2 - h_{f2}) = 1\,(h_{f8} - h_{f7}) \qquad ...(1)$$

Using the heat balance equation at HE_2

$$m_2(h_3 - h_{f3}) + m_1(h_{f2} - h_{f3}) = (1 - m_1 - m_2)\,(h_{f6} - h_{f5}) \qquad ...(2)$$

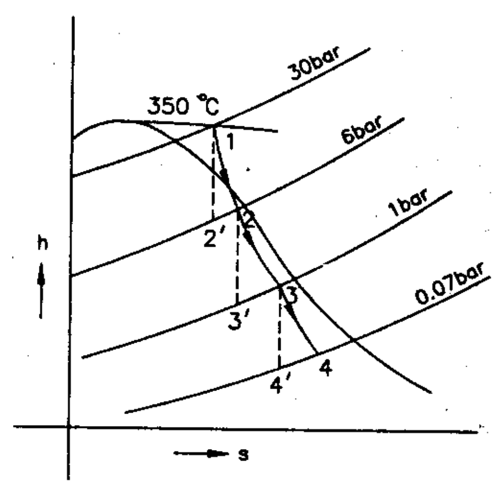


Fig. Prob. 22.14 (*b*).

At the point '*P*' two streams of feed at different temperatures are meeting

$$(1 - m_1 - m_2)\,h_{f6} + (m_1 + m_2)\,h_{f3} = 1 \times h_{f7} \qquad ...(3)$$

Now $h_2 = 2811$ kJ/kg $\quad h_3 = 2560$ kJ/kg

$h_{f2} = 777$ kJ/kg (at 6 bar)

$h_{f3} = 415$ kJ/kg (bar)

$h_{f5} = 162$ kJ/kg (saturated liquid heat at 0.07 bar from steam table)

$h_{f8} = h_{f2} = 666$ kJ/kg
$h_{f6} = h_{f3} = 415$ kJ/kg.

It is always assumed that the temperature of feed (formed from feed steam) is always equal to the temperature of feed (coming from condenser) to be heated.

From equations (1) and (3), we can eliminate h_7 and we get

$$m_1 (h_2 + h_{f3} - h_{f2} - h_{h6}) + m_2 (h_{f3} - h_{f6}) = h_{f8} - h_{f6}$$

$$\therefore \quad m_1 (h_2 - h_{f2}) = h_{f8} - h_{f6} \text{ as } h_{f3} = h_{f6} \quad ...(4)$$

Substituting the enthalpy values in equations (2) and (4), we get

$$504\, m_1 + 2398\, m_2 = 253$$

and

$$2145\, m_1 = 251$$

Solving the above two equations, we get

$m_1 = 0.177$ kg/kg of steam generated and
$m_2 = 0.081$ kg/kg of steam generated

The power developed per kg of steam supplied to the turbine

$= (h_1 - h_2) + (1 - m_1)(h_2 - h_3) + (1 - m_1 - m_2)(h_3 - h_4)$

$= (3106 - 2811) + (1 - 0.117)(2811 - 2560) + (1 - 0.177 - 0.081) \times (2560 - 2259)$

$= 295 + 0.883 \times 251 + 0.802 \times 301 = 295 + 221.6 + 241.4 =$ **758 kJ/kg.**

If m_s is the quantity of steam passing through the turbine per minute, then

$$(m_s) \times 758 = 10 \times 1000$$

$\therefore$ $m_s = 13.2$ kg/sec = **792 kg/min.**

$\therefore$ Quantity of steam extracted per minute at 6 bar pressure

$= 792 \times 0.117 =$ **92.5 kg/min**

Quantity of steam extracted per minute at 1 bar pressure

$= 792 \times 0.081 =$ **64.2 kg/min**

$\therefore$ Capacity of feed extraction pump = 92.5 + 64.2 = **156.7 kg/min.**

Problem 22.15. *The steam at 40 bar and 400°C is supplied to the steam turbine. The steam is bled for regenerative heating at 2 bar and 0.5 bar. The condenser pressure is 0.05 bar. Assuming the isentropic efficiencies in first, second and third stage of the turbine 75%, 80% and 85% respectively and neglecting the pump work and heat losses, find (a) steam bled for regenerative heaters per kg of steam supplied to turbine (b) power developed by the turbine in kW if the steam flow in 10 kg/sec (c) the thermal efficiency of the cycle.*

In each heater, the feed water is heated to saturation temperature of steam at heater pressure and bled steam in the heater is pumped into the feed line immediately ahead of each heater.

Solution. The arrangement of the components is shown in Fig. Prob. 22.15 (*a*). The conditions of steam passing through the turbine are represented on $h - s$ chart as shown in Fig. Prob. 22.15 (*b*).

Locate the point '1' as the condition of the point '1' is known and then draw a vertical line through 1 till it cuts the pressure line of 2 bar and locate the point 2′.

$\therefore$ $h_1 = 3210$ kJ/kg, $h_2' = 2562$ kJ/kg

$\therefore$ $h_1 - h_2 = 0.75 (h_1 - h_2')$ as given in problem $= 0.75 (3210 - 2562) = 486$ kJ

$\therefore$ $h_2 = 3210 - 486 =$ **2724 kJ/kg**

Now locate the point '2' as pressure and enthalpy at that point are known and then draw the vertical line through point '2' till it cuts the pressure line of 0.5 bar and locate the point 3′.

$\therefore$ $h_3' = 2508$ kJ/kg

$\therefore$ $h_2 - h_3 = 0.8 (h_2 - h_3')$ as given in problem $= 0.8 (2724 - 2508) = 216$

$\therefore$ $h_3 = 2724 - 216 =$ **2508 kJ/kg.**

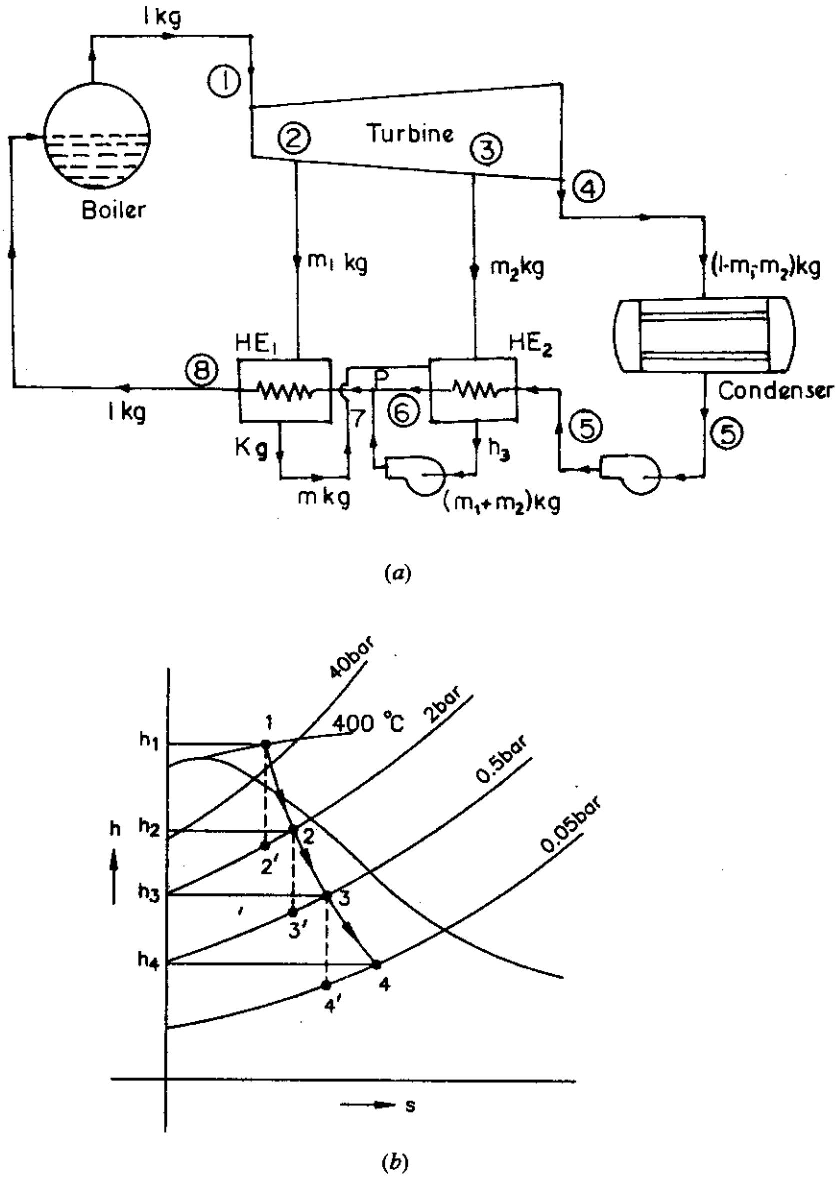


Fig. Prob. 22.15.

Now locate the point '3' as pressure and enthalpy at that point are known and draw the vertical line through point '3' till it cuts the pressure line of 0.05 bar and then locate the point 4'.

$\therefore \qquad h_4' = 2232$ kJ/kg.

$\therefore \qquad h_3 - h_4 = 0.85\,(h_3 - h_4')$ as given in problem = 0.85 (2508 – 2232) = 235

$h_4 = 2508 - 235 =$ **2273 kJ/kg.**

$h_{f10} = h_{f8} = 502$ kJ/kg (enthalpy of saturated water at 2 bar from steam table)

$h_{f9} = h_{f6} \approx 339$ kJ/kg (enthalpy of saturated water at 0.5 bar from steam table)

$h_{f5} = 136$ kJ/kg (enthalpy of saturated water at 0.05 bar from steam table)

In the above calculations the effect of pump work is neglected.

Considering enthalpy balance at HE_1

$$m_1\,(h_2 - h_{f10}) = (1 - m_1)\,(h_{f8} - h_{f7}) = (1 - m_1)\,(h_{f8} - h_{f6}) \qquad ...(1)$$

As $\qquad h_{f6} = h_{f9}$ therefore $h_{f7} = h_{f6} = h_{f9}$

Considering the enthalpy balance at HE_2

$$m_2 \cdot (h_3 - h_{f9}) = (1 - m_1 - m_2)(h_{f6} - h_{f5}) \qquad ...(2)$$

Substituting the enthalpy values in equation (1), we get,

$$m_1 (2724 - 502) = (1 - m_1)(502 - 339)$$

$$2222\, m_1 = 163 - 163\, m_1$$

$$\therefore \qquad m_1 = \frac{163}{2385} = \mathbf{0.0684\ kJ/kg.}$$

Substituting the values in equation (2), we get,

$$m_2 (2508 - 339) = (1 - 0.0684 - m_2)(339 - 136)$$

$$2569\, m_2 = 189 - 203\, m_2$$

$$\therefore \qquad m_2 = \frac{189}{2772} = \mathbf{0.0682\ kJ/kg.}$$

Work done per kg of steam supplied to the turbine

$= (h_1 - h_2) + (1 - m_1)(h_2 - h_3) + (1 - m_1 - m_2)(h_3 - h_4)$

$= (3210 - 2724) + (1 - 0.0614) \times (2724 - 2508)) + (1 - 0.0684 - 0.0682) \times (2508 - 2273)$

$= 486 + 203 + 204 = \mathbf{893\ kJ/kg.}$

Power developed by the turbine = 893 × 10 = **8930 kW.**

Heat supplied per kg of steam

$= h_1 - h_{f10} = h_1 - h_{f8}$ as $h_{f8} = h_{f10}$

$= 3210 - 502 = 2708$ kJ/kg.

$\therefore$ Thermal efficiency of the cycle $= \frac{893}{2708} = 0.33 = \mathbf{33\%}.$

Problem 22.16. *The steam at 70 bar and 459°C is supplied to a steam turbine. The steam expands in H.P. turbine to 25 bar. The steam coming out of H.P. turbine is reheated to 420°C. The steam then expands to condenser pressure of 0.07 bar in the L.P. cylinder. A single stage feed heating is used to increase the temperature of feed to 179°C. The isentropic efficiencies of H.P., I.P. and L.P. cylinders are 78.5%, 83% and 83% respectively.*

Neglecting the pump work and other losses, find the thermal efficiency of the cycle.

(b) Quantity of steam supplied per minute to the plant if the power developed by the plant is 20 MW. Take mechanical efficiency of the turbine 85%, transmission efficiency 95% and generation efficiency 95%.

(c) Overall efficiency of the plant.

Solution. The arrangement of the components of the system is shown in Fig. Prob. 22.16 (*a*).

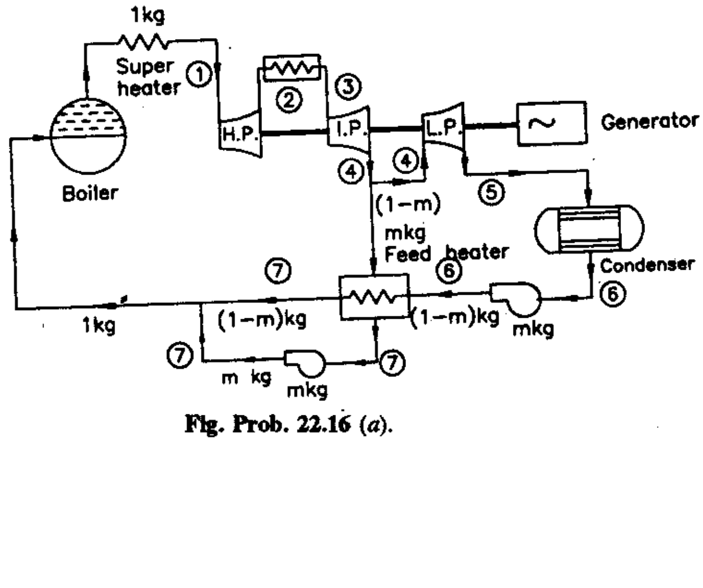


Fig. Prob. 22.16 (*a*).

The expansion and reheating processes of steam passing through the different components are shown on $h-s$ diagram as shown in Fig. Prob. 22.16 (*b*).

The minimum pressure at which the bleeding of steam is necessary is the saturation pressure corresponding to the saturation temperature of 179°C.

The saturation pressure at 179°C saturation temperature

= 10 bar (from steam table).

∴ The bleeding of the steam is done at 10 bar pressure.

We can locate the points '1' and 2' as the condition of 1 is known and process 1-2' is isentropic. We can also read the enthalpies at 1 and 2' directly from $h-s$ chart.

∴ $h_1 = 3280$ kJ/kg $\quad h_2' = 2997$ kJ/kg

As the isentropic efficiency of the expansion 1-2 is 78.5%

∴ $h_1 - h_2 = 0.785\,(h_1 - h_2') = 0.785\,(3280 - 2997) = 222$

∴ $h_2 = 3280 - 222 = 3058$ kJ/kg

Now locate the point '2' on $h-s$ chart

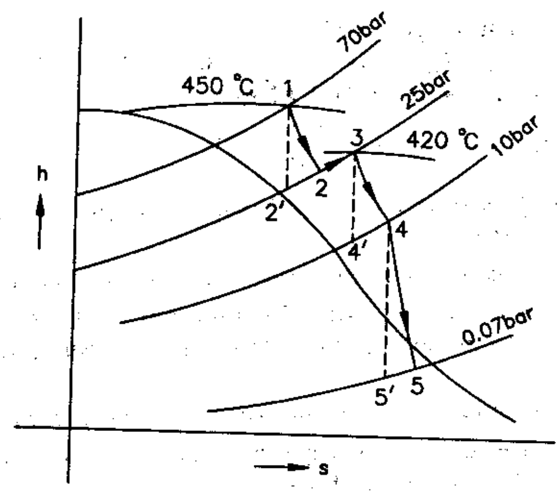


Fig. Prob. 22.16 (*b*).

We can locate the point '3' on $h-s$ chart as condition of 3 is known and point '4' can also be located as the process 3-4' is isentropic.

∴ $h_3 = 3277$ kJ/kg $\qquad h_4' = 3020$ kJ/kg

As the isentropic efficiency of the expansion 3-4 is 83%

∴ $h_3 - h_4 = 0.83\,(h_3 - h_4') = 0.83\,(3277 - 3020) = 213$

∴ $h_4 = 3277 - 213 = 3064$ kJ/kg

Now locate the point 4 on $h-s$ chart and draw a vertical line through 4 till it cuts the pressure line of 0.07 bar and locate the point 5'.

∴ $h_5' = 2220$ kJ/kg

As the isentropic efficiency of the expansion 4-5 is 83%

∴ $h_4 - h_5 = 0.83\,(h_4 - h_5') = 0.83\,(3064 - 2220) = 700.5.$

∴ $h_5 = 3064 - 700.5 = 2363.5$ kJ/kg

For finding the bled steam 'm' for feed heating, we can use the enthalpy balance for feed-heater

$\therefore \quad m\,(h_4 - h_{f7}) = (1 - m)\,(h_{f7} - h_{f6})$

h_{f6} (saturated liquid heat at 0.07 bar) = 162 kJ/kg from steam table

h_{f7} (saturated liquid heat at 10 bar) = 758 kJ/kg from steam table

$\therefore \quad m\,(3064 - 758) = (1 - m)\,(758 - 162)$

$\therefore \quad 2306\,m = 596 - 596\,m,$

$\therefore \quad m = \dfrac{596}{2902} = \mathbf{0.205\ kg.}$

Work done per kg of steam supplied to the H.P. turbine,

$$= (h_1 - h_2) + (h_3 - H_4) + (1 - m)\,(h_4 - h_5)$$
$$= (3280 - 3058) + (3277 - 3064) + (1 - 0.205)\,(3064 - 2363.5)$$
$$= 222 + 213 + 0.795 \times 700.5 = \mathbf{992\ kJ/kg}$$

Energy converted for generating the electrical energy per kg of steam supplied to H.P. turbine

$$= 992 \times 0.85 \times 0.95 \times 0.95 = \mathbf{761\ kJ}$$

$\therefore$ Steam generated per minute $= \left(\dfrac{20 \times 1000}{1}\right) \times \dfrac{1}{761} = \mathbf{26.3\ kg/sec = 1577\ kg/min.}$

Heat supplied per kg of steam generated

$$= (h_1 - h_{f7}) + (h_3 - h_2) = (3280 - 758) + (3277 - 3058)$$
$$= 2522 + 219 = \mathbf{2741\ kJ/kg.}$$

$\therefore$ Thermal efficiency of the cycle $= \dfrac{992}{2741} = 0.361 = \mathbf{36.1\%}$

Problem 22.17. *A small power plant of 500 kW capacity is supplied with a steam at 30 bar and 300°C. The condenser pressure is 0.04 bar :*

(a) Determine the Rankine efficiency and steam generation rate of boiler in kg/sec.

(b) If a feedheater of mixing type is used at 7 bar pressure, then find the efficiency of the cycle and steam generation rate of boiler in kg per sec. Also find the amount of cooling water if the rise in temperature is limited 5°C.

In both cases, assume the expansions throughout are isentropic and pump works are neglected.

Solution. (*b*) The arrangement of the components of the system is shown in Fig. Prob. 22.17 (*a*) and the processes are shown on $T - s$ and $h - s$ charts.

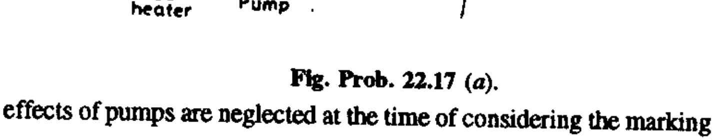


Fig. Prob. 22.17 (*a*).

Note. The effects of pumps are neglected at the time of considering the marking on points on component line diagram.

The process 4-5-6 is represented as shown on $h - s$ chart.

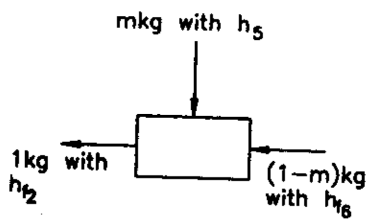


Fig. Prob. 22.17 (*b*).

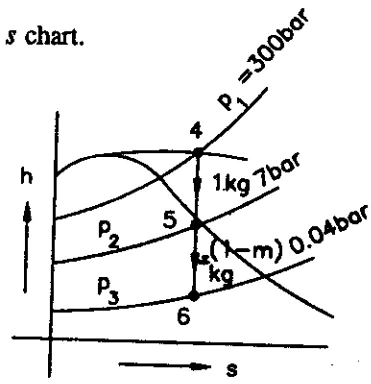


Fig. 22.17 (*c*)

Considering energy balance for the feed heater

$$\therefore \quad 1 \, . \, h_{f2} = m \, . \, h_5 + (1 - m) \, . \, h_{f1} \qquad ...(a)$$

From h – s chart

$$h_4 = 3000, \; h_5 = 2700 \text{ and } h_6 = 1970$$

and h_{f1} (from steam table 0.04 at bar) = 121.4

h_{f2} (from steam table 7 bar) = 697

Now substituting the values in equation (*a*)

$$697 = m \times 2700 + (1 - m)\, 121.4$$
$$= m\,(2700 - 121.4) + 121.4$$
$$\therefore \quad m = \frac{697 - 121.4}{2700 - 121.4} = \frac{575.6}{2577.6} = \mathbf{0.223 \text{ kg}}$$

Now work done in the turbine per kg of steam generated in the boiler is given by

$$W = 1\,(h_4 - h_5) + (1 - m)\,(h_5 - h_6)$$
$$= (3000 - 2700) + (1 - 0.223)\,(2700 - 1970)$$
$$= 300 + 567.2 = \mathbf{867.2 \text{ kJ/kg}}$$

Q_s (heat supplied) $= h_4 - h_{f2} = 3000 - 697 = 2303$ kJ/kg

$$(i) \quad \therefore \quad \eta_s = \frac{W}{Q_s} = \frac{867.2}{2303} = 0.376 = \mathbf{37.6\%}$$

(*ii*) If m_s is the steam generated per second, then

$m_s \, . \, W = m_s \times 867.2$ W is the capacity

$$\therefore \quad m_s \times 867.2 = 500$$
$$\therefore \quad m_s = \frac{500}{867.2} = 0.5765 \text{ kg/sec} = \mathbf{2075.6 \text{ kg/hr}}$$

(*iii*) Heat gained by water = Heat lost by steam in condenser

$$C_{pw}\, m_w\, (\Delta T) = (h_6 - h_{f1})\, m_s\, (1 - m)$$
$$4.2\, m_w \times 5 = (1970 - 121.4) \times 0.5765 \times (1 - 0.223)$$
$$\therefore \quad m_w = \frac{0.5765 \times 1848.6 \times 1.777}{4.2 \times 5} = \mathbf{39.3 \text{ kg/sec}}$$

(*a*) If there is no feed heater, then

$$W = h_4 - h_6 = 3000 - 1970 = 1030 \text{ kJ/kg}$$
$$Q_s = h_4 - h_{f1} = 3000 - 121.4 = 2878.6 \text{ kJ/kg}$$
$$\therefore \quad \eta = \frac{W}{Q_s} = \frac{1030}{2878.6} = 0.358 = \mathbf{35.8\%}$$

For finding steam generation rate in the boiler

$$m_s \times 1030 = 500$$

$$\therefore \quad m_s = \frac{500}{1030} = 0.485 \text{ kg/sec} = \mathbf{1747.5 \text{ kg/hr}}$$

For finding cooling water

$$4.2 \times m_w \times 5 = 0.5765\ (1970 - 121.4)$$

$$\therefore \quad m_w = \mathbf{50.57 \text{ kg/sec.}}$$

Problem 22.18. *A small steam power plant is supplied with a steam at 40 bar and 500°C. The condenser is maintained at 0.040 bar. One regenerative non-mixing type heater is provided at 10 bar. Determine the percentage of steam bled at 10 bar and thermal efficiency of the plant. If the boiler generation rate is 50 tons per hour, find out the generating capacity of the plant assuming mechanical* $\eta_m = 85\%$ *and electrical generation* $\eta_g = 95\%$.

Neglect the pump work and assume the expansion throughout is isentropic.

Solution. The arrangement of the components is shown in Fig. Prob. 22.18 (*a*) and processes are shown on $T - s$ and $h - s$ diagrams in Fig. Prob. 22.18 (*b*) and Fig. Prob. 22.18 (*c*).

The process 1-3 is shown on $h - s$ chart and the point 2 is marked as pressure (10 bar) is known. The enthalpies at points 1, 2 and 3 are noted from $h - s$ chart.

$$h_1 = 3400 \text{ kJ/kg},\ h_2 = 3050 \text{ kJ/kg and } h_3 = 2150 \text{ kJ/kg}$$

The enthalpies at 4 and 5 are noted from steam tables as they are saturated liquid enthalpies at 0.04 bar and 10 bar respectively,

$$h_{f4} = 121.4,\ h_{f5} = 762.6. \text{ And } h_{f6} = h_{f5} \text{ as } 5 - 6 \text{ is throttling}$$

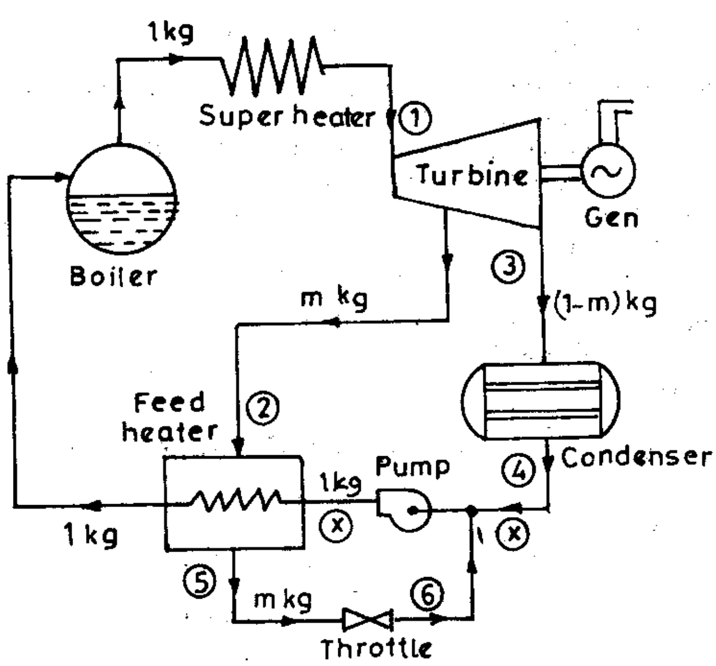


Fig. Prob. 22.18 (*a*), (*b*).

Now m kg of water at h_{f6} and $(1 - m)$ kg of water at h_{f4} are mixed together at point x.

$$\therefore \quad m\, h_{f6} + (1 - m) h_{f4} = m h_{fm} \qquad ...(i)$$

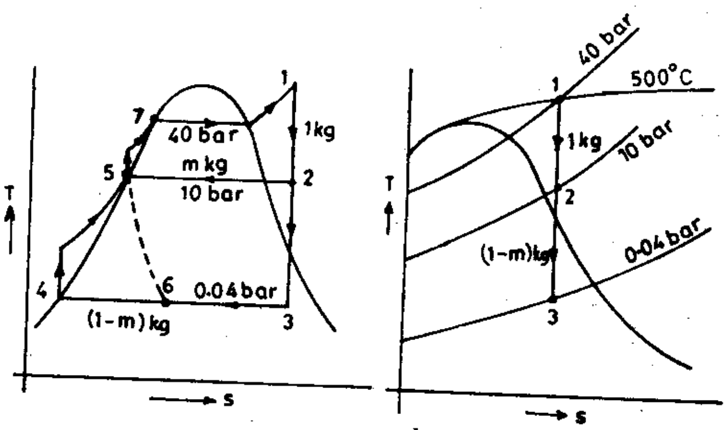


Fig. Prob. 22.18 (*c*).

where h_{fm} is the enthalpy after mixing.

Considering enthalpy equilibrium for the given feed-heater, we can write

$$m\ (h_2 - h_{f5}) = 1\ (h_{f5} - h_{fm}) \qquad ...(ii)$$

In the above two equations m and h_{fm} are not known.

$$m \times 762.6 + (1 - m)\ 121.4 = m.h_{fm} \qquad ...(a)$$

$$m\ (3050 - 762.6) = (762.6 - h_{fm}) \qquad ...(b)$$

Solving above two equations, we get

$$m = 0.22 \text{ and } h_{fm} = 263$$

Note. Students are advised to solve these equations by trial and error, first assuming the value of m, find h_{fm} from equation (*a*) and using the value of h_m and m, check whether equation (*b*) gets balanced or not.

Therefore bled steam = 22% of boiler generation.

The work developed per kg of steam generated in the boiler is given by

$$W = 1\ (h_1 - h_2) + (1 - m)\ (h_2 - h_3)$$
$$= (3400 - 3050) + (1 - 0.22)(3050 - 2150)$$
$$= 350 + 792 = \mathbf{1142\ kJ/kg}$$

Heat supplied in the boiler per kg of steam generated

$$Q_s = (h_1 - h_{f5}) = (3400 - 762.6) = \mathbf{2637.4\ kJ/kg}$$

$$\therefore \quad \eta = \frac{W}{Q_s} = \frac{1142}{2637.4} = 0.433 = \mathbf{43.3\%}$$

Power developed = Work developed in turbine $\times \eta_m \times \eta_g$

$= m_b(W) \times \eta_m \times \eta_g$ where m_b is the steam generated in the boiler per sec

$$= \frac{50 \times 1000}{3600} \times 1142 \times 0.85 \times 0.95$$

$= 12808$ kW $\approx$ **12.8 MW.**

Problem 22.19. *A steam power plant of 100 MW capacity is supplied with steam at 80 bar and 50°C of superheat. The condenser pressure is maintained at 0.05 bar. The plant is equipped with one mixing type regenerator and one reheater. The steam is extracted at 7 bar pressure for feed heating and remainder is reheated to 350°C. Determine*

(*i*) *The percentage of steam bled.*

(*ii*) *Thermal efficiency of the cycle.*

(*iii*) *Boiler generating rate per hour in tons of steam.*

Assume expansions are isentropic and neglect the pressure losses in the system and pump work.

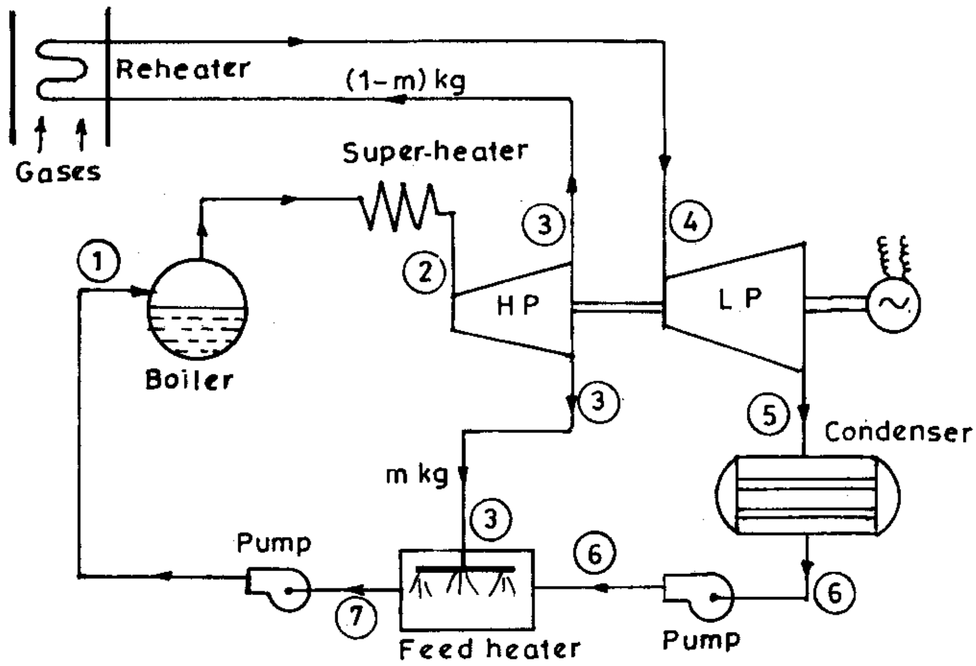


Fig. Prob. 22.19 (*a*).

Solution. The arrangement of the system is shown in Fig. Prob. 22.19 (*a*) and processes on $T-s$ and $h-s$ are represented as shown in Fig. Prob. 22.19 (*b*) and Fig. Prob. 22.19 (*c*).

As the degree of superheat is 50°C, that means its temperature

$= 300 + 50 = 350°C$

The points are marked on $h-s$ diagram and enthalpies are noted as given below

$$h_2 = 2990,\ h_3 = 2530,\ h_4 = 3170 \text{ and } h_5 = 2180$$

The enthalpies at points 6 and 7 are noted from steam table

$$h_{f6} = 138 \text{ and } h_{f7} = 697$$

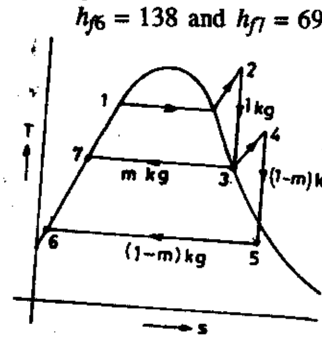


Fig. Prob. 22.19 (*b*).

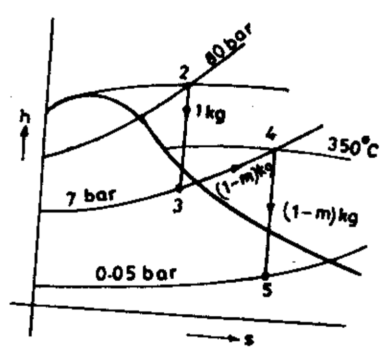


Fig. Prob. 22.19 (*c*).

Considering enthalpy balance at feed heater, we can write

$$m\,(h_3 - h_{f7}) = (1 - m)\,(h_{f7} - h_{f6})$$

$\therefore \quad m\,(3170 - 697) = (1 - m)\,(697 - 138)$

$2473\,m = (1 - m)\,559$

$\therefore \quad m = \mathbf{0.184}$

$\therefore$ Percentage of bled steam = **18.4%**

(*ii*) The efficiency of the power plant is given by

$$\eta = \frac{W}{Q_s}$$

where

$$W = 1\,(h_2 - h_3) + (1 - m)\,(h_4 - h_5)$$
$$= (2990 - 2530) + (1 - 0.184)\,(3170 - 2180)$$
$$= 460 + 808 = \mathbf{1268\ kJ/kg}$$
$$Q_s = (h_2 - h_{f1}) + (1 - m)\,(h_4 - h_3)$$
$$= (2990 - 697) + (1 - 0.184)\,(3170 - 2530)$$
$$= 2293 + 522 = \mathbf{2815\ kJ/kg}$$

$$\therefore \quad \eta = \frac{1268}{2815} = 0.45 = \mathbf{45\%}$$

If m_b is the mass of steam generated by the boiler in kg per second, then we can write

$m_b\,[(h_2 - h_3) + (1 - m)\,(h_4 - h_5)] = 100 \times 1000$

$\therefore \quad 1268\,m_b = 100 \times 1000$

$$\therefore \quad m_b = \frac{100 \times 1000}{1268}\ \text{kg/sec} = \frac{100 \times 1000}{1268} \times \frac{3600}{1000}\ \text{tons/hr} \approx \mathbf{284\ tons/hour}$$

Problem 22.20. *A binary power plant uses mercury and water as working fluids. Mercury cycle works between the temperature range of 556°C and 222°C. The mercury is dry and saturated at higher temperature limit. The steam cycle works between the pressure limit of 17 bar (saturation temperature at 17 bar is 203.4°C) and 745 mm of Hg vacuum in the condenser. The steam is superheated in the superheater to 383°C before entering into the steam turbine. The temperature of the feed water is raised to 203.4°C in the economiser and is evaporated to dry steam in the mercury condenser or steam generator.*

Assuming isentropic expansion of Hg vapour and steam in the respective turbines, find the following :

(*a*) *Mass of Hg required per kg of steam used.*

(*b*) *Work done by mercury turbine per kg of steam generated in the mercury condenser.*

(*c*) *Work done by steam turbine per kg of steam.*

(*d*) *The theoretical overall efficiency of cycle.*

(*e*) *If the steam flow through the steam turbine is 20 kg/sec, find the generating capacity of the plant in kW assuming the mechanical efficiency 80%, transmission efficiency 95% and generator efficiency 85% and actual work developed is 50% of theoretical in the cycles. Neglect pump work in steam as well as in mercury cycle.*

Use the data given below :

Mercury

Temperature (in 0°C)	*Heat in kJ/kg*			*Entropy in kJ/kg-K*		
	h_f	h_{fg}	h_g	s_f	$\frac{h_{fg}}{T}$	s_g
556	76	290	366	0.152	0.359	0.511
222	29	302	331	0.08	0.626	0.706

Steam

Pressure in (bar)	Heat in kJ/kg			Entropy in kJ/kg-°K	
	h_f	h_{fg}	h_g	s_f	s_g
17	874	1932	2806	2.37	6.42
0.035 bar or 745 mm of Hg vacuum	111	2453	2564	0.388	0.15

Solution. The working of the binary cycle is represented on *T-s* diagram as shown in Fig. Prob. 22.20.

The cycle 1-2-3-4-1 represents mercury cycle and cycle *a-b-c-d-a* represents steam cycle.

For finding the work done per kg of mercury, we have to first find out the condition of the mercury vapour at point 2.

Equating the entropies at point '1' and '2'

$$s_{v1} = s_{l2} + \frac{x_2 h_{fg2}}{T_2}$$

$$\therefore \quad x_2 = \frac{(s_{v1} - s_{l2})}{h_{fg2}/T_2} = \frac{0.511 - 0.08}{0.626}$$
$$= \mathbf{0.694}$$

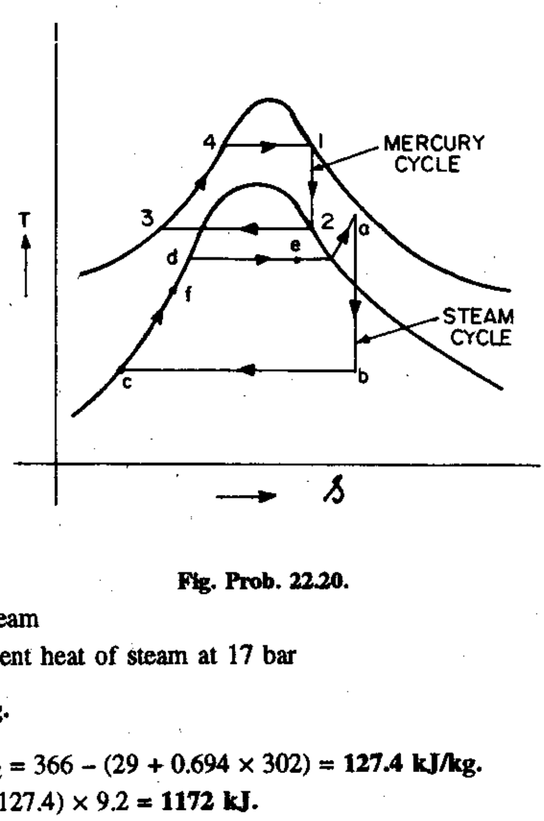


Fig. Prob. 22.20.

(*a*) m_{hg} is the mass of Hg required to be circulated in the mercury condenser to raise one kg of steam.

$\therefore$ Heat lost by mercury = Heat gained by steam

$\therefore$ $m_{hg} \times x_2 h_{fg2} = 1 \times h_{fg}$ where h_{fg} is the latent heat of steam at 17 bar

$$\therefore \quad m_{hg} = \frac{1932}{0.694 \times 302} = \mathbf{9.2\ kg.}$$

(*b*) Work done per kg of Hg vapour = $h_1 - h_2$ = 366 − (29 + 0.694 × 302) = **127.4 kJ/kg.**

Work done by 9.2 kg of Hg in Hg-turbine = (127.4) × 9.2 = **1172 kJ.**

(*c*) Again we have to find out the condition of steam at point '*b*'.

Equating the entropies at points '*a*' and '*b*'

$$s_a = s_b$$

$$s_{ga} + C_p \log_e \left(\frac{T_{sup}}{T_{sa}}\right) = s_{fb} + \frac{x_b h_{fg}}{T_b}$$

$$6.42 + 2 \log_e \left(\frac{380 + 273}{203.4 + 273}\right) = 0.388 + \frac{x_b \times 2453}{(26.5 + 273)}$$

$$\therefore \quad x_b = \mathbf{0.87}$$

Work done per kg of steam = $h_a - h_b$

= [2806 + 2.0 (383 − 203.4)] − [111 + 0.72 × 2453] = 3165 − 2245 = **920 kJ/kg**

(*d*) Total work done in the cycle = 1172 + 920 = **2092 kJ**

Heat supplied per kg of steam generated

$$= \text{Heat given to Hg in combustion chamber} + \text{Heat given to feed water} + \text{Heat supplied to superheat the steam}$$

$$= m_{hg}\,(h_{g1} - h_{f2}) + 1\,(h_d - h_c) + 1 \times C_p\,(T_{sup} - T_s)$$

$$= 9.2\,(366 - 29) + (874 - 111) + 2\,(383 - 203.4)$$

$$= 3100.4 + 763 + 360 = \mathbf{4223.4\ kJ}$$

$$\text{Overall efficiency of the cycle} = \frac{\text{Work done}}{\text{Heat supply}} = \frac{2092}{4223.4} = 0.495 = \mathbf{49.5\%}$$

(*e*) Total energy generated per sec

$$= [(20) \times 2092 \times (0.5)] \times 0.8 \times 0.95 \times 0.85$$

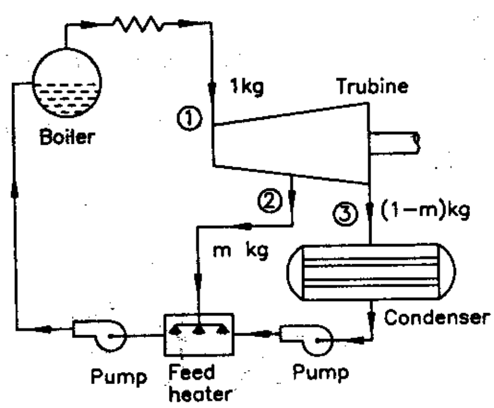
$$= 13514 \text{ kW} = \mathbf{13.514\ MW}$$

SOLVED PROBLEMS FROM UNIVERSITY QUESTION PAPERS

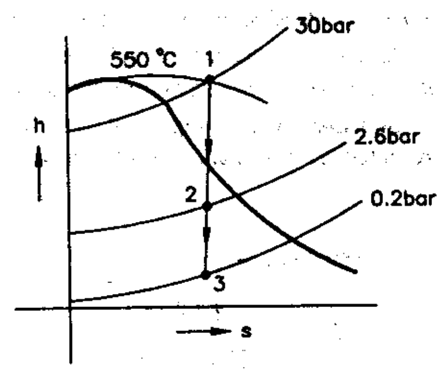
Problem 22.21. *A simple steam turbine power plant is supplied with steam at 30 bar and 550°C. Steam is bled at 2.6 bar for feed heating which is mixed with feed water coming from condenser. The condenser pressure is 0.2 bar. If the steam supplied by the boiler is 30 kg/sec, find the power generating capacity of the plant and % increase in efficiency over a simple Rankine cycle.*

Consider expansion is isentropic.

Solution. The arrangement is shown in Fig. Prob. 22.21 (*a*) and the processes are represented on *h-s* diagram as shown in Fig. Prob. 22.21 (*b*).



(*a*)



(*b*)

Fig. Prob. 22.21.

The points 1, 2 and 3 are marked on *h-s* chart as shown in Fig. Prob. 22.21 (*b*) and enthalpies are noted down

$$h_1 = 3580 \text{ kJ/kg},\ h_2 = 2870 \text{ kJ/kg},\ h_3 = 2440 \text{ kJ/kg}$$

h_{f2} (liquid heat at 2.6 bar) = 541 kJ/kg

h_{f3} (liquid heat at 0.2 bar) = 251.5 kJ/kg

Considering the feed heater

Heat lost by the steam = Heat gained by feed water

$$m\,(h_2 - h_{f2}) = (1 - m)\,(h_{f2} - h_{f3})$$
$$m\,(2870 - 541) = (1 - m)\,(541 - 251.5)$$
$$2329\,m = 289.5 - 289.5\,m$$
$$m = \frac{289.5}{2618.5} = 0.11 \text{ kg}$$

(*a*) Power developed by the turbine

$= m_s\,(h_1 - h_2) + m_s\,(1 - m)\,(h_2 - h_3)$ where m_s is steam supplied/sec.

$= 30\,[(3580 - 2870) + (1 - 0.11)\,(2870 - 2440)]$

$= 30\,(710 + 382.7) = 32780 \text{ kW} = \mathbf{32.78\ MW}$

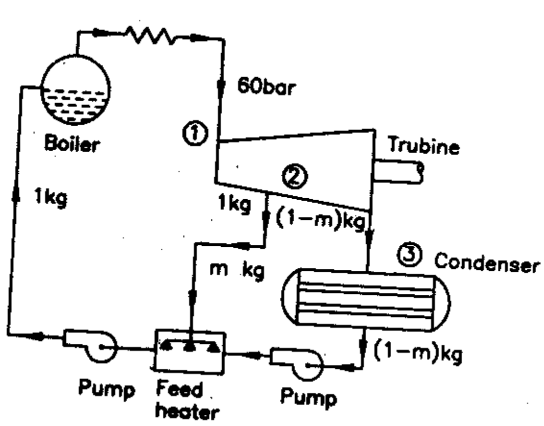
$$\eta \text{ (Rankine cycle)} = \frac{h_1 - h_3}{h_1 - h_{f3}} = \frac{3580 - 2440}{3580 - 251.5} = \frac{1140}{3328.5} = 0.34 = \mathbf{34\%}$$

$$\eta \text{ (with bleed heating)} = \frac{(h_1 - h_2) + (1 - m)\,(h_2 - h_3)}{(h_1 - h_{f2})}$$
$$= \frac{(3580 - 2870) + (1 - 0.11)\,(2870 - 2440)}{3580 - 541}$$
$$= \frac{710 + 382.7}{3039} = \frac{1092.7}{3039} = 0.36 = \mathbf{36\%}$$

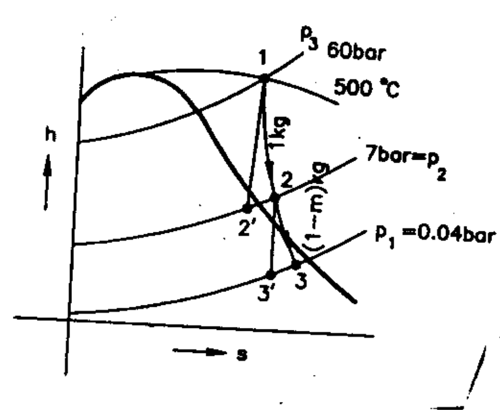
Problem 22.22. *Steam at 60 bar and 500°C is supplied to a steam power plant of 30 MW capacity. The condenser vacuum is 730 mm of Hg when barometer reads 760 mm of Hg. The steam is bled at 7 bar and used for feed heating in direct contact feed heater.*

Assuming the isentropic efficiency of each turbine 90%, determine (a) Fraction of steam bled for feed heating and (b) Boiler generating capacity.

Solution. The arrangement of the components is shown in Fig. Prob. 22.22 (*a*) and the processes are represented on *h-s* chart as shown in Fig. Prob. 22.22 (*b*).



(*a*)



(*b*)

Fig. Prob. 22.22.

The point 1 is marked as pressure (60 bar) and temperature 500°C are known. The point 2′ is marked by drawing verticle line through 1 till it cuts the 7 bar pressure line. Then point 2 is marked as 1-2 is 90% of 1-2′. The point 3′ is marked by drawing verticle line through point 2 where it cuts the pressure line

$$= 760 - 730 = 30 \text{ Hg of pressure}$$

$$= \frac{30 \times 133.3}{10^5} = 0.04 \text{ bar}$$

The point 3 is marked as 2-3 is 90% of 2-3′.

Enthalpies are read from h-s chart

$$h_1 = 3420 \text{ kJ/kg},\ h_2' = 2860 \text{ kJ/kg},\ h_2 = 2900 \text{ kJ/kg}$$

$$h_3 = 2410 \text{ kJ/kg},\ h_3' = 2190 \text{ kJ/kg}$$

$$\left.\begin{array}{l} h_{f3} \text{ (liquid heat at 0.04 bar)} = 121.5 \text{ kJ/kg} \\ h_{f2} \text{ (liquid heat at 7 bar)} = 697 \text{ kJ/kg} \end{array}\right\} \text{From steam tables}$$

Considering the feed heater

Heat lost by the steam = Heat gained by feed water

$$m\,(h_2 - h_{f2}) = (1 - m)\,(h_{f2} - h_{f3})$$

$$m\,(2900 - 697) = (1 - m)\,(697 - 121.5)$$

$$2203\,m = (1 - m) \times 575.5$$

$$\therefore \quad m = \frac{575.5}{2778.5} = \mathbf{0.207\ kg}$$

If m_s is the steam generated by the boiler per second

$$m_s\,(h_1 - h_2) + m_s\,(1 - m)\,(h_2 - h_3) = 30 \times 10^3$$

$$m_s\,[(3420 - 2900) + (1 - 0.207)\,(2900 - 2190)] = 30 \times 10^3$$

$$m_s\,[(520) + 0.793 \times 710] = 30 \times 10^3$$

$$m_s = \frac{30 \times 10^3}{520 + 563} = \frac{30 \times 10^3}{1083} = \mathbf{27.7\ kg/sec}$$

Problem 22.23. *A steam power plant of 120 MW capacity is equipped with regenerative as well as reheat arrangement. The steam is supplied at 86 bar and 50°C of superheat. The steam is extracted at 7 bar for feed heating and the remaining steam is reheated to 350°C and then expanded to 0.35 bar in the L.P. stage. Assume indirect contact type of feed heaters. Determine (a) The ratio of steam bled to steam generated (b) The boiler generating capacity in tons of steam/hr (c) Thermal efficiency of the cycle. Assume no losses and ideal processes of expansion.*

Solution. The arrangement of the components is shown in Fig. Prob. 22.23 (*a*) and the processes are represented on h-s chart as shown in Fig. Prob. 22.23 (*b*).

The point 1 is marked as a cross point of 86 bar pressure and 350°C as (T_s at 86 bar = 300°C). Then point 2 is marked drawing verticle line through point 1 till it cuts the 7 bar pressure line. The point 3 is marked as the cross point of 7 bar and 350°C temperature line. Then point '4' is marked by drawing verticle line through the point 3 till it cuts the 0.35 bar pressure line. The enthalpies are noted from the h-s chart.

$$h_1 = 2980 \text{ kJ/kg},\ h_2 = 2520 \text{ kJ/kg},\ h_3 = 3170 \text{ kJ/kg},\ h_4 = 2550 \text{ kJ/kg}$$

h_{f1} (at 0.35 bar, liquid heat) = 304.3 kJ/kg

h_{f2} (at 7 bar, liquid heat) = 697 kJ/kg

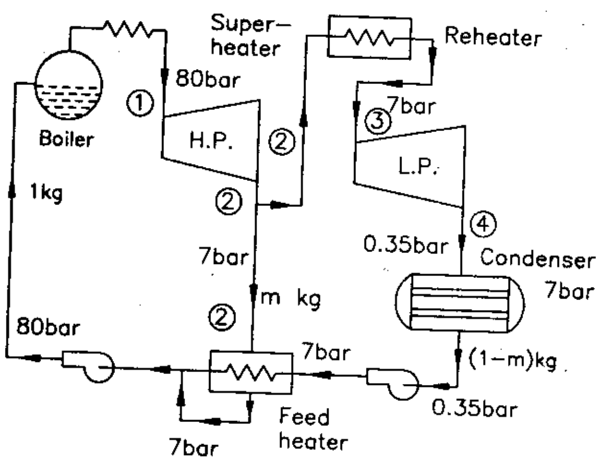
T_s (at 0.35 bar) = 72.7°C

T_s (at 7 bar) = 165°C

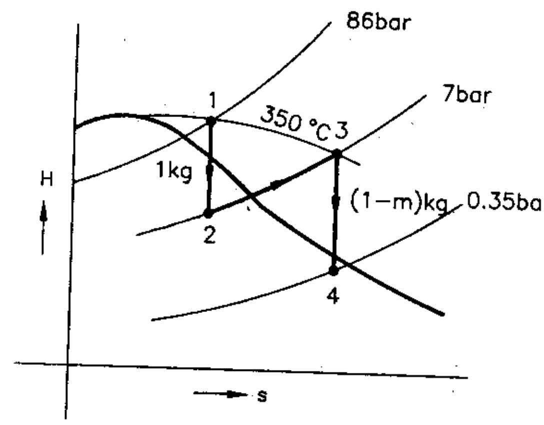
(*a*) For finding the steam bled at point (2), we can consider energy transfer in the feed heater.

Heat lost by m kg of steam = Heat gained by $(1 - m)$ kg of steam

$$m\,(h_2 - h_{f2}) = (1 - m)\,(h_{f2} - h_{f4})$$



(a)



(b)

Fig. Prob. 22.23.

$$m\ (2520 - 697) = (1 - m)\ (697 - 304.3)$$

$$1823\ m = (1 - m) \times 392.7 = 392.7 - 397.7\ \text{m}$$

$$\therefore \quad 2215.7\ m = 392.7$$

$$\therefore \quad m = \frac{392.7}{2215.7} = \mathbf{0.177\ kg}$$

Amount of steam bled per kg of steam supplied to the turbine = 0.177 kg.

$$\therefore \quad \frac{\text{Steam generated}}{\text{Steam bled}} = \frac{1}{0.177} = \mathbf{5.64.}$$

(*b*) If m_s is the mass of steam supplied to the power plant per second, then the work developed is given by

$$m_s\ (h_1 - h_2) + m_s\ (1 - m)\ (h_3 - h_4) = 120 \times 10^3$$

$$m_s\ [(2980 - 2520) + (1 - 0.177)\ (3170 - 2550)] = 120 \times 10^3$$

$$m_s\ (460 + 510) = 120 \times 10^3$$

$$m_s = \frac{120 \times 10^3}{970} = 123.7 \text{ kg/sec} = \frac{123.7 \times 3600}{1000} = \mathbf{445.4\ tons/hr}$$

(*c*) Thermal efficiency of the cycle

$$= \frac{\text{Output/kg of steam}}{\text{Input/kg of steam}}$$

$$= \frac{(h_1 - h_2) + (1 - m)(h_3 - h_4)}{(h_1 - h_{f4}) + (1 - m)\ (h_3 - h_2)}$$

$$= \frac{(2980 - 2520) + (1 - 0.177)\ (3170 - 2550)}{(2980 - 304.3) + (1 - 0.177)\ (3170 - 2520)}$$

$$= \frac{460 + 0.823 \times 620}{2675.7 + 0.823 \times 650} = \frac{970.3}{3210.6} = 0.302 = \mathbf{30.2\%}.$$

Problem 22.24. *A boiler supplies 2/3 steam at 30 bar and 300°C and 1/3 steam at 5 bar and 270°C. The high pressure steam is supplied to H.P. turbine, where it expands to 5 bar. Part of the steam is bled and supplied to the direct mixing type feed water heater. The remainder is mixed with other 1/3 of the steam coming from another boiler and is then supplied to the L.P. turbine where it expands to 0.07 bar. The condenser enters the feed water heater at 38°C and leaves at 180°C. Determine the fraction of steam bled and efficiency of the plant. If the steam generating capacity of the boiler is 20 tons/hr, find the power output capacity of the plant.*

Solution. The arrangement is shown in Fig. Prob. 22.24 (*a*) and the processes are shown on *h-s* chart as shown in Fig. Prob. 22.24 (*b*).

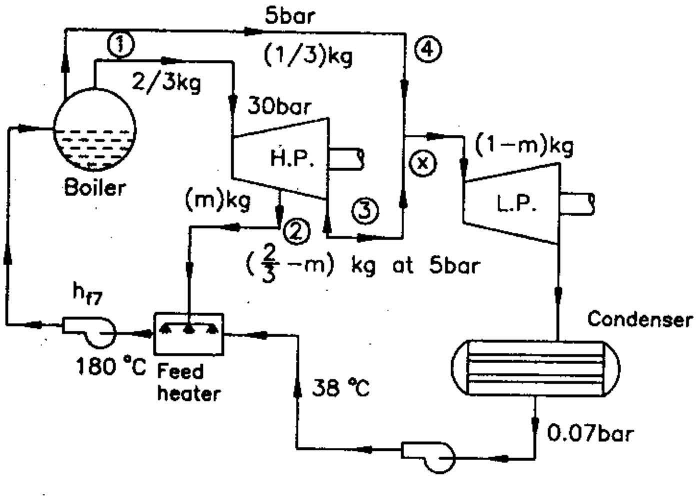


Fig. Prob. 22.24 (*a*).

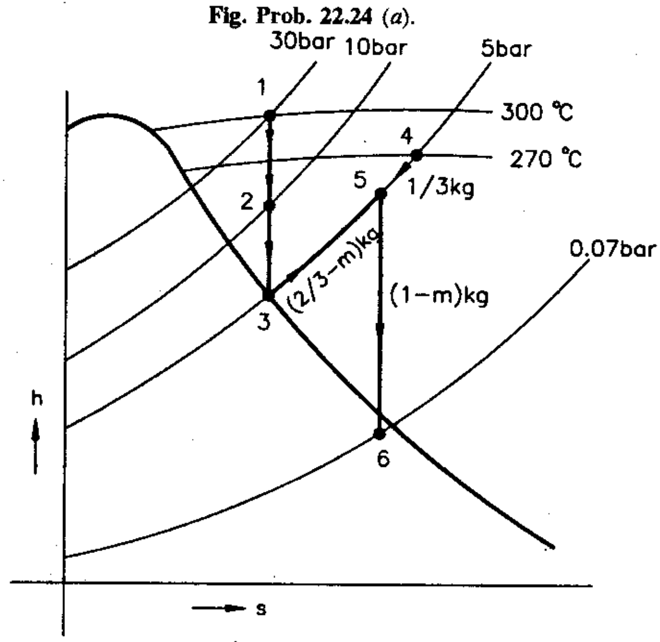


Fig. Prob. 22.24 (*b*).

As the outlet temperature of feed water coming out of feed heater is 180°C, the minimum temperature of steam bled from the turbine must be saturation pressure corresponding to saturation temperature of 180°C and it is 10 bar (from steam tables).

The points 1, 2 and 3 are marked as shown on *h-s* chart and enthalpies are noted as

$$h_1 = 3000 \text{ kJ/kg}, \ h_2 = 2780 \text{ kJ/kg}, \ h_3 = 2640 \text{ kJ/kg}$$

Considering the feed heater

Heat lost by steam (m kg) = Heat gain by water $(1 - m)$ kg

$$m (h_2 - h_{f2}) = (1 - m) \ C_{pw} \ (180 - 38)$$

where h_{f2} is the liquid heat at 10 bar pressure

$$m (2780 - 762.5) = (1 - m) \times 4.2 \times 142$$

$$\therefore \quad 2017.5 \ m = 596.5 - 596.5 \ m$$

$$\therefore \quad m = \frac{596.5}{2614} = \mathbf{0.2282 \ kg}$$

At point x as shown in Fig. Prob. 22.24 (*a*), $\frac{1}{3}$ kg of steam at 5 bar and 270°C (condition shown by point 4 on *h-s* chart) is mixed with the $\left(\frac{2}{3} - m\right)$ kg of steam at condition 3 shown on *h-s* chart.

$$\frac{1}{3} h_4 + \left(\frac{2}{3} - m\right) h_3 = (1 - m) \ h_5 \text{ (at 5 bar)}$$

$$h_4 = 3000 \text{ kJ/kg (from } h\text{-}s \text{ chart)}$$

$$\therefore \quad \frac{1}{3} \times 3000 + \left(\frac{2}{3} - 0.2282\right) \times 2640 = h_s \ (1 - m)$$

$$1000 + 1157.5 = (1 - 0.2282) \ h_5$$

$$\therefore \quad h_5 = \frac{2157.5}{0.7718} = 2795 \text{ kJ/kg}$$

Now we can mark the point '5' as pressure (5 bar) and enthalpy h_5 are known. The point 6 is the point intersecting the line drawn vertically from point 5 and pressure line 0.07 bar.

$$\therefore \quad h_6 = 2150 \text{ kJ/kg (from } h\text{-}s \text{ chart)}$$

The efficiency of the plant is given by

$$\eta = \frac{\text{Output}}{\text{Input}}$$

$$= \frac{\frac{2}{3} (h_1 - h_2) + \left(\frac{2}{3} - m\right) (h_2 - h_3) + (1 - m) \ (h_5 - h_6)}{\left(\frac{2}{3} h_1 + \frac{1}{3} h_4\right) - h_{f1}}$$

where $h_{f1} = h_{f2} = 762.5$

$$\therefore \quad \eta = \frac{\frac{2}{3} (3000 - 2780) + \left(\frac{2}{3} - 0.2282\right) (2780 - 2640) + (1 - 0.2282)(2795 - 2150)}{\frac{2}{3} \times 3000 + \frac{1}{3} \times 3000 - 762.5}$$

$$= \frac{147.4 + 61.85 + 67.7}{3000 - 762.5} = \frac{277}{2237.5} = 0.124 = \mathbf{12.4\%}$$

Steam generated per second in kg

$$= \frac{20 \times 1000}{3600} = \mathbf{5.55\ kg}$$

Power generating capacity of the plant

$$= 5.55 \times 277 \text{ (as 1 kg developes 277 kJ of power)}$$
$$= \mathbf{1539\ kW.}$$

Problem 22.25. *Steam at 60 bar and 500°C is supplied to a turbine the plant whose capacity is 30 MW. The condenser vacuum is 73 cm of Hg when barometer reads 76 cm of Hg. Steam is bled from the turbine at 3 bar and is used for feed water heating in a direct contact feed heater. Assuming isentropic efficiency of the turbine as 90% throughout, calculate*

(*a*) *Fraction of steam bled for feed heating.*

(*b*) *Steam supplied by the boiler in 60 bar kg/hr.* *(P.U. Dec. 2000)*

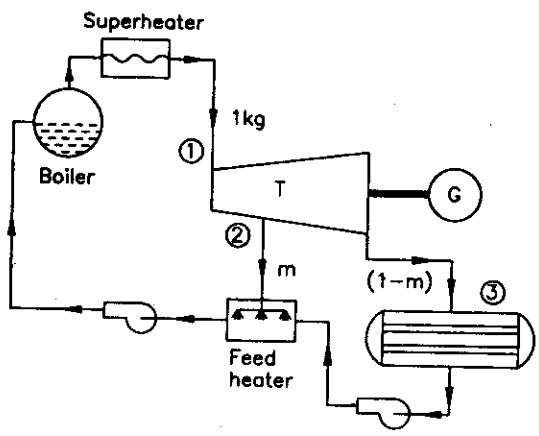


Fig. Prob. 22.25 (*a*).

Solution. The component layout is shown in Fig. Prob. 22.25 (*a*) and the processes as per given conditions are represented on *h-s* chart as shown in Fig. Prob. 22.25 (*b*) and the enthalpies at all the points are noted from *h-s* chart and steam table.

The condenser pressure $= \frac{76-73}{76} \times 1.013 = \mathbf{0.04\ bar.}$

$$h_1 = 3410,\ h_2 = 2720,\ h_3 = 2220$$
$$h_{f2} = 361.4 \text{ (3 bar)},\ h_{f3} = 121.4 \text{ (0.04 bar)}$$

Considering the energy balance at feed heater, we can write

$$m\,(h_2 - h_{f2}) = (1-m)\,(h_{f2} - h_{f3})$$
$$m\,(2720 - 361.4) = (1-m)\,(361.4 - 121.4)$$
$$2358.6\,m = 240\,(1-m)$$
$$\therefore \quad m = \mathbf{0.092\ kg/kg\ of\ steam}$$

Neglecting the pump work we can write the following equation for power developed by the turbine.

$m_s\,[(h_1 - h_2) + (1 - m_s)\,(h_2 - h_3)] = 30 \times 1000$ where m_s is the steam generated by the boiler per second

$m_s\,[(3410 - 2720) + (1 - m_s)\,(2720 - 2220)] = 30000$

$690\,m_s + (1 - m_s) \times 500 = 30000$

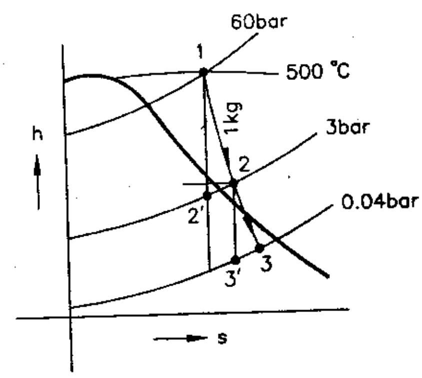


Fig. Prob. 22.25 (*b*).

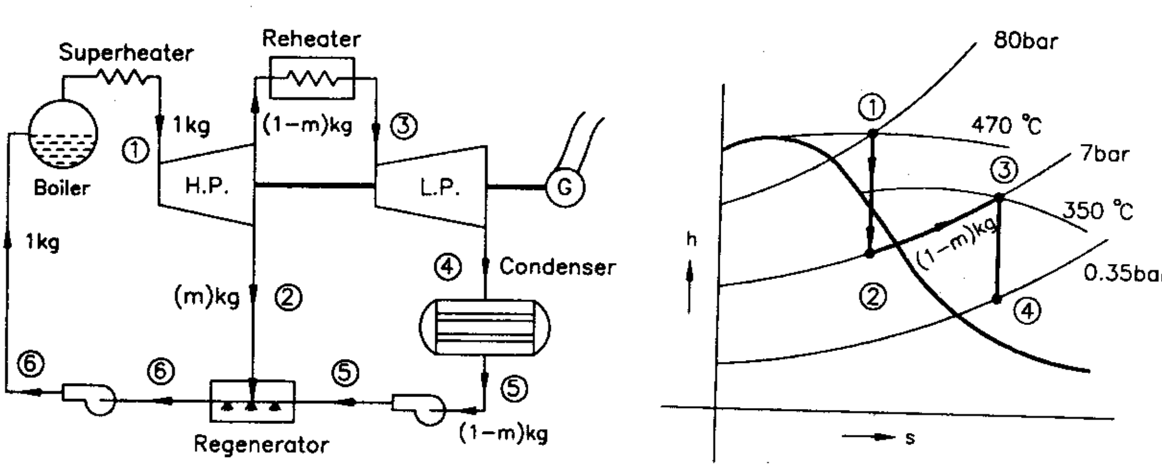
m_s [690 + (1 – 0.092) × 500] = 30000

$$\therefore \qquad m_s = \frac{30000}{690 + 454} = \frac{30{,}000}{1144} = \mathbf{26.2\ kg/sec} = \frac{26.2 \times 3600}{1000} = \mathbf{94.4\ tons/hr}$$

Problem 22.26. *A steam power plant equipped with regenerative as well as reheat arrangement is supplied with steam to the H.P. turbine at 80 bar and 470°C. For feed heating, a part of steam is extracted at 7 bar and the remainder of steam is reheated to 350°C in a reheater and then expanded in LP turbine down to 0.035 bar. Determine (a) Amount of steam bled off for feed heating (b) Amount of steam in LP turbine (c) Heat supplied in the boiler and reheater (d) Cycle efficiency (e) Output of turbine.*

The steam supplied by the boiler is 50 kg/sec. *(B.U., Dec. 2000)*

Solution. The arrangement of the components of the system is shown in Fig. Prob. 22.26 (*a*) and the processes are represented on *h-s* diagram as shown in Fig. Prob. 22.26 (*b*).



(*a*) (*b*)

Fig. Prob. 22.26.

The enthalpies at different points are noted from h-s chart and steam tables

$$h_1 = 3310$$
$$h_2 = 2780$$
$$h_3 = 3170$$
$$h_4 = 2220$$
$$h_6 = h_{f2} = 697$$
$$h_5 = h_{f4} = 111.85 \text{ (from steam tables)}$$

Considering the energy balance at regenerator.

Heat lost by steam = Heat gained by water

$$m (h_2 - h_{f2}) = (1 - m)(h_6 - h_5) = (1 - m)(h_{f2} - h_{f4})$$

$$\therefore \quad m (2780 - 697) = (1 - m)(697 - 111.85)$$

$$2083\, m = (1 - m)\, 585.15$$

$$\therefore \quad m = \frac{585.15}{2083 + 585.15} = \mathbf{0.22\ kg}$$

(*a*) $\therefore$ Amount of steam bled-off is 22% of the steam generated by the boiler.

(*b*) Amount of steam supplied to L.P. turbine

= 100 – 22 = 78% of the steam generated by boiler

(*c*) Heat supplied by the boiler per kg of steam generated

$= h_1 - h_6 = h_1 - h_{f2} = 3310 - 697 =$ **2613 kJ/kg**

Heat supplied in the reheater per kg of steam generated by the boiler

$= (1 - m)(h_3 - h_2)$

$= (1 - 0.22)(3170 - 2780) =$ **343.2 kJ**

Total amount of heat supplied by the boiler and reheater per kg of steam generated by the boiler

$Q_s = 2613 + 343.2 =$ **2956.2 kJ/kg**

(*d*) Amount of work done per kg of steam generated by the boiler

$W = (h_1 - h_2) + (1 - m)(h_3 - h_4)$ (the pump works are neglected)

$= (3310 - 2780) + (1 - 0.22)(3170 - 2220) = 530 + 836 =$ **1366 kJ/kg**

$$\therefore \quad \text{Cycle } \eta = \frac{W}{Q_s} = \frac{1366}{2956.2} = 0.462 = \mathbf{46.2\%}$$

(*e*) Power developed by the system

$$= m_s \,.\, W = 50 \times 1366 \text{ kJ/sec} = \frac{50 \times 1366}{1000} = \mathbf{68.3\ MW.}$$

Problem 22.27. *A steam power plant operates on ideal Rankine cycle using reheater and regenerative feed water heaters. It has one open feed heater. Steam is supplied at 150 bar and 600°C. The condenser pressure is 10 kPa. Some steam is extracted from the turbine at 40 bar for closed feed water heater and remaining steam is reheated at 40 bar to 600°C. Extracted steam is completely condensed in this closed feed water heater and is pumped to 150 bar before mixing with the feed water heater. Steam for the open feed water heater is bled from L.P. turbine at 5 bar. Determine (i) fraction of steam extracted from the turbines at each bled heater and (ii) thermal efficiency. Draw the line diagram of the component and represent the cycle on T-s diagram.* *(P.U. Dec. 2000)*

Solution. The arrangement of the components is shown in Fig. Prob. 22.27 (*a*) and the processes are represented on T-s diagram as shown in Fig. Prob. 22.27 (*b*).

All the enthalpies are noted from h-s chart and steam tables.

All enthalpies are in kJ/kg

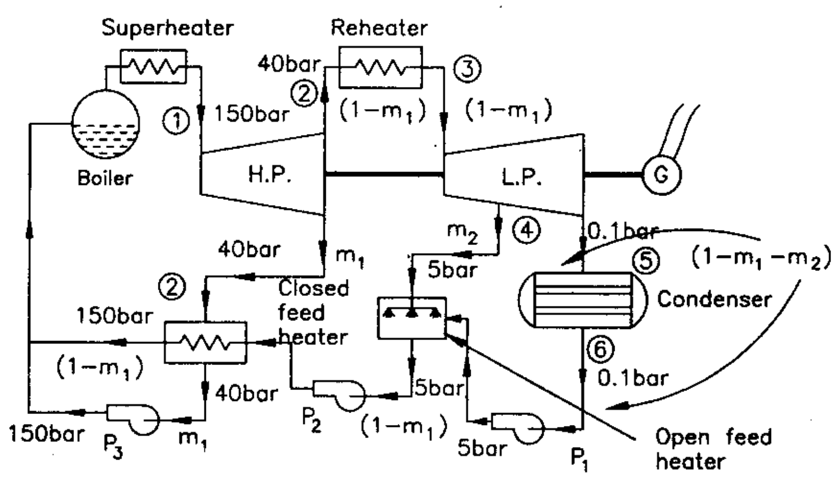


Fig. Prob. 22.27 (*a*).

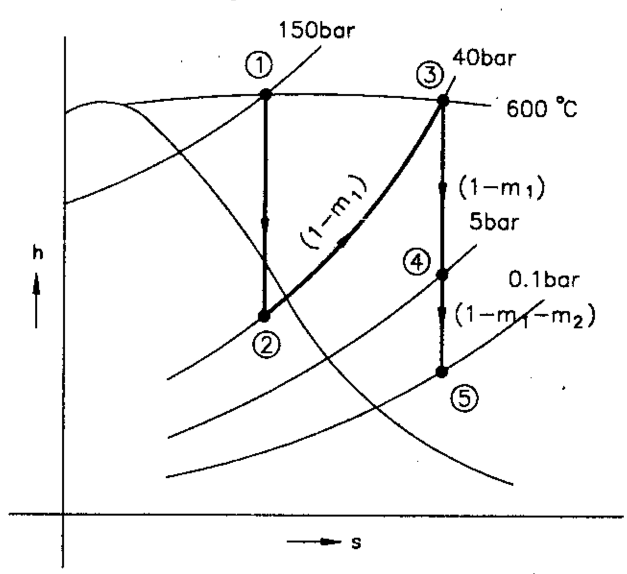


Fig. Prob. 22.27 (*b*).

$h_1 = 3570,\ h_2 = 3280,\ h_3 = 3650,\ h_4 = 2920,\ h_5 = 2280$

$h_{f1} = 1610$ (at 150 bar)

$h_{f2} = 1087$ (at 40 bar)

$h_{f4} = 640$ (at 5 bar)

$h_{f5} = 192$ (at 0.1 bar)

(1) Considering closed feed heater

$$m_1\,(h_2 - h_{f2}) = (1 - m_1)\,(h_{f2} - h_{f4})$$

$$m_1\,(3280 - 1087) = (1 - m_1)\,(1087 - 640)$$

$\therefore$ $m_1 =$ **0.17 kg/kg** of steam supplied by boiler

(2) Considering open feed heater

$$m_2\,(h_4 - h_{f4}) = (1 - m_1 - m_2)\,(h_{f4} - h_{f5})$$

$\therefore\ m_2\,(2920 - 640) = (1 - 0.17 - m_2)\,(640 - 192)$

$\therefore\qquad m_2 = \mathbf{0.136\ kg/kg}$ of steam supplied by boiler

Total work done per kg of steam supplied by the boiler

$$= 1 \times (h_1 - h_2) + (1 - m_1)\,(h_3 - h_4) + (1 - m_1 - m_2)\,(h_4 - h_5)$$
$$= (3570 - 3280) + (1 - 0.17)\,(3650 - 2920) = (1 - 0.17 - 0.136)\,(2920 - 2280)$$
$$= 290 + 606 + 444 = \mathbf{1340\ kJ/kg}$$

W_{p1} (Work done by the pump P_1)

$$= \frac{v_{w1}\,(1 - m_1 - m_2)\,(5 - 0.1) \times 10^5}{1000}$$
$$= \frac{1}{1000}\,(1 - 0.17 - 0.136) \times 4.9 \times 10^2 = \mathbf{0.34\ kJ/kg}$$

(Work done by pump P_2)

$$W_{p2} = \frac{v_{w2}\,(1 - m_1)\,(150 - 5) \times 10^5}{1000}$$
$$= \frac{1}{1000} \times (1 - 0.17) \times 145 \times 10^2 = \mathbf{12\ kJ/kg}$$

(Work done by pump P_3)

$$W_{p3} = \frac{1}{1000}\,\frac{m_1\,(150 - 40) \times 10^5}{1000}$$
$$= \frac{1}{1000} \times 0.17 \times \frac{110 \times 10^5}{1000} = 1.87$$

where $\qquad v_{w1} = v_{w2} = v_{w3} = \dfrac{1}{1000}\ \text{m}^3/\text{kg}$

W_{pt} (total) $= W_{p1} + W_{p2} + W_{p3} = 0.34 + 12 + 1.87 = 14.21$ kJ/kg of steam supplied by boiler

$\therefore$ Net work (W_n) done by the turbine per kg of steam supplied by the boiler

$$W_n = 1340 - 14.21 = \mathbf{1325.8\ kJ}$$

Heat of feed water entering into the boiler

$$= (1 - m_1) \times 1610 + m_1 \times 1610 = \mathbf{1610\ kJ}$$

Heat supplied by the boiler per kg of steam

$$Q_{s1} = h_1 - 1610 = 3570 - 1610 = \mathbf{1960\ kJ}$$

Q_{s2} = Heat supplied in the reheater

$$= (1 - m_1)\,(h_3 - h_2)$$
$$= (1 - 0.17)\,(3650 - 3280) = \mathbf{307\ kJ/kg}$$ of steam supplied by the boiler

Q_{st} (total heat supplied)

$$= Q_{s1} + Q_{s2} = 1960 + 307 = \mathbf{2267\ kJ/kg}$$

$$\eta\ (\text{system}) = \frac{W_n}{Q_{st}} = \frac{1325.8}{2267} = 0.585 = \mathbf{58.5\%}.$$

Problem 22.28. *A steam power plant working on reheat-regenerative cycle has the following data.*

Steam entery to H.P. turbine at 80 bar and 470°C.

Steam extraction for feed heating at 7 bar.

Remaining steam reheated to 350°C in reheater.

Exhaust pressure from L.P. turbine 0.035 bar.

Determine the following :

(*i*) *Fraction of steam bled off for reheating.*

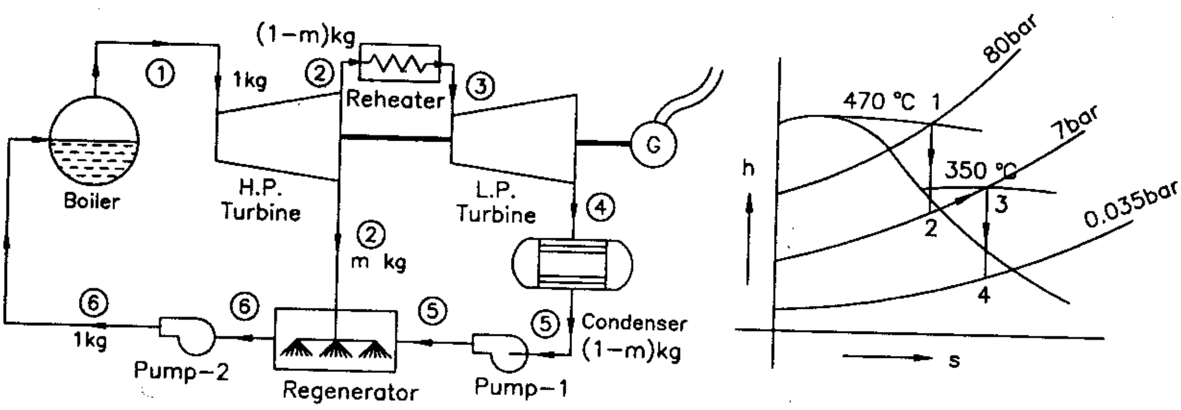
(*ii*) *Heat supplied per kg of steam in boiler and reheater.*

(*iii*) *Power output of the plant if the steam flow rate is 100 kg/sec.*

(*iv*) *Thermal efficiency of the plant.*

Neglect the pumps work and assume all ideal conditions. (*P.U., Dec. 2001*)

Solution. The arrangement of the components is shown in Fkig. Prob. 22.28 (*a*) and processes are represented on *h-s* diagram as shown in Fig. Prob. 22.28 (*b*).



(*a*) (*b*)

Fig. Prob. 22.28.

The points 1, 2, 3 and 4 are marked on *h-s* diagram as shown in Fig. Prob. 22.28 (*b*) and enthalpies are noted from the *h-s* chart.

$$h_1 = 3350 \text{ kJ/kg}, \; h_2 = 2770 \text{ kJ/kg}$$
$$h_3 = 3170 \text{ kJ/kg}, \; h_4 = 2220 \text{ kJ/kg}$$

The following enthalpies are noted from steam tables

$$h_{f5} = 112 \text{ kJ/kg and } h_{f6} = 697 \text{ kJ/kg}$$

Considering energy balance at regenerator, we can write

$$m\,(h_2 - h_{f6}) = (1 - m)\,(h_{f6} - h_{f5})$$
$$\therefore \quad m\,(2770 - 697) = (1 - m)\,(697 - 112)$$
$$\therefore \quad m = \mathbf{0.22.}$$

(*i*) The fraction of steam bled for reheating is 22%.

(*ii*) Heat supplied in the boiler and reheater per kg of steam generated in the boiler

$$= (h_1 - h_{f6}) + (1 - m)\,(h_3 - h_2)$$
$$= (3350 - 697) + (1 - 0.22)\,(3170 - 2770)$$
$$= 2653 + 312 = \mathbf{2965 \text{ kJ/kg}}$$

(*iii*) Power output per kg of steam generated by the boiler

$$= (h_1 - h_2) + (1 - m)\,(h_3 - h_4)$$
$$= (3350 - 2270) + (1 - 0.22)\,(3170 - 2220)$$
$$= 580 + 740 = 1320 \text{ kJ/kg}$$

∴ Capacity of the plant

$$= 1320 \times 100 \text{ kJ/sec}$$
$$= \mathbf{132000\ kW = 132\ MW}$$

(*iv*) The efficiency of the plant is given by

$$\eta = \frac{\text{Power output per kg of steam}}{\text{Heat supplied per kg of steam}} = \frac{1320}{2965} \times 100 = \mathbf{44.5\%}$$

Students are advised to repeat the problem considering the following :

(*i*) Pump work should be considered.

(*ii*) The isentropic efficiency of both expansion is 85%.

(*iii*) There is 5°C undercooling of the condensate coming out of condenser.

Problem 22.29. *A power plant works on a binary vapour cycle operates on mercury and steam as working fluids. Saturated mercury vapour at 4.5 bar is supplied to the mercury turbine and it exhausts at 0.04 bar. The mercury condenser is used to generate saturated steam at 15 bar which expands in steam turbine to 0.04 bar. Determine :*

(*i*) *The overall efficiency of the cycle.*

(*ii*) *If 48000 kg/hr of steam flows through the steam turbine, what is the flow of mercury through mercury turbine.*

(*iii*) *Find the useful work done in the binary vapour cycle for the given steam flow.*

Assume all processes are reversible.

Use the following given properties of the mercury.

Pressure (bar)	*Temp. (°C)*	h_f *(kJ/kg)*	h_g *(kJ/kg)*	s_f *(kJ/kg-K)*	s_g *(kJ/kg-K)*	v_{sf} (m^3/kg)	v_{sg} (m^3/kg)
4.5	450	62.9	356	0.135	0.539	80×10^{-6}	0.068
0.04	217	30.0	330	0.081	0.693	76.5×10^{-6}	5.178

Neglect the pump works. *(B.U. Dec. 2001)*

Solution. The arrangement of the components for the binary cycle is shown in Fig. Prob. 22.29 (*a*) and corresponding processes are represented on $h - s$ chart as shown in Fig. Prob. 22.29 (*b*).

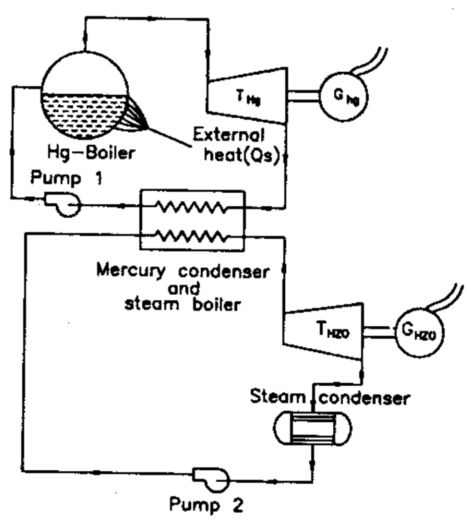


Fig. Prob. 22.29 (*a*).

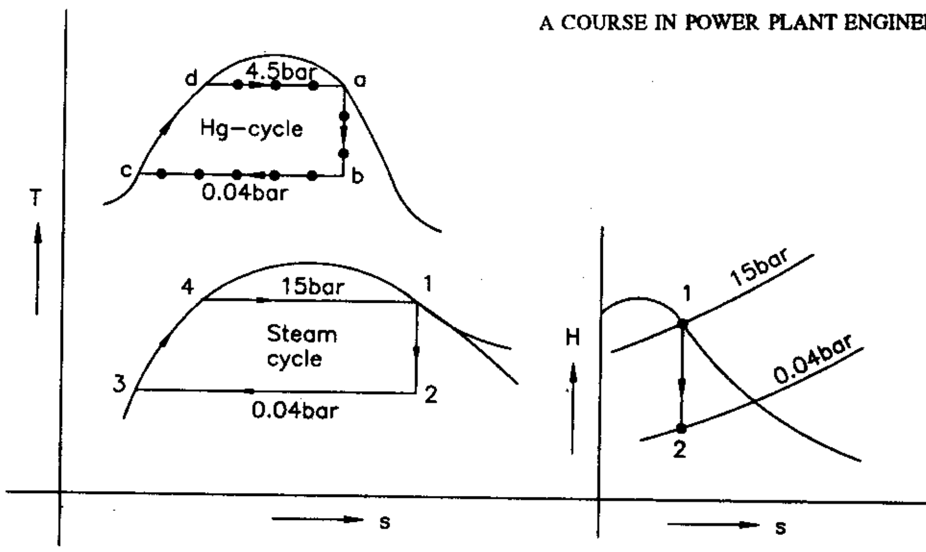


Fig. Prob. 22.29 (*b*).

$$m_{H2O} = \frac{48000}{3600} = \mathbf{13.33\ kg/sec.}$$

Considering first mercury-cycle

For the process *ab*

$$s_a = s_b$$

$\therefore$ $(s_g)_{\text{at 4.5 bar}} = [s_{fb} + x_b\,(s_{gb} - s_{fb})]_{\text{at 0.04 bar}}$

$\therefore$ $0.539 = 0.081 + x_b\,(0.693 - 0.081)$

$\therefore$ $x_b = \mathbf{0.75}$

$\therefore$ $h_b = h_{fb} + x_b\,(h_{gb} - h_{fb})$
$= 30.0 + 0.75\,(330 - 30) = \mathbf{255\ kJ/kg}$

$h_c = h_{fc} = \mathbf{30\ kJ/kg}$

Considering Steam-cycle

We can represent the process 1-2 on the *h-s* chart and we can note-down the following enthalpies

$h_1 = 2800$ kJ/kg, $h_2 = 1970$ kJ/kg

$h_{f3} = 121.4$ kJ/kg, $h_{f4} = 844.6$ kJ/kg (from steam tables)

Now considering the energy balance at mercury condenser

$$m_{hg}\,(h_b - h_{fc}) = m_{H2O}\,(h_1 - h_{f3})$$

$\therefore$ $m_{hg}\,(255 - 30) = 13.33\,(2800 - 121.4)$

$\therefore$ $$m_{hg} = \frac{13.33 \times 2678.6}{225} = \mathbf{158.7\ kg/sec}$$

$\therefore$ $$\frac{m_{hg}}{m_{H2O}} = \frac{158.7}{13.33} = \mathbf{11.9}$$

Total work done per second (W_t) is given by

$W_t = W_{Hg} + W_{H2O}$
$= m_{Hg}\,(h_a - h_b) + m_{H2O}\,(h_1 - h_2)$
$= 158.7\,(356 - 255) + 13.33\,(2800 - 1970)$
$= 158.7 \times 101 + 13.33 \times 830 = 16028.7 + 11063.9 = \mathbf{27092.6\ kW}$
$= \mathbf{27.93\ MW}$

The total heat supplied (Q_s) is given by

$$Q_s = (Q_s)_{Hg}$$
$$= m_{Hg}\,(h_a - h_{fc})$$
$$= 158.7\,(356 - 30)$$
$$= 158.7 \times 326$$
$$= 51736.0 \text{ kJ/sec}$$

$$\therefore \qquad \eta_{overall} = \frac{W_t}{Q_s} = \frac{27092.6}{51736.0} \times 100 = \mathbf{52.4\%}.$$

EXERCISES

22.1. What are the difficulties experienced in carrying out Carnot cycle ?
Describe Rankine cycle and show how it differs from a Carnot cycle by sketching both cycles on p-v and T-s diagrams for water and steam, the steam is dry and saturated after evaporation in both cycles.

22.2. What should be the properties of working fluid so that Rankine cycle approaches the Carnot cycle ?

22.3. Explain why the Rankine cycle rather than Carnot cycle is used as a standard of reference for the performance of steam plants.

22.4. Draw $T - s$ diagram of Rankine cycle using dry-saturated steam and develop the equation for the rankine cycle efficiency.

22.5. Sketch a schematic diagram of a condensing steam power plant and draw the sequence of operation of $T - s$ diagram.

22.6. Discuss with the help of $T - s$ diagram, the effect of the change of the following variables on efficiency and power output
(*a*) Inlet pressure
(*b*) Inlet temperature with inlet pressure maintaining constant
(*c*) Condenser pressure.

22.7. Prove that the thermal efficiency of a Rankine cycle using superheated steam is greater than the thermal efficiency of a corresponding Rankine cycle using steam without superheat. Both the cycles operate between the same boiler and condenser pressure limits.

22.8. What are the effects of temperature and pressure of supply steam and condenser pressure on Rankine cycle efficiency ? What are the limitations of these factors to increase the efficiency ?

22.9. What are the advantages claimed for relating the steam in high pressure steam plants ?

22.10. What are the different methods used for reheating the steam ? Discuss the merits and demerits of different systems.

22.11. When reheating of steam is economically justified ?

22.12. State briefly the advantages of a regenerative feed heating in steam power cycle.

22.13. Show that the thermal efficiency of a regenerative cycle is always greater than that of a staight Rankine cycle regardless of where steam is tapped off.

22.14. What are the different arrangements used for the disposal of bled steam condensate ? List out merits and demerits of each over others.

22.15. What do you understand by a binary vapour cycle ? List out the advantages of binary vapour cycle over the Rankine cycle.

22.16. In case of binary-cycle prove that the overall thermal efficiency of the cycle is given by

$$\eta = \eta_{hg}\,(1 - \eta_s) + \eta_s$$

where η_{hg} and η_s are the Rankine efficiencies of mercury and steam cycles respectively.

22.17. What do you understand by Topping cycle ? When it is economically justified ?

22.18. When the adoption of economiser and air-preheater are economically justified ?

22.19. Discuss the different types of air-heaters used in practice. List out the merits and demerits of each over other. What factors are considered in their design ?

22.20. Why superheating of the steam is essential in power plants ? What different types of superheaters are used in modern power plants ?
Discuss the different methods used to control the superheat temperature of the steam.

Reheat Cycles

22.21. Steam at 28 bar and 50°C superheat is passed through a turbine. It expands to a pressure when the steam is dry and saturated. It is then reheated at constant pressure passing through the second stage of turbine. Assuming the expansions are isentropic, find out

(*a*) work done per kg of steam and thermal efficiency taking reheating into account and

(*b*) if the steam expands direct to exhaust pressure without reheating.

Taking condenser pressure in both cases = 0.2 bar.

22.22. Compare the theoretical heat consumption in kJ per kW-hr and final wetness of exhaust steam for the following cycles in which steam is supplied to a turbine at 70 bar and exhausted at 0.035 bar.

(*a*) Initial steam temperature 480°C. Overall efficiency ratio 0.82 and alternator efficiency 96%.

(*b*) Initial steam temperature 480°C. Steam is reheated to 480°C. Steam pressure entering reheater 8 bar and leaving reheater is 6.5 bar. Overall efficiency ratio of each stage is 0.82 and alternator efficiency is 96%.

22.23. Steam at 40 bar and 450°C is supplied to the steam turbine and it is exhausted at 0.06 bar. The turbine develops 2500 kW at 3000 r.p.m. The expansion takes place in two stages. The steam is reheated to 410°C after leaving the HP stage. Assuming both stages develop equal power and neglecting the pump work and other losses in the system, find out

(*a*) the pressure of reheating,

(*b*) steam consumption per minute,

(*c*) thermal efficiency of the cycle. Take efficiency ratio of each stage of 0.8.

Compare this efficiency with thermal efficiency of Rankine cycle.

22.24. Two turbines *A* and *B* operate with steam as 100 bar and 500°C. In each turbine the steam expands in high-pressure stage to 8.5 bar with stage efficiency of 80%. In turbine *A*, the expansion is further continued in low pressure turbine from 8.5 bar to condenser pressure of 0.03 bar with stage efficiency of 75%. In turbine *B*, the steam is reheated to 500°C after leaving the high pressure stage and then expands in low pressure stage to condenser pressure of 0.03 bar. Compare the two power cycles with respect to (*a*) thermal efficiency and (*b*) steam consumption at full load out-put of 50 MW.

22.25. Steam at 80 bar and 500°C is supplied to a steam turbine and exhaust takes place at 0.05 bar. Steam is reheated at 20 bar to its original temperature. Due to friction ; there is a drop of pressure in the reheater. Find out the pressure drop in the reheater if gain in thermal efficiency due to reheating neutralises the loss due to pressure drop. Assume the expansions are isentropic.

22.26. A steam turbine is divided into two sections H.P. and L.P. with a reheater interposed in between the two sections. The steam on its way to the turbine at 30 bar and 500°C passes through a reheater where it gives up heat at constant pressure to heat the steam flowing from H.P. turbine to L.P. turbine. The steam then enters the H.P. turbine at 30 bar and 80°C. The steam leaves the H.P. turbine at 7 bar and L.P. turbine at 0.07 bar. Neglecting the losses and assuming an internal efficiency of 80% for both the sections, find out thermal efficiency of the cycle and condition of steam at the entrance to the L.P. section.

22.27. In a steam turbine plant, the steam at 30 bar and 400°C passes through a reheater placed between H.P. and L.P. stages of the turbine. The steam at 30 bar and 300°C enters into the H.P. turbine and expands to 5 bar with an efficiency ratio (isentropic efficiency) of 0.85. The steam after passing through the reheater enters L.P. turbine and expands to 0.06 bar with same efficiency ratio. The pressure loss in the reheater is 0.5 bar. Find out the output and thermal efficiency of the plant if the steam flow rate through the turbine is 80 kg/sec.

Regenerative Cycles

22.28. A single stage regenerative feed heating is used in a turbine plant. The steam at 50 bar and 300°C is supplied to the turbine and exhausted to the condenser at 0.06 bar. The steam is bled for heating at 18 bar. Assuming the expansion through the turbine is isentropic, find out the improvement in thermal efficiency due to regenerative feed heating.

22.29. The following readings were taken during the test on steam turbine equipped with single stage regenerative feed-heating.

Steam pressure at stop valve = 20 bar.

Steam temperature at stop valve = 300°C.

Steam pressure at nozzle box = 16 bar.

Bled steam pressure = 3 bar.

Exhaust pressure = 0.07 bar.
Temperature of condensate = 50°C.
Temperature of drain water leaving the drain cooler = 55°C.
Temperature of feed water leaving the feed heater = 125°C.
Find out the quantity of bled steam, thermal efficiency of the plant and power developed by the turbine per kg of feed.
Neglect the pump work and assume the expansion is isentropic.

22.30. 136 tons of steam per hour at 135 bar and 600°C is supplied to the steam turbine and exhausted to the condenser at 0.07 bar. The steam is extracted at 20 bar and 10 bar for feed heating. Find out
(*a*) steam bled per hour at each point.
(*b*) power developed in kW.
(*c*) thermal efficiency. Assume the expansion is isentropic. Neglect the pump work and other losses.

22.31. A steam at 17 bar and 208°C is supplied to a steam turbine. Two stage bled heating is used to increase the thermal efficiency of the cycle at 3.4 bar and 0.6 bar respectively. The exhaust pressure of the steam is 0.06 bar. The stage (or isentropic) efficiency between 17 bar and 3.4 bar is 70% and in the other two stages is 65%. Assuming the temperature of feed water is raised to that of bled steam temperature and neglecting pump work and other losses, find out
(*a*) bled steam at each heater per kg of steam supplied to the turbine.
(*b*) work done per kg of steam supplied to the turbine and
(*c*) the thermal efficiency of the cycle.

22.32. The steam at 17.5 bar and 300°C is supplied to a turbine. The condition of steam at exit of the turbine is 0.9 dry and its pressure is 0.07 bar. Two stage feed heating is used to increase the thermal efficiency of the cycle. The bleeding is done at 4 bar and 0.8 bar. In each feed heater, the feed water is heated to saturation temperature corresponding to the pressure at that point and there is no undercooling of condensate in the feed heaters. The low pressure feed heater drains to the condenser and condensate from high pressure heater is pumped directly into the boiler feed line ahead.
Assuming the condition-line for the turbine is straight and neglecting pump work and other losses, find out
(*a*) Steam bled to each heater per kg of steam supplied to the turbine.
(*b*) Thermal efficiency.
(*c*) Power developed by the turbine if steam flow rate is 4.45 kg/sec.
(*d*) Percentage gain in thermal efficiency due to bleed heating.
(*e*) Increase in steam consumption rate with regeneration.

22.33. Steam at 30 bar and 320°C is supplied to a turbine. The plant is provided with two stages of bleed heating. The exhaust of steam takes place at 0.07 bar. The temperature of the condensate coming out of condenser is 37°C. The first tapping is taken at 2 bar. Assuming the rise in temperature of feed has to be divided equally between two heaters, find out
(*a*) pressure for second tapping. Take isentropic or stage efficiency for each expansion 70%. Also find out
(*b*) the quantities of steam tapped at each point and
(*c*) gain in thermal efficiency due to bleed heating.

22.34. The steam at 30 bar and 400°C is supplied to a turbine which exhausts at 0.07 bar. The steam is bled off for feed heating at 6 bar and 1 bar. The temperature of the condensate coming out from condenser is 38.7°C. The temperatures of the feed leaving the first and second heaters are 95°C and 154°C respectively. The bled steam is condensed in the feed heaters and there is no undercooling of the condensate formed from bled steam. The drain from the first heater is passed into the second heater through a steam trap and combined drain from the second heater is pumped by a drain pump into feed pipe after the second heater. Assuming isentropic efficiency of each stage 80% and neglecting the pump work and other losses, find out
(*a*) thermal efficiency of the plant
(*b*) capacity of the drain pump if the power developed by the turbine is 12 MW.

22.35. Steam at 20 bar and 300°C is supplied to a 3-stage pressure compounded turbine. The exhaust pressure of the steam is 0.06 bar. Steam is bled at 2 bar and 0.4 bar for feed heating. The drain of the first heater is passed through the second heater and total drain of the second cooler is fed to the condenser through one more drain cooler. The temperature of the drain coming out from second cooler is reduced to condenser temperature. Assuming isentropic efficiency of each expansion 80%, and neglecting pump work and other losses, find out

(*a*) the percentage increase in thermal efficiency due to bleeding

(*b*) the percentage increase in boiler capacity for the same output

(*c*) percentage reduction in condenser capacity.

22.36. A steam turbine is supplied with steam at 26 bar and 360°C and exhausts at 0.03 bar. Steam is bled off at 5 bar, 1.4 bar and 0.3 bar. The bled steam being cascaded from heater to heater until it discharges through a drain cooler to the condenser. Assuming that the turbine expansion is isentropic and drain cooler raises the combined heater condensate to condenser temperature, find out :

(*a*) Total mass of bled steam per kg of feed.

(*b*) The gain in efficiency over the corresponding Rankine cycle.

(*c*) Steam consumption per kWh with and without feed heating.

(*d*) The amount of heat carried away by condenser cooling water per kWh with and without feed heating.

Reheat and Regenerative Cycles

22.37. Steam at 77.5 bar and 400°C is supplied to a steam turbine. The steam is bled for feed heating at 26 bar and remaining steam expands to 17 bar. The steam coming out of 17 bar is reheated to 420°C and then the expansion further proceeds to condenser pressure of 0.07 bar. Assuming the expansion at all stages is isentropic, find out the percentage steam bled and thermal efficiency of the cycle. Pressure loss in reheater is 1 bar.

22.38. Steam at 20 bar and 300°C is supplied to a steam turbine. The steam expands isentropically in H.P. turbine to 8 bar and then it is reheated to its original temperature at constant pressure. The reheated steam expands isentropically in I.P. turbine to 1 bar. The part of the steam from exit of I.P. turbine is removed for feed heating and remaining steam expands isentropically in L.P. cylinder to condenser pressure of 0.035 bar. The condensate from feed heater is pumped ahead into the feed water line. Find out the efficiency of the cycle and compare with the Rankine cycle working in the same pressure limit.

22.39. Steam is supplied to a steam turbine at 90 bar and 500°C. After expansion in the turbine to 10 bar, a portion of steam is bled for feed heating and remaining steam is reheated to its original temperature at constant pressure. The steam coming out of reheater again expands in L.P. stage of the turbine to condenser pressure of 0.07 bar isentropically. Neglect the pump work and other losses in the system, find out the thermal efficiency of the cycle and steam consumption in kg per kW-hr.

22.40. Steam at 100 bar and 500°C is supplied to a steam turbine. The steam at 10 bar and 1 bar is reheated to its original temperature at constant pressure. The part of the steam is also extracted at 10 bar and 1 bar for feed heating. The condenser pressure is 0.07 bar. The feed condensate coming out from each heater is pumped in the feed line by separate pump ahead in the feed line. Assuming the isentropic efficiency of expansion at all stages of 80%, find out (*a*) thermal efficiency of the cycle and (*b*) The power developed by the generator in MW if the flow of steam is 10 kg/sec. Take mechanical efficiency, transmission efficiency and generator efficiency as 90%, 95% and 96% respectively.

Neglect the pump work and other losses in the system.

22.41. A two-stage turbine receives 2/3 of its total steam supply at 14 bar and 320°C and the steam expands in H.P. stage to 3.5 bar. Part of the exhaust steam from H.P. turbine for feed heating and remaining is mixed with other 1/3 of steam which is supplied at 3 bar and 270°C and the mixture is allowed to pass through the L.P. turbine where it expands to 0.07 bar. The isentropic efficiency of H.P. turbine is 80% and L.P. turbinee is 76%. The overall mechanical efficiency of the turbine is 90%. Feed water enters the feed heater at 38°C and comes out at 120°C. The feed heater is direct contact type. Find out steam flow rate in kg/sec to produce 25000 kW power.

Binary Cycles

22.42. In a mercury-steam binary cycle, the dry steam is generated at 15 bar in mercury condenser and it is then superheated to 310°C in a separate superheater. The steam expands to 2 bar and part of the steam at 2 bar is taken for feed heating and the remaining steam expands to condenser pressure of 0.04 bar. Take the isentropic efficiency for mercury turbine and both stages of steam turbine as 80%. The condensate from feed heater is pumped in the feed line ahead. The mercury cycle operates between 5 bar and 0.1 bar. Assuming the mercury vapour entering into the mercury turbine is dry and saturated, find out (*a*) Steam generated per kg of Hg circulated, (*b*) The overall thermal efficiency of the plant.

Assume both mercury and steam cycle operate on Rankine cycle and neglect the pump work and other losses in the system. Take the following properties of mercury :

Pressure bar	*Saturation Temp. °C*	*Total heat kJ/kg*		*Entropy kJ/kg-K*	
		Liquid	*Total*	*Liquid*	*Total*
5	—	—	358	—	0.54
0.1	1048	35	335	0.09	0.66

22.43. In a binary turbine plant, the mercury turbine works between 11 bar and 0.2 bar and steam turbine works between 40 bar and 0.07 bar. The mercury vapour is supplied dry and saturated but steam is superheated by 100°C. Steam is tapped at 4 bar and used for feed heating in an open (direct) feed heater. Taking the efficiency ratio of each turbine 80%, find out the mass of mercury circulated per kg of steam and the thermal efficiency of the system.

22.44. A binary vapour plant uses mercury between a temperature range of 205°C and 540°C. The mercury is dry and saturated at higher temperature limit. The steam cycle works between 17.35 bar supply pressure and 736.6 mm of Hg vacuum in condenser. The supply temperature of steam is 370°C. The feed water is raised to 205°C in the economiser and is evaporated to dry steam in the mercury condenser and it is superheated by the gases. Assuming the isentropic expansion in mercury and steam turbines and using the value given in the following tables, find out

(*a*) Mass of mercury per kg of steam generated.

(*b*) The work done by mercury and steam separately per kg of steam generated.

(*c*) The ideal efficiency of the plant.

Temperature	*Total heaters*			*Entropy*		
	h_f	h_{fg}	h_g	s_f	h_{fg}/T	s_g
540°C	75.5	291.5	367	0.152	0.359	0.54
205°	26	301.5	327.5	0.079	0.625	0.704

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