Chapter 1

Steam Power Plant

Important definitions:

Saturation temperature: is the temperature a pure substance start boiling at certain pressure, this pressure is called saturation pressure.

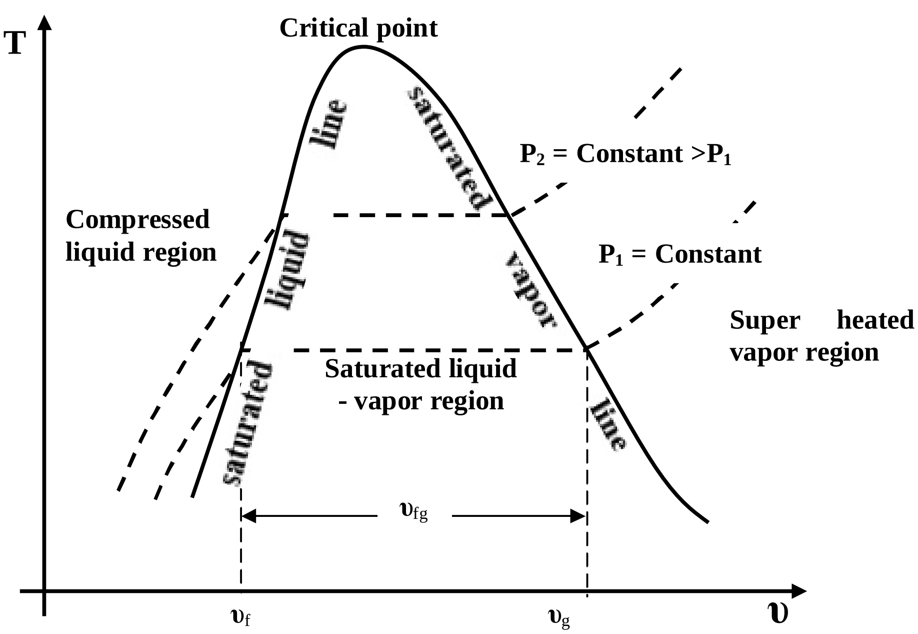
Saturated liquid: if a pure substance exists as liquid at saturation temperature and pressure, it is called saturated liquid.

Wet mixture: is the mixture of liquid and its vapor.

Saturated vapor: if a pure substance exists as vapor at saturated temperature and pressure, it is called saturated vapor.

Moisture content: is the ratio of liquid mass to the total mass (mass of liquid and mass of vapor).

$$y = \frac{m_L}{m_V + m_L} = \frac{m_L}{m_f}$$



Dryness fracture (x): is a ratio of vapor mass to the total mass.

$$x = \frac{m_V}{m_V + m_L}$$

$\therefore \quad x + y = 1$

As shown in the above fig.

υ_f : specific volume of saturated liquid.

υ_g : specific volume of saturated vapor.

υ_{fg} : difference between υ_g and υ_f (that is , $\upsilon_{fg} = \upsilon_g - \upsilon_f$)

Enthalpy of vaporization (h_{fg}): or latent heat of vaporization, It represent the amount of energy needed to vaporize a unit mass of saturated liquid at a given temperature or pressure. It decreases as the temperature or pressure increases, and it is becomes Zero at the critical point.

Super heated vapor: When the temperature of the vapor is higher than the saturated temperature of this vapor is called super heated vapor.

Degree of super heated: is the difference between the saturated temperature and super heated temperature.

$$\text{Degree} = T_{sup.} - T_{sat.}$$

Enthalpy of water (h_f): is the enthalpy of heat absorbed by unit mass of water at constant pressure until it reaches to the temperature of vapor forming from (0 C°).

$$h_f = c\,(T - 0)$$

T: temp. of vapor forming.

c: specific heat of water (4.2 kJ/kg.k)

Enthalpy of dry steam (h_g): is the quantity of head which needed to change unit mass of water at (0 C°) to dry steam.

$$h_g = h_f + L(h_{fg})$$

Enthalpy of wet steam (h_x):

$$h_x = (1 - x)\, h_f + X\, h_g$$

This relation can also be expressed as

$$h_x = h_f + x\, h_{fg}$$

Where ($h_{fg} = h_g - h_f$)

Solving for quality, we obtain $$x = \frac{h_x - h_f}{h_{fg}}$$

Heat of super heated: it is a quantity of heat which added to saturated vapor in order to change it to super heated vapor.

$$Q_{sup} = h_{sup} - h_{set.\,Vapor}$$

$$Q_{sup} = Cp_{av.}\,(T_{sup} - T_{sat.\,vapor})$$

Enthalpy of super heated vapor:

$$h_{sup} = h_f + L(h_{fg}) + Cp\,(T_{sup} - T_{sat})$$

$$h_{sup} = h_g + Cp\,(T_{sup} - T_{sat})$$

Cp: specific heat of super heated vapor.

Specific volume of wet steam:

$$\upsilon = \frac{\text{volume of liquid} + \text{volume of dry vapor}}{\text{total mass of wet vapor}}$$

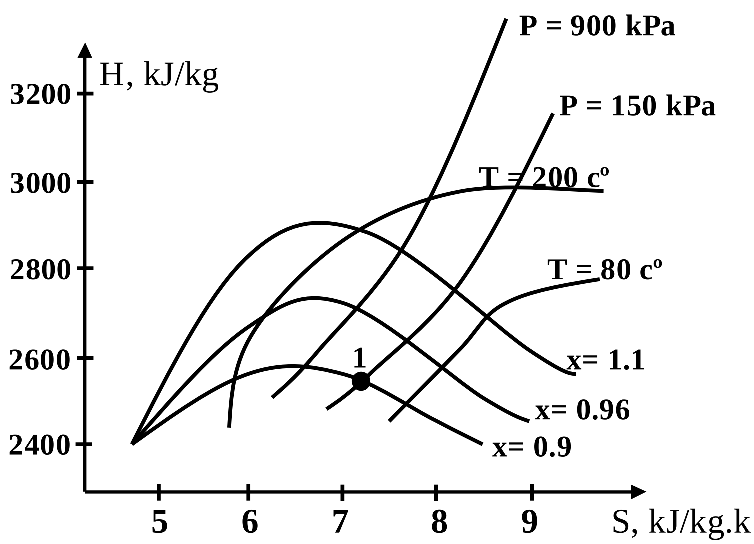
$$\upsilon = (1 - x)\,\upsilon_f + x\,\upsilon_g$$

or $\upsilon = \upsilon_f + x\,\upsilon_{fg}$

for incompressible fluid $\upsilon_f << \upsilon_g$ $\qquad \upsilon = x\,\upsilon_g$

The h - S diagram:

The h - S diagram is also called MOLLier diagram. The general features of h - S diagram are shown in the following fig. . The h - S diagrams are commonly used in practice to determine the properties of steam with reasonable accuracy.



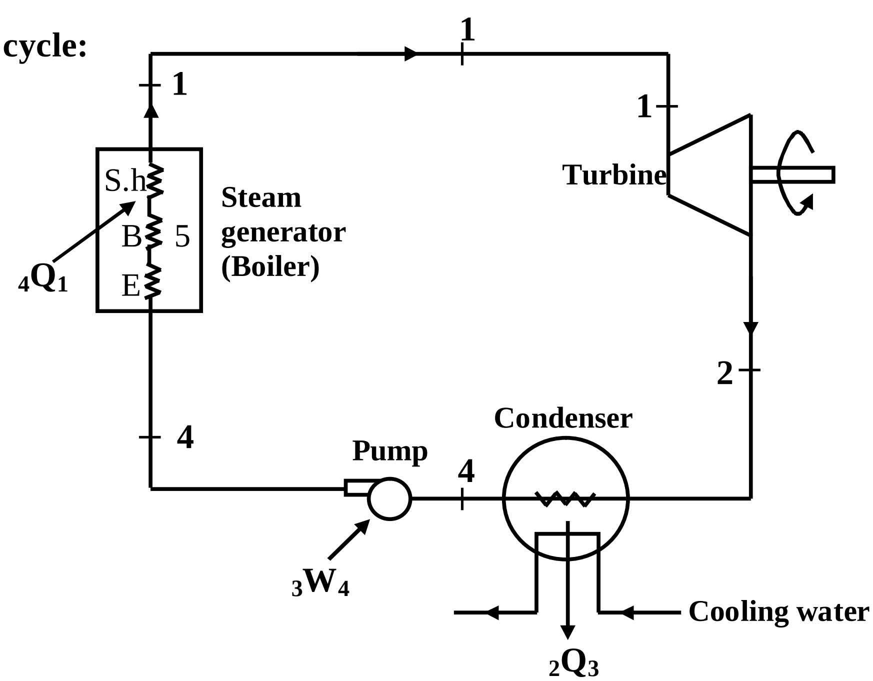
** Any two independent properties which appear on the chart are sufficient to define the state, (e.g. P= 150 kPa and x= 0.9 define the state 1, and h1 can be read off the vertical axis). In the super heat region, pressure and temperature can define the state.

Ex. find the enthalpy drop, when the steam expansion from 0.3 MPa pressure and 180 c° to 0.07 MPa through the isothermal process. Also find the final condition of the steam using Mollier diagram.

The steam power plant cycles:

Steam is most common working fluid used in vapor power plant cycles because of its many desirable characteristics, such as low cost, availability, and high enthalpy of vaporization. Steam power plants are commonly referred to as coal plant, nuclear plant, or natural gas plant, depending on the type of fuel used to apply heat to the steam. But the steam goes through the same basic cycle in all of them. Therefore, all can be analyzed in the same manner.

1- The ideal Rankine cycle:



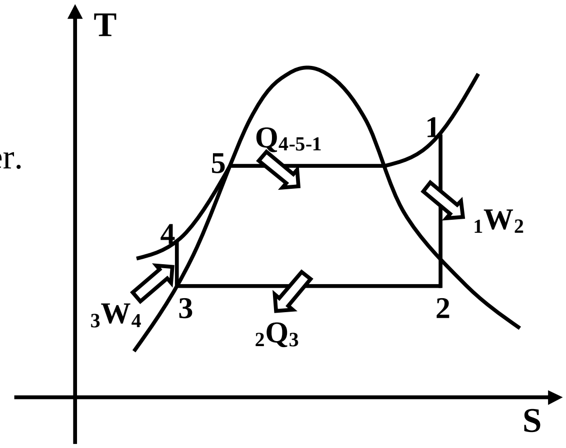
The ideal Rankine cycle dose not involves any internal irreversibilities and consists of the following four processes:

1-2 Isentropic expansion in a turbine

2-3 P= Constant , heat rejection in a condenser.

3-4 Isentropic compression in the pump.

4-5-1 P= Constant , heat addition in a boiler.



Each process can be analyzed using the steady flow energy equation, ΔKE and ΔPE may be neglected.

i. e. $h_i + Q = h_e + W$

Boiler: $h_1 + Q_{4.5.1} = h_1 + W$

$W = 0$, $Q_{4.5.1} = h_1 - h_4$

Turbine: $h_1 + {}_1Q_2 = h_2 + {}_1W_2$

$Q = 0$, ${}_1W_2 = h_1 - h_2$

Condenser: $h_2 + {}_2Q_3 = h_3 + W$

since $W = 0$, ${}_2Q_3 = -(h_2 - h_3)$

$\therefore$ heat rejected in condenser $= h_2 - h_3$

Pump: $h_3 + {}_3Q_4 = h_4 + {}_3W_4$

$Q = 0$, ${}_3W_4 = h_3 - h_4 = -(h_4 - h_3)$

Work input to pump $= (h_4 - h_3)$

${}_3W_4$ is a small quantity in comparison with ${}_1W_2$. Hence, it is usually neglected (especially when boiler pr. are low).

Net work don in the cycle, $W_{net} = {}_1W_2 - {}_3W_4$

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

Or, if the feed pump work (${}_3W_4$) is neglected,

$$W = (h_1 - h_2)$$

The heat supplied in the boiler, $Q_{4.5.1} = h_1 - h_4$

$$\text{Rankine efficincy } \zeta_R = \frac{\text{net work output}}{\text{heat supplied in the boiler}}$$

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

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

if the feed pump work $(h_4 - h_3)$ is neglected:

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

* When the feed pump work is to be included it is necessary to determine the quantity $({}_3W_4)$

$$\text{Pump work} = -{}_3W_4 = (h_4 - h_3)$$

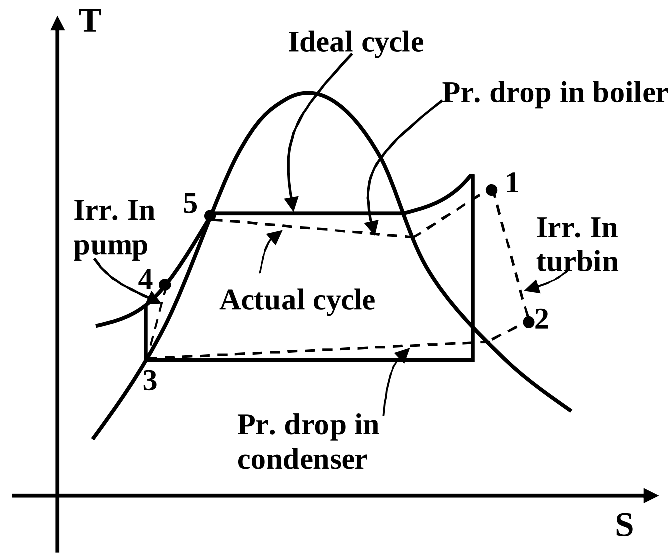
** For liquid which is assumed to be incompressible (i.e. V=Constant):

$$h_4 - h_3 = \upsilon (P_4 - P_3) \qquad \text{(prove that H.W.)}$$

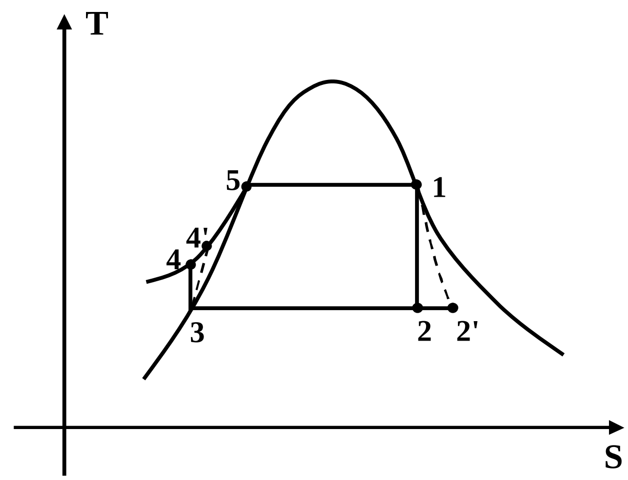
Where (υ) can be taken from tables for water at pressure P_3 .

Deviation of actual vapor cycles from idealized one:

The actual power vapor cycle defers from the ideal Rankine cycle as shown in the fig. as a result of irreversibility in various components. Fluid friction and undesired heat loses to the surrounding are the two most common sources of irreversibilities.



The actual expansion process (1-2') and the actual compression process (3-4') are irreversible as shown in the fig., the isentropic efficiency of process is defined by:



$$\zeta_T = \frac{W_a}{W_s} \quad \text{for an expansion process}$$

$$\zeta_P = \frac{W_s}{W_a} \quad \text{for an comprssion process}$$

$$\zeta_T(\text{turbine isentropic eff.}) = \frac{{}_1W_2'}{{}_1W_2'} = \frac{h_1 - h_2'}{h_1 - h_2}$$

$$\text{Work ratio} = \frac{\text{net Work}}{\text{gross Work}}$$

Specific steam consumption (s.s.c):

The steam flow indicates the size of plant and its component parts. S.S.C is a means where by the relative size different plants can be compared.

The S.S.C is the steam flow in kg/h required to develop 1kw

$$\text{s.s.c} = \frac{3600}{W_{net}} \quad \text{kg/kw.h}$$

Neglecting the feed pump work, we have W=h_1 - h_2

$$\text{s.s.c.} = \frac{3600}{h_1 - h_2} \quad \text{kg/kw.h}$$

Where (h_1 & h_2 are in kJ/kg)

===

Ex. 1: a steam power plant operates between a boiler pressure of 42 bar and a condenser pressure of 0.035 bar. Calculate for these limits the cycle efficiency, the work ratio and the specific steam consumption:

a- For Carnot cycle using wet steam.

b- For Rankine cycle with dry saturated steam at entry to the turbine.

c- For the Rankine cycle of (b) when the expansion process has an isentropic efficiency of 80%.

Sol.

(a) A Carnot cycle is shown in fig. 7.5.

T_1 saturation temperature at 42 bar
$= 253{\cdot}2 + 273 = 526{\cdot}2$ K

T_2 = saturation temperature at 0·035 bar
$= 26{\cdot}7 + 273 = 299{\cdot}7$ K

Then from equation 6.1

$$\eta_{\text{Carnot}} = \frac{T_1 - T_2}{T_1} = \frac{526{\cdot}2 - 299{\cdot}7}{526{\cdot}2} = 0{\cdot}432 \quad \text{or} \quad 43{\cdot}2\%$$

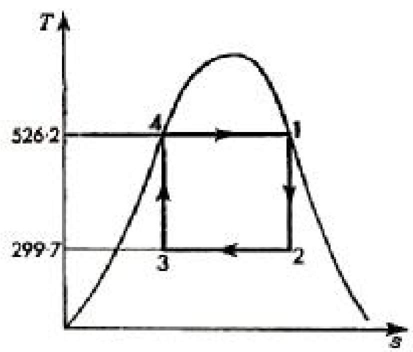


Fig. 7.5

Also,

Heat supplied $= h_1 - h_4 = h_{fg}$ at 42 bar $= 1698$ kJ/kg

Then, $\eta_{\text{Carnot}} = \dfrac{W}{Q} = 0{\cdot}432 \qquad \therefore\ W = 0{\cdot}432 \times 1698$

i.e. $W = 734$ kJ/kg

To find the gross work of the expansion process it is necessary to calculate h_2, using the fact that $s_1 = s_2$.

From tables,

$$h_1 = 2800 \text{ kJ/kg} \quad \text{and} \quad s_1 = s_2 = 6{\cdot}049 \text{ kJ/kg K}$$

Using equation 5.10

$$s_2 = 6{\cdot}049 = s_{f_2} + x_2 s_{fg_2} = 0{\cdot}391 + x_2 8{\cdot}13$$

$$\therefore\ x_2 = \frac{6{\cdot}049 - 0{\cdot}391}{8{\cdot}13} = 0{\cdot}696$$

Then using equation 3.2

$$h_2 = h_{f_2} + x_2 h_{fg_2} = 112 + 0{\cdot}696 \times 2438 = 1808 \text{ kJ/kg}$$

Hence, from equation 7.2

$$W_{12} = (h_1 - h_2) = (2800\text{–}1808) = 992 \text{ kJ/kg}$$

Therefore we have, using equation 7.13,

$$\text{Work ratio} = \frac{\text{net work}}{\text{gross work}} = \frac{734}{992} = 0{\cdot}739$$

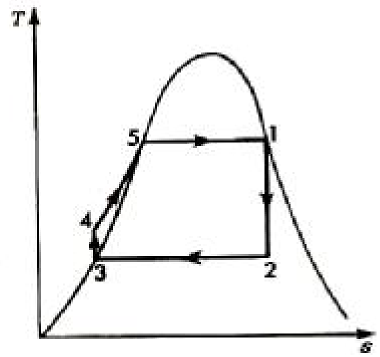


Fig. 7.6

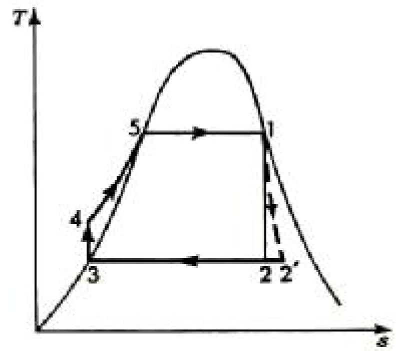


Fig. 7.7

Using equation 7.14

$$\text{Specific steam consumption, s.s.c.} = \frac{3600}{W} = \frac{3600}{734}$$

i.e. $\quad \text{s.s.c.} = 4{\cdot}9 \text{ kg/kW h}$

(b) The Rankine cycle is shown in fig. 7.6.

As in part (a)

$$h_1 = 2800 \text{ kJ/kg} \quad \text{and} \quad h_2 = 1808 \text{ kJ/kg}$$

Also, $\quad h_3 = h_f \text{ at } 0{\cdot}035 \text{ bar} = 112 \text{ kJ/kg}$

Using equation 7,10, with $v = v_f$ at 0·035 bar

$$\text{Pump work} = v_f(p_4 - p_3) = 0{\cdot}001 \times (42 - 0{\cdot}035) \times \frac{10^5}{10^3}$$
$$= 4{\cdot}2 \text{ kJ/kg}$$

Using equation 7.2

$$W_{12} = h_1 - h_2 = 2800 - 1808 = 992 \text{ kJ/kg}$$

Then using equation 7.8

$$\eta_R = \frac{(h_1-h_2)-(h_4-h_3)}{(h_1-h_3)-(h_4-h_3)} = \frac{992-4\cdot2}{(2800-112)-4\cdot2} = 0\cdot368$$

i.e. $\eta_R = 36\cdot8\%$

Using equation 7.13

$$\text{Work ratio} = \frac{\text{net work}}{\text{gross work}} = \frac{992-4\cdot2}{992} = 0\cdot996$$

Using equation 7.14

$$\text{Specific steam consumption, s.s.c.} = \frac{3600}{W}$$

i.e. $$\text{s.s.c.} = \frac{3600}{992-4\cdot2} = 3\cdot64 \text{ kg/kW h}$$

(c) The cycle with an irreversible expansion process is shown in fig. 7.7.

Using equation 7.12

$$\text{Isentropic efficiency} = \frac{h_1-h_{2'}}{h_1-h_2} = \frac{W_{12'}}{W_{12}}$$

$$\therefore\ 0\cdot8 = \frac{W_{12'}}{992}$$

i.e. $$W_{12'} = 0\cdot8\times992 = 793\cdot6 \text{ kJ/kg}$$

Then the cycle efficiency is given by

$$\text{Cycle efficiency} = \frac{(h_1-h_{2'})-(h_4-h_3)}{\text{heat supplied}}$$

$$= \frac{793\cdot6-4\cdot2}{(2800-112)-4\cdot2} = 0\cdot294$$

i.e. $$\text{Cycle efficiency} = 29\cdot4\%$$

$$\text{Work ratio} = \frac{W_{12'}-\text{pump work}}{W_{12'}} = \frac{793\cdot6-4\cdot2}{793\cdot6} = 0\cdot995$$

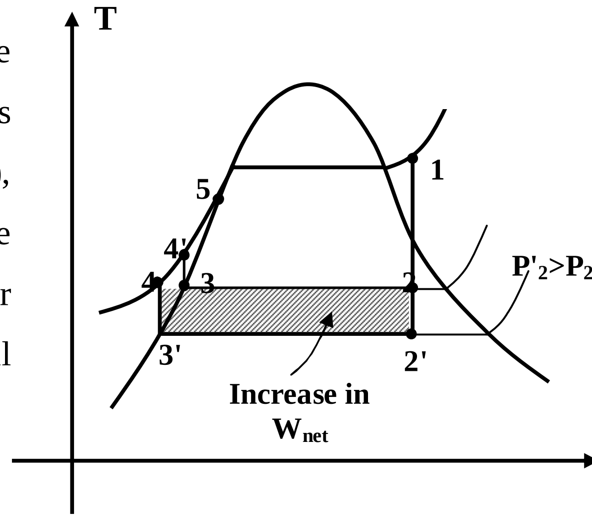
Also,

$$\text{s.s.c.} = \frac{3600}{793\cdot6-4\cdot2} = \frac{3600}{789\cdot4} = 4\cdot56 \text{ kg/kW h}$$

How can we increase the efficiency of the Rnkine cycle?

Steam power plants are used for the production of most of the electric power in the world, therefore, as small increases in thermal efficiency can mean large savings from the fuel requirement.

1- Lowering the condenser pressure:



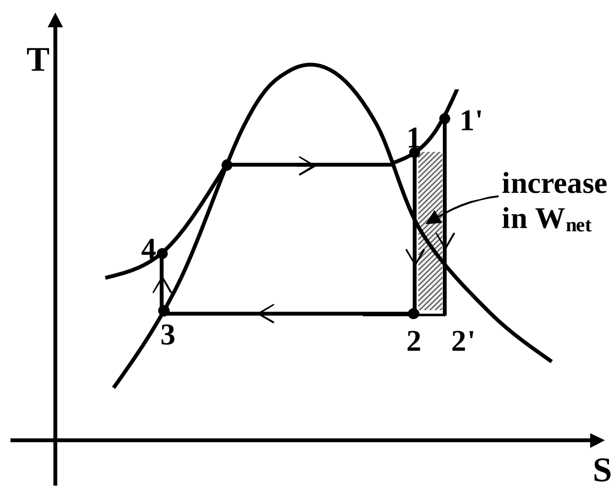
The dashed area in this diagram represents the increase in W_{net}. The heat input also increases (represented by the area under curve 4 - 4'), but this increase is very small. Thus, the overall effect of lowering the condenser pressure is an increase in the thermal efficiency of the cycle.

Disadvantages:

1- It creates the problem of air leakage into the condenser.

2- It increases the moisture content of the steam at the final-stages of the turbine. The large quantities of moisture are highly undesirable because it erodes the turbine blades.

2- Superheating the steam to high temperature:



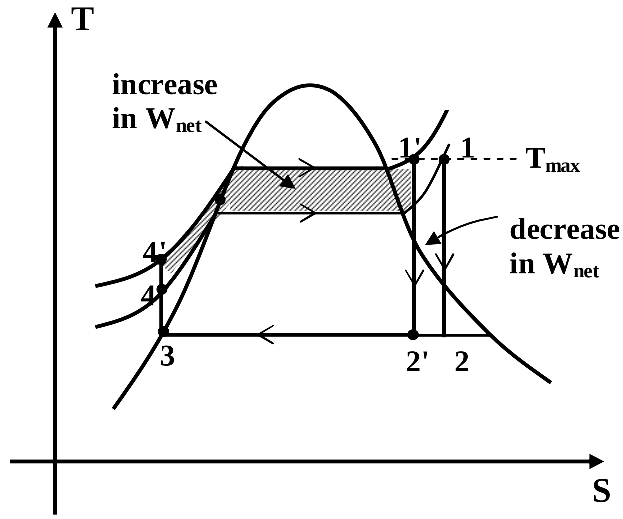
Both the W_{net} and heat input increase as a result of superheating. The overall effect is an increase in (ζ_R).

* Superheating decreasing the moisture content of the steam at the turbine exit as shown in T-S diagram (the quality (x) at state 2' is higher than that at state 2).

** Presently the highest steam temperature allowed at the turbine inlet is about 620 c°. Any increase in this value depends on improving the present material of the blades.

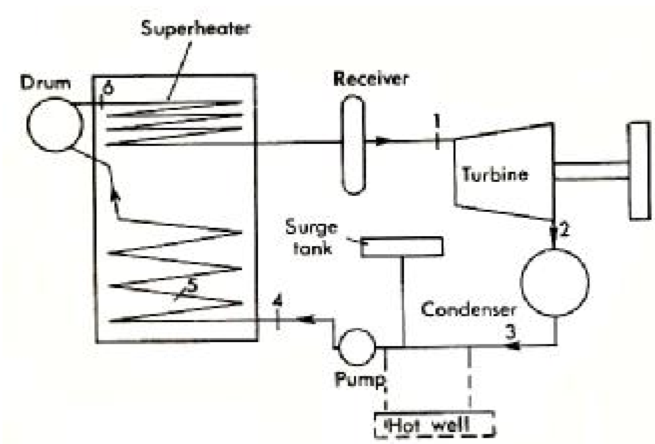
3- Increasing the boiler pressure:

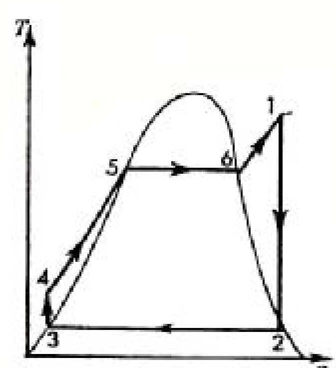
It raises the average temp. at which heat is added to the steam and thus, raises the (ζ). For a fixed turbine inlet temp., the cycle shifts to the left and the moisture content of the steam at the turbine exit increases. This side effect can be corrected by reheating the steam.



Rankine cycle with superheat:

The average temp. in the boiler can be increased by superheating the steam. Usually, the dry saturated steam from the boiler drum is passed through a second bank of smaller bore tubes within the boiler. The Rankine cycle with superheat in fig. below:





The superheater bank is situated such that it is heated by the hot gases from furnace until the steam reaches the required temp.

Ex. 2: Compare the Rankine cycle performance of Ex. 1 with that obtained when the steam is superheated to 500 c°. Neglect the feed pump work.

Sol.:

From tables, by interpolation, at 42 bar:

$$h_1 = 3442{\cdot}6 \text{ kJ/kg} \quad \text{and} \quad s_1 = s_2 = 7{\cdot}066 \text{ kJ/kg K}$$

Using equation 5.10

$$s_2 = s_{f_2} + x_2 s_{fg_2} \qquad \therefore\ 0{\cdot}391 + x_2 8{\cdot}13 = 7{\cdot}066$$

i.e. $$x_2 = 0{\cdot}821$$

Using equation 3.2

$$h_2 = h_{f_2} + x_2 h_{fg_2} = 112 + (0{\cdot}821 \times 2438) = 2113 \text{ kJ/kg}$$

From tables $$h_3 = 112 \text{ kJ/kg}$$

Then, using equation 7.2

$$W_{12} = h_1 - h_2 = 3442{\cdot}6 - 2113 = 1329{\cdot}6 \text{ kJ/kg}$$

Neglecting the feed pump term, we have

$$\text{Heat supplied} = h_1 - h_3 = 3442{\cdot}6 - 112 = 3330{\cdot}6 \text{ kJ/kg}$$

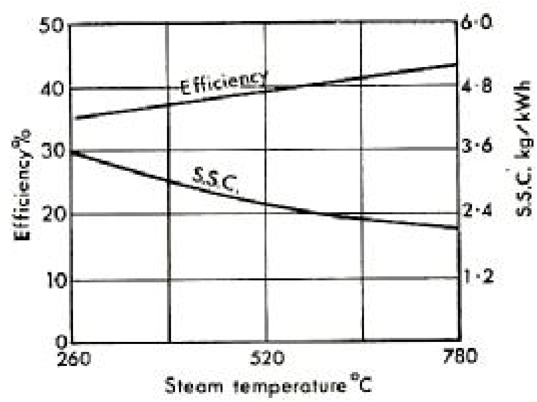
Using equation 7.9

$$\text{Thermal efficiency} = \frac{h_1 - h_2}{h_1 - h_3} = \frac{1329{\cdot}6}{3330{\cdot}6} = 0{\cdot}399 \quad \text{or} \quad 39{\cdot}9\%$$

Also, using equation 7.14

$$\text{s.s.c.} = \frac{3600}{W_{12}} = \frac{3600}{1329{\cdot}6} = 2{\cdot}71 \text{ kg/kW h}$$

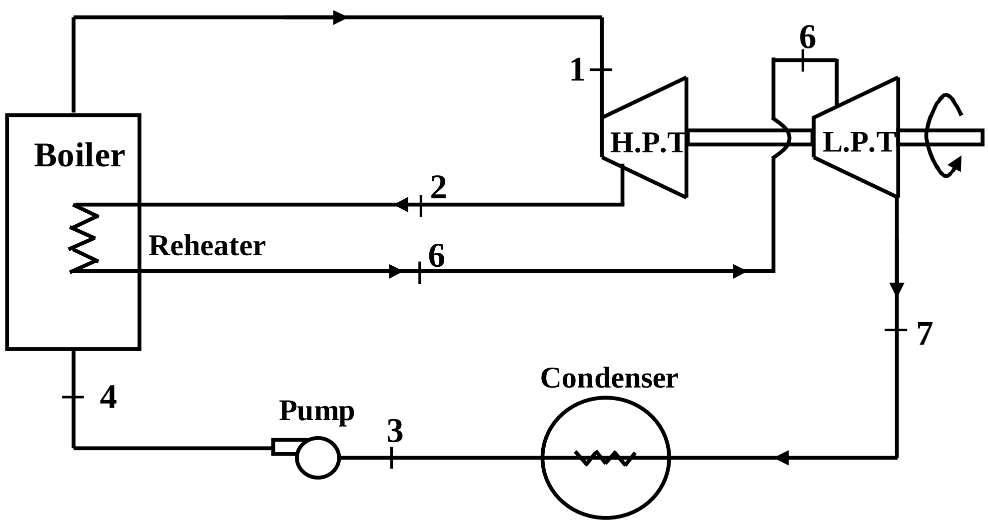
* For given boiler and condenser pressure as the superheat temperature increase, the (ζ_R) increases and S.S.C decreases as shown in the fig.



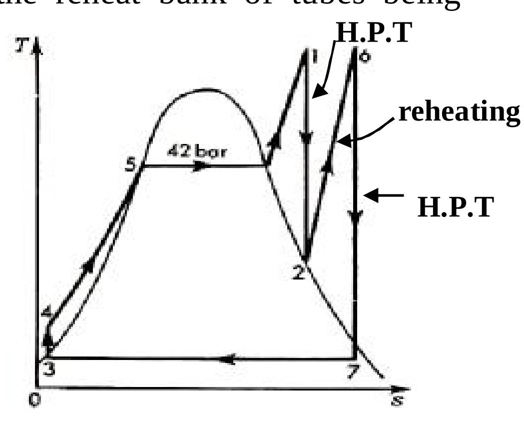
The Reheat Rnakine cycle:

It is desirable to keep the steam as dry as possible in the lower pressure stages of the turbine. The wetness at exhaust should be not greater than 10%. The increasing of boiler pressure increases the thermal efficiency, but it also increases the wetness of the exhaust steam to unacceptable levels.

The exhaust steam can be improved by REHEAT the steam, the expansion being carried out in two stages.



The reheat can be carried out by returning the steam to the boiler, and passing it through a special bank of tubes, the reheat bank of tubes being situated in the proximity of the superheat tubes.



1-2 isentropic expansion in H.P.T

6-7 isentropic expansion in L .P.T

2-6 reheated steam at constant pressure process.

Heat supplied = Q_{451}+ Q_{26}

$Q_{451} = h_1+h_3$, $Q_{26} = h_6+h_2$ (for reheated process)

Neglecting the feed pump work,

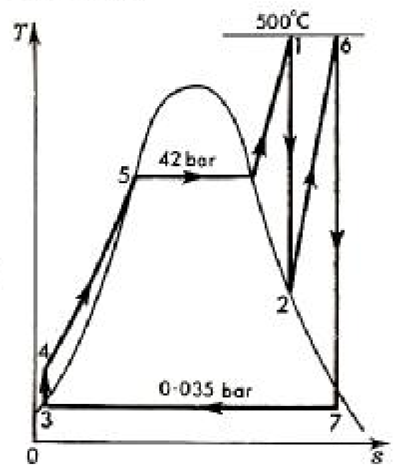
Work output = 1W2 + 6W7 = $(h_1- h_2) + (h6 - h_7)$

$$\text{cycle efficiency} = \frac{{}_1W_2 + {}_6W_7}{Q_{451} + Q_{26}} = \frac{(h_1 - h_2) + (h_6 - h_7)}{(h_1 - h_3) + (h_6 - h_2)}$$

Ex. 3: Calculate the ζ_R and S.S.C if reheat is included in the plant of ex. 2. the steam conditions at inlet to the turbine are 42 bar and 500 cº, and the condenser pressure is 0.035 bar as before. Assuming that the steam is just dry saturated on leaving the first turbine, and is reheated to its initial temperature. Neglect the feed pump term.

Sol.: By using (h - S) chart (Moiller diagram):

$h_1 = 3442{\cdot}6$ kJ/kg; $h_2 = 2713$ kJ/kg (at 2·3 bar); $h_6 = 3487$ kJ/kg (at 2·3 bar and 500°C); $h_7 = 2535$ kJ/kg.



From tables,

$$h_3 = 112 \text{ kJ/kg}$$

Then,

$$\text{Turbine work} = (h_1 - h_2) + (h_6 - h_7)$$
$$= (3443 - 2713) + (3487 - 2535)$$

i.e.
$$\text{Turbine work} = 1682 \text{ kJ/kg}$$

$$\text{Heat supplied} = (h_1 - h_3) + (h_6 - h_2)$$
$$= (3443 - 112) + (3487 - 2713)$$

i.e.
$$\text{Heat supplied} = 4105 \text{ kJ/kg}$$

$$\therefore \text{ Cycle efficiency} = \frac{1682}{4105} = 0{\cdot}41 \quad \text{or} \quad 41\%$$

Also,

$$\text{Specific steam consumption, s.s.c.} = \frac{3600}{W} = \frac{3600}{1682}$$

i.e.
$$\text{s.s.c.} = 2{\cdot}14 \text{ kg/kW h}$$

By comparing these answers with that in Ex. 2 , we get :

s.s.c. is redused from (2.71 kg/kw.hr) to (2.14 kg/kw.hr)

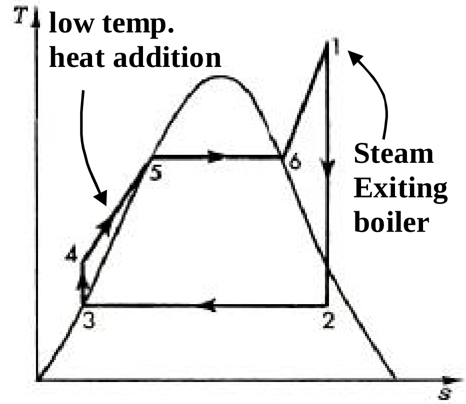
The effect of (ζ) is very small (it increased from 39.9% to 41%).

The regenerative Rankine cycle:

In simple Rankine cycle, the heat is added to the working fluid at relatively low temperature. This lowers the average temp. at which heat is added and thus lowers the cycle efficiency (see the figure below).

* Three ways can be discussed to raise the temp. of the liquid leaving the pump (called the feed water) before it enters the boiler.

The first pant of the heat addition process in the boiler takes place at relatively low temperature.

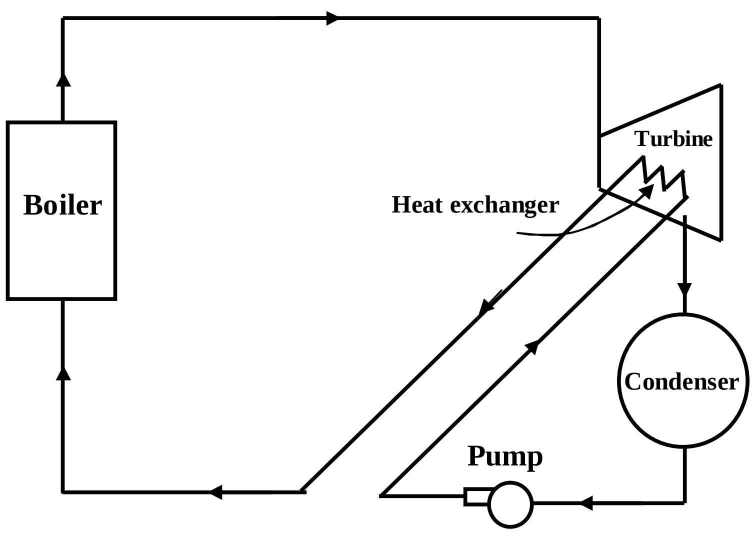


A: it is to compress the feed water isentropically to a high temperature as in the Carnot cycle. It is need extremely high pressure and is therefore not practical.

B: it is to transfer heat to the feed water from the expanding steam in the counter flow heat exchanger built into the turbine, that is, to use RREGENRATION. This solution is also not practical because:

1- It is difficult to design such a heat exchanger.

2- It will increase the moisture content of the steam at the final stages of the turbine. See the following fig.



C: Practical regeneration process in steam P.P. is accomplished by bleeding steam from the turbine at various points. This steam is used to heat the feed water. The device where the feed water is heated by regeneration is called a REGENERATOR, or FEED WATER HEATER.

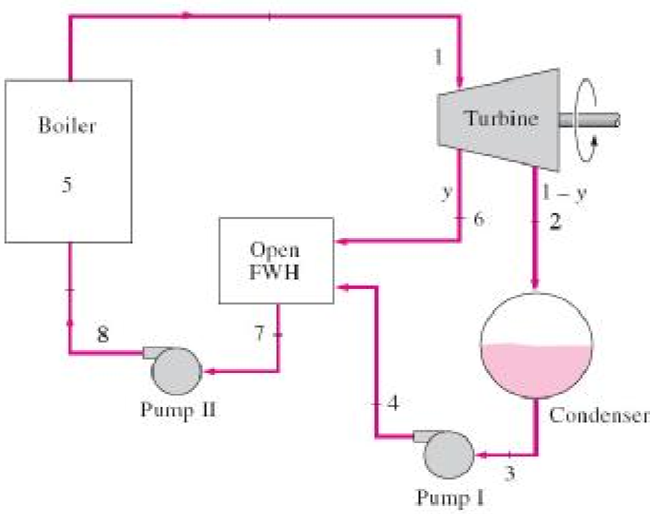
Note: regeneration not only improves cycle efficiency, but also provides a convenient means of de-aerating the feed water (i.e. removing the air that leaks in the condenser to prevent corrosion in the boiler).

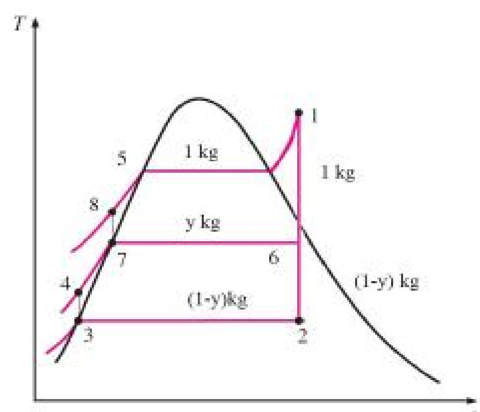
Feed water heaters:

A feed water heater is basically a heat exchanger where heat is transferred from the steam to the feed water either by the two fluid streams (open feed-water heaters) or without mixing them and (closed feed-water heaters).

Open feed-water heaters:

An open (or direct-contact) feed water heater is basically a MIXING CHAMBER, where the steam extracted from the turbine mixes with the feed water exiting the pump. Ideally, the mixture leaves the heater as saturated liquid at the heater pressure.





* It is necessary to determine the bleed (extract) pressure when one or more feed water heaters are used. It can be assumed that the bleed temperature is the mean temperature at 5 and 2 (see above fig.).

i.e. $$t_{bleed} = \frac{t_5 + t_2}{2}$$

q_{in}(heat supplied) = $h_1 - h_8$

$q_{out} = (1-y)(h_1 - h_3)$

$W_{turb.out} = (h_1 - h_6) + (1-y)(h_6 - h_2)$

$W_{pump.in} = (1-y)\ {}_3W_4 + {}_7W_8$

$$y = \frac{\dot{m}_6}{\dot{m}_1} \quad \text{(fraction of steam bleeded)}$$

${}_3W_4 = v_3 (P_4 - P_3)$

${}_7W_8 = v_7 (P_8 - P_7)$

The thermal efficiency of the Rankine cycle increases as a result of regeneration. This is because the generation raises the average temperature at which heat is added to the steam in the boiler by raising the temp. of the feed water before it inters the boiler. Many large P.P. in operation today use as many as eight feed water heaters (FWH). The optimum number of (FWH) is determine from economical consideration. The use additional (FWH) cannot be justified unless it saves more from the fuel costs than its own cost.

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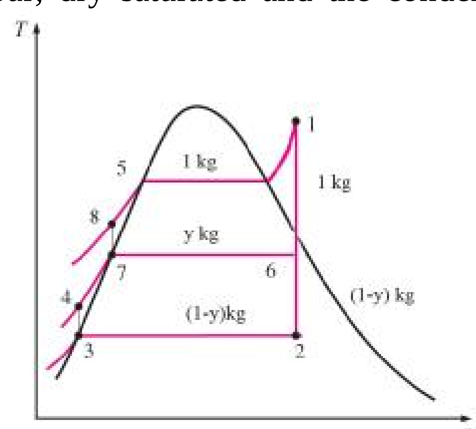
Ex. 4: if the Rankine cycle of Ex. 1 modified to include one feed water heater, calculate the cycle efficiency and the s.s.c.

Sol.: the steam enters the turbine at 42 bar, dry saturated and the condenser pressure is 0.035 bar.



At 42 bar, t_1=253.2 c^o

At 0.035 bar, t_2=26.7 c^o

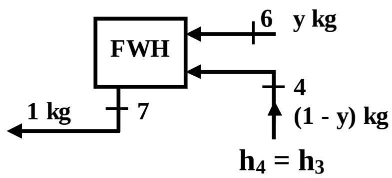
$$t_6 = \frac{253{\cdot}2 + 26{\cdot}7}{2} = 140°C$$

At t = 140 c^o, the bleed pr. ≈3.5 bar.

To determine the fraction y, consider the adiabatic mixing process at the (FWH).

Energy balance 1 ** the feed pump can be neglected (i.e. $h_4 = h_3$)

$y\, h_6 + (1+y)\, h_3 = 1 * h_7$



i.e. $$y = \frac{h_7 - h_3}{h_6 - h_3}$$

Now, $h_7 = 584$ kJ/kg; $h_3 = 112$ kJ/kg; and $s_1 = s_6 = s_2 = 6{\cdot}049$ kJ/kg K.

$$\therefore\ x_6 = \frac{6{\cdot}049 - 1{\cdot}727}{5{\cdot}214} = 0{\cdot}829$$

and $$x_2 = \frac{6{\cdot}049 - 0{\cdot}391}{8{\cdot}130} = 0{\cdot}696$$

Hence,

$$h_6 = h_{f_6} + x_6 h_{fg_6} = 584 + (0{\cdot}829 \times 2148) = 2364 \text{ kJ/kg}$$

and $$h_2 = h_{f_2} + x_2 h_{fg_2} = 112 + (0{\cdot}696 \times 2438) = 1808 \text{ kJ/kg}$$

$$\therefore\ y = \frac{584 - 112}{2364 - 112} = 0{\cdot}21 \text{ kg}$$

Neglecting the second feed pump term (i.e. $h_7 = h_8$), we have,

$$\text{Heat supplied in boiler} = (h_1 - h_7) = 2800 - 584 = 2216 \text{ kJ/kg}$$

Total work output, $W = W_{16} + W_{62} = (h_1 - h_6) + (1-y)(h_6 - h_2)$

i.e. $$\text{Work output} = (2800 - 2364) + (1 - 0{\cdot}21)(2364 - 1808)$$
$= 876$ kJ per kg of steam delivered by the boiler

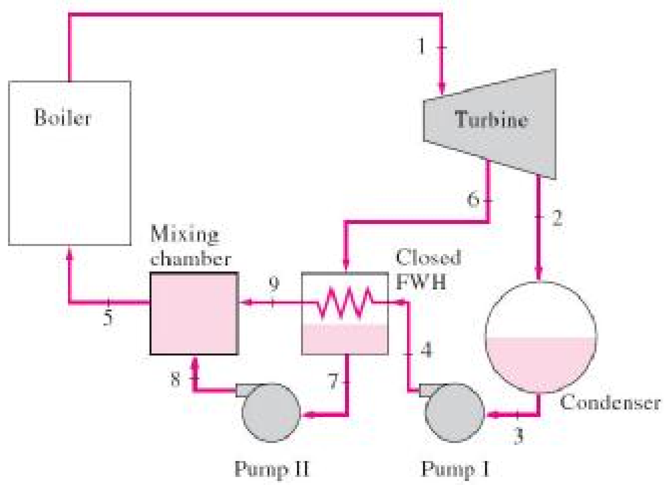
Therefore,

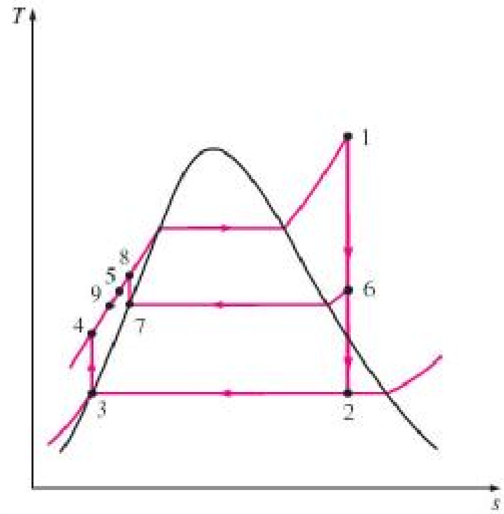
$$\text{Cycle efficiency} = \frac{W}{Q} = \frac{876}{2216} = 0{\cdot}396 \quad \text{or} \quad 39{\cdot}6\%$$

and $$\text{s.s.c.} = \frac{3600}{W} = \frac{3600}{876} = 4{\cdot}12 \text{ kg/kW h}$$

Closed feed water heaters:

In this type, the heat is transferred from the extracted (bleeded) steam to the feed water without any mixing taking place. The two streams now can be at difference pressures, since they do not mix. In an ideal closed (FWH), the feed water heated to exit temperature of the extracted (bleeded) steam, which ideally leaves the heater as a saturated liquid at the extraction (bleeding) pressure. See the following figs.

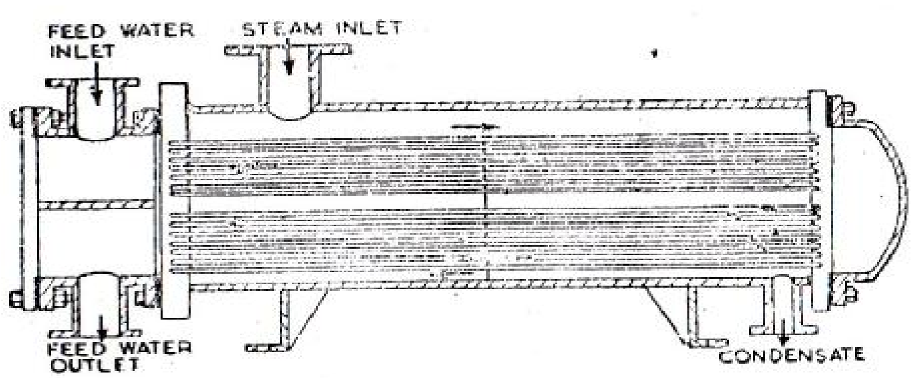




The condensed steam is then either pumped to the feed water line as shown in the above fig. or routed to another heater or to the condenser through a device called a TRAP. A trap allows the liquid to be throttled to a lower pressure region but traps the vapor (as shown in the following example).

Note: the enthalpy of steam remains constant during this throttling process.

* closed feed water heaters have usually shell and tube constructive and may be made horizontal or vertical. The construction of a closed FWH is shown in the fig.



* The closed feed water heaters are mostly used in large power stations, but one or two of open type are also used to serve the requirement of dearation of feed water.

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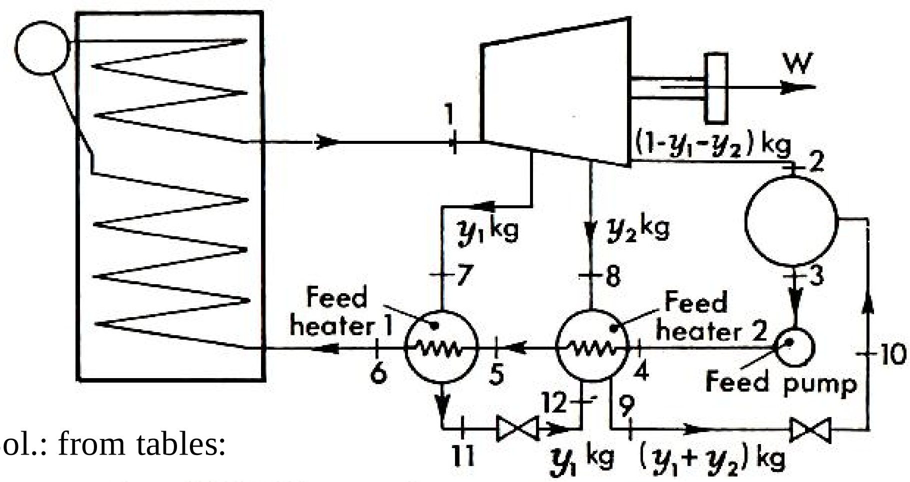
Ex. 5: In a regenerative steam cycle, employing two closed feed water heaters the steam is supplied to the turbine at 40 bar and 500 c° and is exhausted to the condenser at 0.035 bar. The intermediate pressures are obtained such that the saturation temperature intervals are approximately equal, giving pressure of 10 and 1.1 bar. Calculate the amount of steam bleed at each stage, the work output of the plant in kJ/kg of boiler steam and the thermal efficiency of the plant. Assume ideal process where required.



Sol.: from tables:

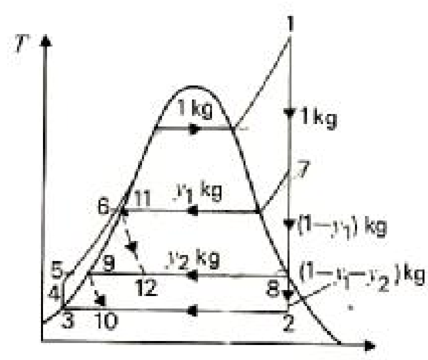
$$h_1 = 3445 \text{ kJ/kg} \quad \text{and} \quad s_1 = 7{\cdot}089 \text{ kJ/kg K} = s_2$$

At state 2, $\quad 0{\cdot}391 + x_2 \times 8{\cdot}13 = 7{\cdot}089$

$$\therefore \; x_2 = \frac{6{\cdot}698}{8{\cdot}13} = 0{\cdot}824$$

i.e. $\quad h_2 = 112 + 0{\cdot}824 \times 2438 = 2117 \text{ kJ/kg}$

Also, $\quad h_3 = h_f \text{ at } 0{\cdot}035 \text{ bar} = 112 \text{ kJ/kg}$



For the first stage of expansion, 1–7, $s_7 = s_1 = 7{\cdot}089$ kJ/kg K, and from tables at 10 bar $s_g < 7{\cdot}089$, hence the steam is superheated at state 7. By interpolation between 250°C and 300°C at 10 bar we have:

$$h_7 = 2944 + \left(\frac{7{\cdot}089 - 6{\cdot}926}{7{\cdot}124 - 6{\cdot}926}\right)(3052 - 2944) = 2944 + \frac{0{\cdot}163}{0{\cdot}198} \times 108$$

i.e.

$$h_7 = 3032{\cdot}9 \text{ kJ/kg}$$

For the throttling process, 11–12, we have:

$$h_6 = h_{11} = h_{12} = 763 \text{ kJ/kg}$$

For the second stage of expansion, 7–8, $s_7 = s_8 = s_1 = 7{\cdot}089$ kJ/kg K, and from tables at 1·1 bar $s_g > 7{\cdot}089$ kJ/kg K, hence the steam is wet at state 8. Therefore,

$$1{\cdot}333 + (x_8 \times 5{\cdot}994) = 7{\cdot}089$$

$$\therefore\ x_8 = 0{\cdot}961$$

i.e. $$h_8 = 429 + (0{\cdot}961 \times 2251) = 2591 \text{ kJ/kg}$$

For the throttling process, 9–10:

$$h_5 = h_9 = h_{10} = 429 \text{ kJ/kg}$$

Applying an energy balance to the first feed heater, remembering that there is no work or heat transfer:

$$y_1 h_7 + h_5 = y_1 h_{11} + h_6$$

i.e. $$y_1 = \frac{h_6 - h_5}{h_7 - h_{11}} = \frac{763 - 429}{3032{\cdot}9 - 763} = 0{\cdot}147$$

Similarly for the second heater, taking $h_4 = h_3$:

$$y_2 h_8 + y_1 h_{12} + h_4 = h_5 + (y_1 + y_2) h_9$$

i.e. $$y_2(h_8 - h_9) + y_1 h_{12} + h_4 = h_5 + y_1 h_9$$

$$y_2(2591 - 429) + (0{\cdot}147 \times 763) + 112 = 429 + (0{\cdot}147 \times 429)$$

$$\therefore\ y_2 = \frac{267{\cdot}8}{2162} = 0{\cdot}124$$

The heat supplied to the boiler, Q_1, per kg of boiler steam is given by:

$$Q_1 = h_1 - h_6 = 3445 - 763 = 2682 \text{ kJ/kg}$$

The work output, neglecting pump work, is given by:

$$\begin{aligned} W &= (h_1 - h_7) + (1 - y_1)(h_7 - h_8) + (1 - y_1 - y_2)(h_8 - h_2) \\ &= (3445 - 3032{\cdot}9) + (1 - 0{\cdot}147)(3032{\cdot}9 - 2591) \\ &\qquad + (1 - 0{\cdot}147 - 0{\cdot}124)(2591 - 2117) \\ &= 412{\cdot}1 + 376{\cdot}9 + 345{\cdot}5 = 1134{\cdot}5 \text{ kJ/kg} \end{aligned}$$

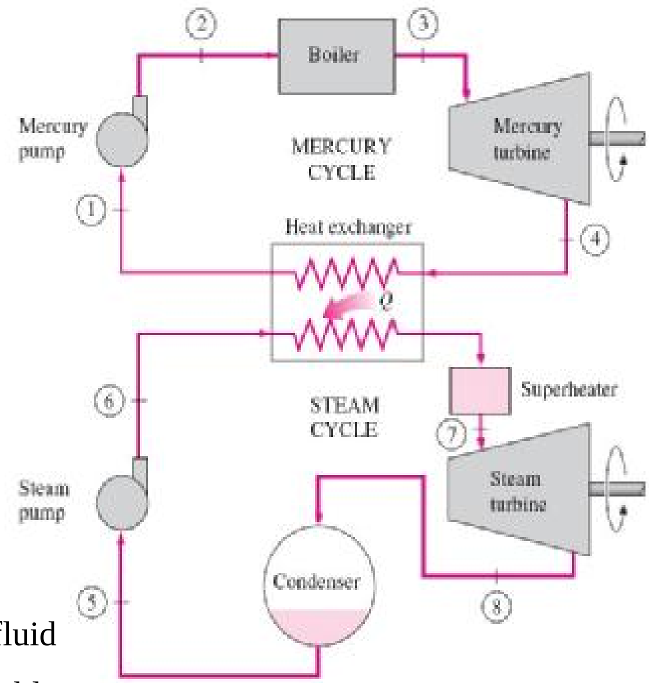
Then,

$$\text{Thermal efficiency} = \frac{W}{Q_1} = \frac{1134{\cdot}5}{2682} = 0{\cdot}423 \quad \text{or} \quad 42{\cdot}3\%$$

The binary vapor cycle:

The working fluid predominantly used in steam power plant is water. Water is the "best" working fluid presently available, but it is far from being the "ideal" one. The binary cycle is an attempt to overcome some the shortcomings of water and to approach the ideal working fluid by using two fluids.

* Water has low critical temp. (374 c^o, well below the metallurgical limit 620 c^o), and very high saturation pressures at high temp. (16.5 MPa at 350 c^o). Very high pressure is undesirable because it creates material strength problems.

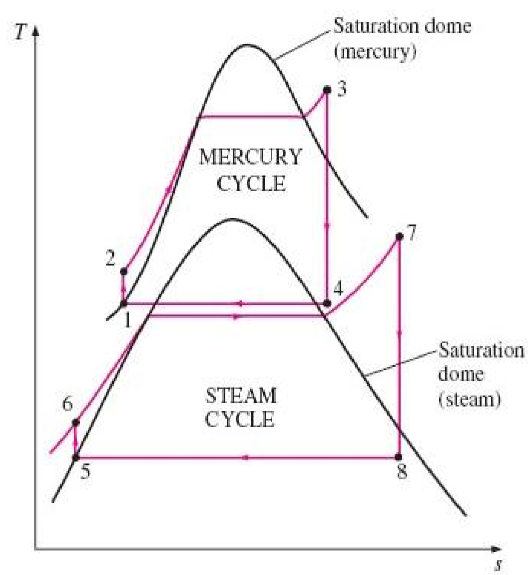


* The critical temp. of mercury is 898 c^o (well above the metallurgical limit 620 c^o) and its critical pressure is only about 18 MPa. Hence, mercury is very suitable working fluid for the "topping cycle" or high temp. cycle.

** Heat output of the high temp. cycle is used as the heat input to the low temp. cycle.

* Mercury is not suitable as the working fluid for the entire cycles because at an acceptable condenser pressure of 7 kPa the sat. temp. of mercury is 237 c^o (well above the normal temp. of condenser(about 620 c^o)). Therefore, the use of mercury as a working fluid is limited to the high temp. cycle.

* Water is suitable as the working fluid for the (bottoming cycle) or low temp. cycle since at 7 kPa condenser pr. , the sat. temp. of H_2O is 37 c^o (within the normal temp. of condenser).



Chapter 2

Steam Generator

The function of a steam generator or a boiler is to convert water into steam at the desired temp. and pressure to suit the turbine which it serves.

The basic components of steam generator are furnace and fuel burning equipment, water walls, boiler surface (drum and tubes), superheater surface, air heater (pre-heater) surface, re-superheater surface, economizer surface (feed water heating), and several accessories.

Boiler types:

1- Shell boiler. 2- Fire-tube boiler. 3- Water-tube boiler.

Shell boiler: in this type, the close tube or drum contents the water inside. The shell is attached with source of heating (such as electrical heater). Its efficiency and ability to generate the steam are low. It is usually used for simple applications as lab. The electrical boiler is one of this type.

Fire tube boiler: in this type, the hot combustion gases are passed inside the tubes, and the tubes are surrounded with water. The fire-tube boilers may be classified in several ways:

1- Externally or internally fired.
2- Horizontal, vertical or inclined.
3- Direct tube or return rube.

In the externally fired boilers, the furnace is places away from the boiler shell, while in the internally forced the furnace is built with the shell.

The horizontal, vertical and inclined designs refer to the arrangement of the drum and fire tubes in it.

In a direct through type of fire tube boiler, flue gases flow from the furnace end to the chimney end without changing ther direction, while in the return tube type the gases first flow to the rear and then come to the front through the fire tubes to a smoke box at the front.

The horizontal return tubular (HRT) boiler with external furnace is a common design in fire tube boilers, and is shown in the following fig.

Fig, construction of (HRT) boiler

It consist of a horizontal drum through which fire tubes are stretched, the tube ends being rolled into tube sheets at each end of the drum. The tubes are submerged in the water contained in the drum. Fuel burns below the drum on the grate and the combustion gases pass to the rear of the unit. From the rear ends gases pass through the tubes and leave at the front end of the boiler through a chimney.

Disadvantages:

1- Since the water and steam are held in a drum, an increase of working pressure need using of thicker plat sections for construction. The max. pr. is limited to only about (10.23 bar).
2- Longer time is required for steam rising due to large quantity of water in the drum.

Advantages:

Fire tube boilers are cheaper for smaller pressure and they are capable of meeting large fluctuations in steam demands due to grater water storage in the drum.

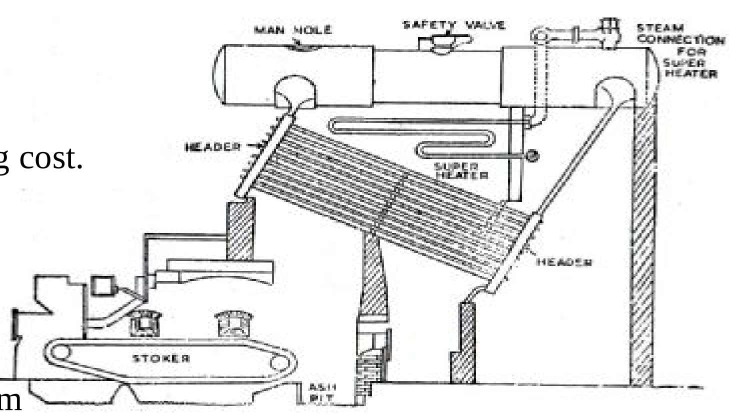
* Doors in the front and rear of the boiler drum provide access to the tubes for cleaning. The usual boiler fittings are mounted on the drum.

Water tube boiler:

For central steam P.P. with large capacities, the water-tube boilers are universally used (pressure is above 17 bar and steam generation is mor than 7000 kg/hr). In these boilers, water flows inside tubes and drums, and receives heat by radiation and conduction from combustion gases flowing over the tube surface. Water- tube boilers may be classified in several ways:

1- Straight or bent tube.
2- Longitudinal or cross tube.
3- Sectional or box header.
4- One or more drums.
5- Cross or parallel buffles.
6- Horizontal, vertical and inclined tubes.
7- Forced or natural circulation.

Straight tube units: the following fig, shows a longitudinal drum boiler with a stoker grate and superheater tubes. The straight inclined parallel tubes are rolled into header at each end, and placed below the longitudinal drum. The drum may be placed cross-wise with respect to the tubes. Incline tubes in parallel connect the two headers (box header). The rear header is provided at the bottom, called the mud drum, to collect solids in the boiler water.



Advantages:

* Lower manufacturing and erecting cost.
* Ease of inspection.
* Ease of cleaning.

** The capacity of longitudinal drum boiler is in the range of only about 2500 - 40000 kg/hr, while cross drum boiler, in which large number of tubes can be used, capacity is as high as 230000 kg/hr.

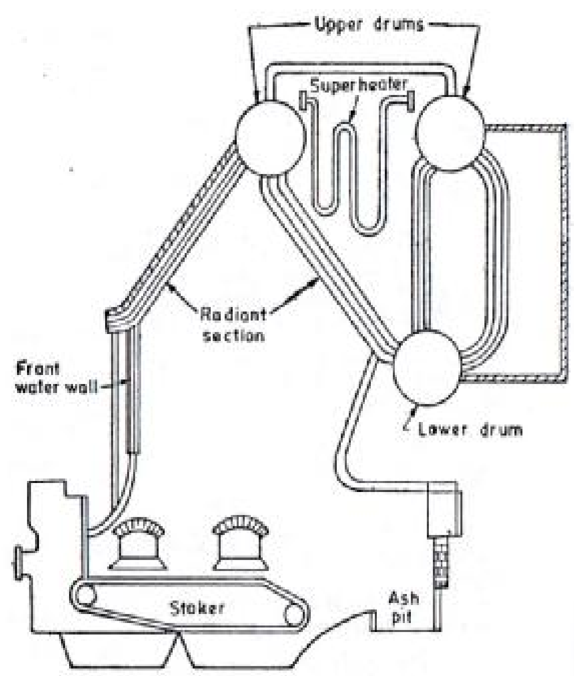
Bent tube units (header less):

The bent tube boiler is one of the most commonly used due to its simplicity. The headers are eliminated there by reducing maintenance costs. The bent tubes inter directly into the drum. They are arranged to give good circulation. The arrangement of the drums is done by placing some drums, high and one or two drums low. These low drums serve as mud drums. As shown in the following fig. , three drums are used, each at a different levels. Water circulates from the upper most drum to the lower most drum, then into intermediate drum, and finally to the upper drum which contains the direct steam.

* Tubes are cleaned from inside the drums, and only one or two manholes need be removed to get inside.

* This design is used for pressure of 32 bar and for capacities of about 18000 kg/hr.

* Water walls are used in these boilers to increase combustion rates and steaming capacity.



Schematic arrangement of a bent tube water boiler.

Water walls in modern boiler:

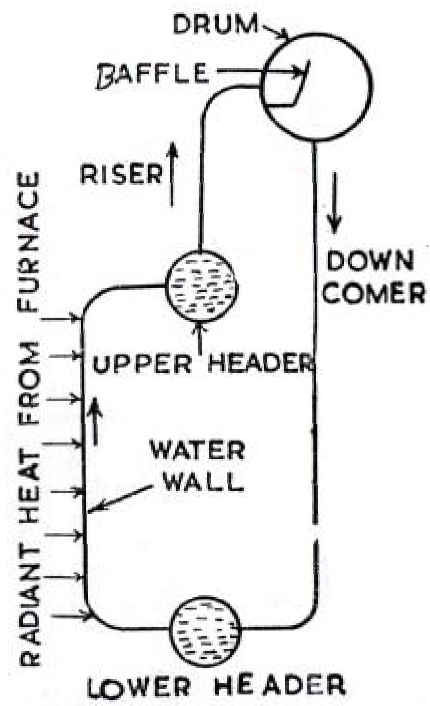
The use of water walls surrounding the furnace of the boiler permits higher combustion rates in the furnace. A water tube wall is a row of small diameters tubes through which water from the boiler is made to circulate constantly. Water walls are used to avoid erosion of tile and insulation the furnace. The furnace should be completely water-cooled on all four sides to full height.

Types of circulation in boiler:

A- Natural circulation:

Most of water-tubeboilers depend upon the natural circulation of water through the tubes. This type of circulation depends upon the difference of density between the water in down comer and mixture water and steam in riser.

The circulation is continued due to apply heat to the riser. The height of riser pipes and difference of average density is the base at which this type depends.



B- Forced circulation:

This type employing pumps to force the water in the tubes.

The advantages clamed for forced circulation are:

1- It can be used of smaller bore diameter and thiner tubes for the higher working pressure.

2- The tubes which constitute a bulk of heating surface can be disposed in a manner so as to obtain max. heat transfer.

3- Due to less weight, cost of foundation is reduced.

4- Scale formation in tubes reduced due to high velocity of water circulation.

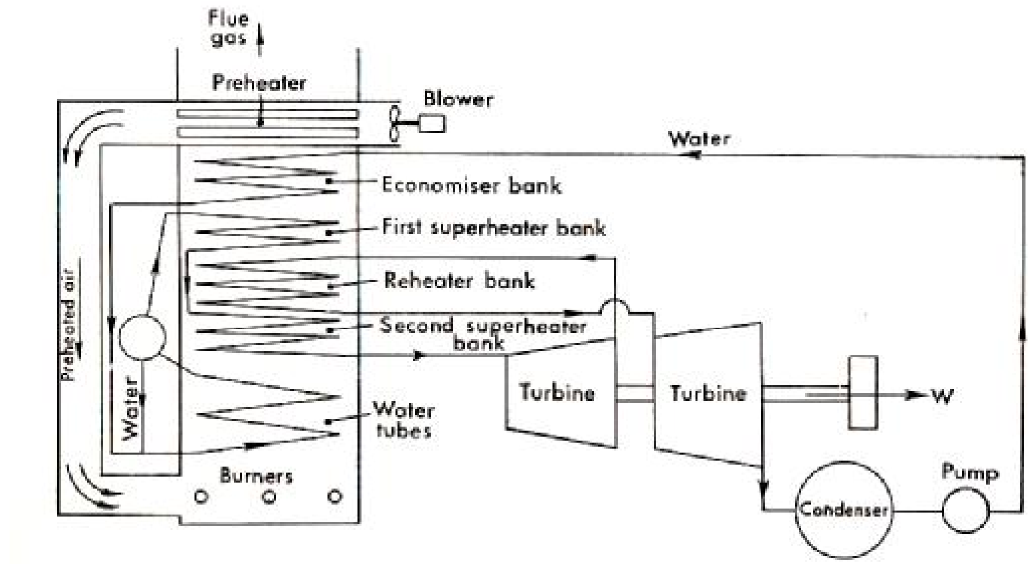
5- Because of rapid circulation of water, the boiler can be brought up to full load in a short time.

Economizer:

It is consist of coil placed in the flue gas stream. It is used to utilize some of the energy in the flue gas. The feed water enters at the top of the economizer coil, and as it is it is heated. See the following fig.

Pre-heater (air heater):

It consist of a coil (i.e. heat exchanger) placed in the flue gases. It can be used to pre-heat the air which is required for the combustion of the fuel. For a given temp. of combustion gases, the higher the initial temp. of the air then the less will be the energy input (i.e. less fuel will be used). Hence, higher plant efficiency will be obtained. See the following fig.



This fig. shows diagrammatically a plant with economizer, pre-heater, superheater, and reheater.

The advantage of using the economizer are:

1- Fuel economy: the economizer helps to save the energy which may be lost in the flue gas. This results in saving in fuel and in increase of overall efficiency of the boiler plant.

2- Increasing the steaming capacity: the economizer is in fact an extension of the boiler heating surface. Hence, the evaporation capacity of the boiler is increased.

3- Long life of the boiler: the range of temp. between the different parts of the boiler will be reduced. This results in reduction of stresses.

Feed water treatment:

Solid and dissolved matter in water deposits out on a heat transfer surface in steam generator. These materials save low conductivities and reduce heat transfer. In high temp. surface of steam generator, this this reduction in heat flow raises the temp. of the metal and may cause it to fail if the deposit is on the water side.

* The various impurities present in the raw water may be in the following form:

1- Desolved salts: such as chloride of calcium, sodium and magnesium, sulphates, and carbonates.
2- Dissolve gases: such as CO_2 and O_2.
3- Suspended matter: such as silica may be present as mud or slats. The slats of calcium and magnesium are very harmful.

The impurities may cause the following troubles:

1- Scale formation: it is either a hard or a soft deposit on the internal surface of a boiler. Scale is mainly due to salts of calcium and magnesium. The heat transfer through the heating surface will be reduced. The heating surfaces (tubes and drum) will get overheated. The scale formation in feed water pipes chokes the flow, which requires higher pressure to maintain the water flow.
2- Corrosion: it takes place due to present of O_2, CO_2 and chlorides dissolved in water. It produces pits, grooves, and cracks of material.
3- Foaming and priming: foaming prevent the free escape of steam pebbles as they rise to the surface of water. Priming is the passing of small water particles with steam as it leaves the boiler.

Methods of water treatment:

1- Mechanical:

a) Sedimentation. b) Coagulation. c) Filtration.

2- Chemical:

a) Lime soda treatment. b) Zeolite treatment. c) Deminarlizing.

3- Thermal treatment:

a) Evaporation: raw water is evaporated with the help of steam. Vapors are collected and condensed and pure water can be obtained.

b) Deaerating heating (degasification): the dissolved gases in the water (i.e. air, O_2, CO_2) and other gases can be removed deaerating heating. Both hotwell condensate and the treated make-up water requiring deaeration prior to passage to the boiler feed pumps. A tray type deaerating heater is shown in the fig.

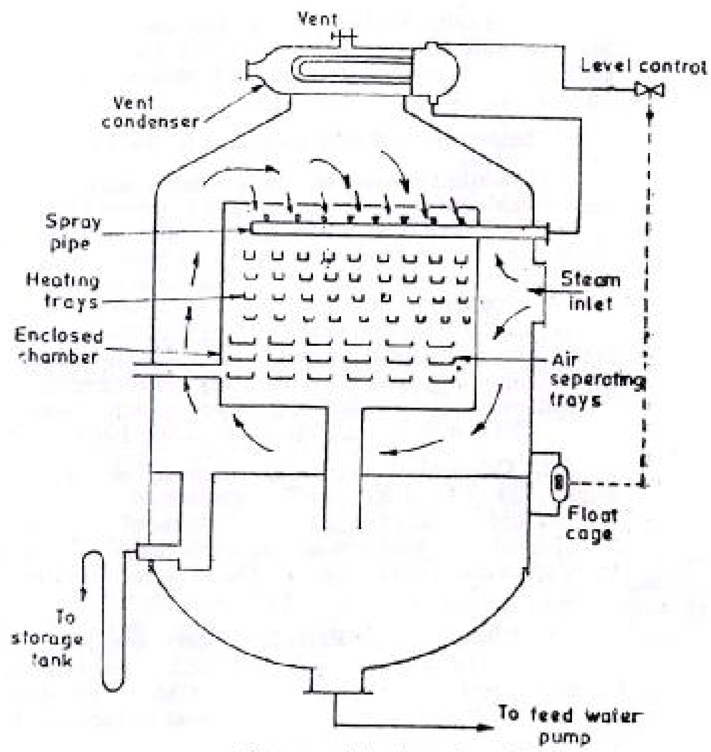


Diagram of tray type deaerating heater.

Boiler calculation:

$$\text{Boiler efficiency} = \frac{\text{heat transfer to working fluid}}{\text{fuel energy supplied}}$$

i.e. $$\zeta = \frac{[h_1 - (\text{enthalpy of the feed water})] \times m_s}{(\text{H.C.V. or L.C.V}) \times m_s}$$

Where h_1 is the enthalpy of the steam entering the turbine;

For wet steam $h_1 = h_f + x\ h_{fg}$

For superheated steam $\quad h_1 = h_f + L\ (h_{fg}) + Cp_s\ (t_{sup.} - t_g)$

$$h_1 = h_g + Cp_s\ (t_{sup.} - t_g)$$

Cp_s is the specific heat for superheated steam.

$$m = \frac{\dot{m}_s}{\dot{m}_f} \qquad \text{(kg of steam/kg of fuel)}$$

$\dot{m}_s$: mass flow rate of steam kg/s

$\dot{m}_f$: mass flow rate of fuel kg/s

Enthalpy of the feed water (h_w) = Cp $\dot{m}$ Δt above 0 c^o

$h_w = Cp * 1\ (t - 0) = 4.2\ t$

Equivalent evaporation: it is defined as the quant.

$$EE = \frac{\text{heat required to generate steam for one unity of fuel}}{\text{standerd unit of evaporation (i.e.2257.0)}}$$

Standard unit of evaporation:

The quantity of heat required to generate 1 kg of dry steam at 100 c^o and standard pressure (101.3 kPa). It is obtained from table (h_{fg}) at 100 c^o and 101.3 kPa, or it is a latent heat required to evaporate 1 kg at 100 c^o and 101.3 kPa.

$$\text{let} \quad \dot{m} = \frac{\dot{m}_s}{\dot{m}_f}$$

1- for wet steam:

$$EE = \frac{\dot{m}(h_f + xh_{fg} - h_w)}{2257}$$

2- for superheated steam:

$$EE = \frac{\dot{m}(h_g + Cp_s(t_{sup.} - t_g) - h_w)}{2257}$$

Ex. 1: the following data obtained from testing boiler (A) and (B), compare the two.

Boiler pressure (absolute pr. kPa), temp. of steam leaving the boiler (c°)

	Boiler (A)	Boiler (B)
Boiler pr. (in kPa absolute)	1721.3	1381.3
Temp. of steam leaving boiler c°	300	194
Dryness fracture of steam	--	0.9
Temp. of feeding water c°	60	30
Mass flow rate of steam kg/hr	9072	8165
Mass of coal which firing each hr kg/hr	907	816
Gross heat value of coal (MJ/kg) G.H.V.	33.5	33.5

If specific heat of super heated steam 2 kJ/kg.c° and for water 4.2 kJ/kg.c°, which boiler has higher E.E. up to 100 c° , what is the thermal efficiency for each one, which one is the best boiler?

Solution: Boiler (A)

$$EE = \frac{\dot{m}(h_g + Cp_s(t_{sup.} - t_g) - h_w)}{2257}$$

$$\dot{m} = \frac{\dot{m}_s}{\dot{m}_f} = \frac{9072}{907} = 10 \text{ kg of steam/kg of fuel}$$

From table at P=1.721 MPa, h_g = 2794 kJ/kg , t_g = 205 c°

$h_w = Cp * \Delta t = 4.2 * 60 = 252$ kJ/kg

$$EE = \frac{10(2794 + 2(300 - 205) - 252)}{2257} = 12 \text{ kg of steam/kg of fuel (ans.)}$$

$$\zeta_b = \frac{\dot{m}(h_g + Cp_s(t_{sup.} - t_g) - h_w)}{G.H.V} = \frac{10(2794 + 2(300 - 205) - 252)}{33.5 \times 10^3}$$

$$\zeta_b = 0.81 \quad \text{or} \quad 81\% \quad \text{(ans.)}$$

Boiler (B):

$$EE = \frac{\dot{m}(h_f + xh_{fg} - h_w)}{2257}$$

From table at P=1.381 MPa, we get h_f = 825 kJ/kg , h_{fg} = 1960 kJ/kg

$$\dot{m} = \frac{\dot{m}_s}{\dot{m}_f} = \frac{8165}{816} = 10 \text{ kg of steam/kg of fuel}$$

$h_w = Cp * \Delta t = 4.2 * 30 = 126$ kJ/kg

$$EE = \frac{10(825 + (0.9 * 1960) - 126)}{2257} = 10.9 \text{ kg of steam/kg of fuel} \quad \text{(ans.)}$$

$$\zeta_b = \frac{\dot{m}(h_f + xh_{fg} - h_w)}{G.H.V} = \frac{10(825 + (0.9 * 1960) - 126)}{33.5 \times 10^3} = 0.74 \quad \text{or} \quad 74\% \quad \text{(ans.)}$$

.: The boiler (A) is the best one.

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Ex. 2: a 5400 kg/hr of steam is supplied by a certain boiler plant at pressure (750 kPa) and dryness fraction 0.98. feed water is supplied at a temp. (41.5 c°). the mass of fuel (coal) is (670 kg/hr) with L.C.V = 31000 kJ/kg. Determine:

a) The boiler efficiency.
b) The equivalent evaporation E.E. at 100 c°.
c) The saving in coal per hour if the economizer is fitted in the flue gases (the feed water temp. can be raised to 100 c°), assuming the other conditions are the same.

Solution: at P = 7.5 bar and x = 0.98, h = 2725.33kJ/kg

h_w =173.825 kJ/kg at t = 41.5 c° , $m = \frac{5400}{670}$

a) $$\zeta = \frac{\dot{m}[h - h_w]}{L.C.V.} = \frac{5400[2725 - 173.825]}{670(31000)} = 66.33\%$$

b) $$E.E = \frac{m[h - h_w]}{2257} = 9.11 \text{ kg of steam/kg of fuel}$$

c) h = 2725 kJ/kg is the same , $h_w = 4.2 * 100 = 420$ kJ/kg

$$\zeta_b = \frac{m(h - h_w)}{m_f * L.C.V} \qquad \therefore m_f = \frac{5400(2725 - 420)}{31000 * 0.66} = 608.35 \text{kg/h r}$$

The saving fuel = 670 - 608.35 = 61.64 kg/hr (ans.)

Heat balance of boiler:

The study of heat balance represent an important step in the design of a steam generator. As a result of this study the optimum design conditions of steam are obtained.

* Useful heat absorbed by steam:

1- In economizer coil.
2- In super heater coil.
3- Reheater coil.

* Some items of heat losses:

1- Heat losses due to dry exhaust gases.
2- Heat losses due to moisture content in the fuel.
3- Heat losses due to CO formation.
4- Heat losses due to unburned fuel.
5- Heat losses by radiation from setting.
6- Unaccounting from any think.

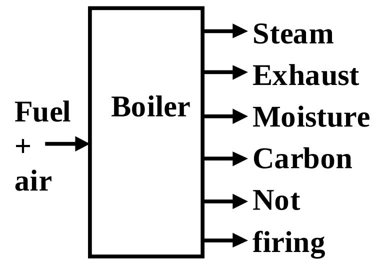
$$\% \text{ heat losses} = 100\% - \zeta_b$$

Ex. : a boiler generates the steam at (930 kPa) gage pr. and 0.9 dryness fracture.
The boiler test gives the following data:
Temp. of feeding water = 83.5 c^o.
Steam generated for each hour = 9072 kg
Coal firing for each hour = 907 kg
Moisture content in the coal = 2%   (i.e. for each kg of fuel)
Ash content at coal firing = 10% (i.e. for each kg of fuel)
Carbon content in ash = 20%
Dry exhaust gases for each kg of coal fire = 30 kg
Temp. Of exhaust gases = 238 c^o
Temp. of the boiler space = 24 c^o
Heating value of dry coal for each kg of air = 35 MJ
Specific heat value for exhaust gases = 1 kJ/kg. c^o
Heating value of carbon for each kg of air = 33.4 MJ
Atmospheric pressure = 101.3 kPa
For this data, find
a- the heat balance table represent all values of heat in kJ/kg of fuel.
b- the equivalent evaporation (EE) for this boiler above 100 c^o.
(Use Cp_{steam} = 2 kJ/kg. c^o , and Cp_w = 4.2 kJ/kg. c^o)

Solution:

Heat supplied by the fuel = (1 - 0.02)* 33900 = 33222 kJ/kg

Heat absorbed by the steam = $\dot{m}[h_f + x\,h_{fg} - h_w]$



Absolute pr. = 930 + 101.3 = 1031 kPa = 1.031 MPa

h_f = 767 kJ/kg , h_{fg} = 2009 kJ/kg

$$\dot{m} = \frac{\dot{m}_s}{\dot{m}_f} = \frac{9072}{907} = 10\,\text{kg of steam/kg of fuel}$$

$h_w = C_p\,\Delta t$ = 4.2 * 83.5 = 350.7 kJ/kg

.: heat absorbed by the steam = 10[767+0.9*2009-(4.2*83.5)] = 22244 kJ/kg

Now we calculate the heat losses

Heat losses due to moisture = $\dot{m}[h_f + C_p(\Delta t) - h_w]$

At t = 100 c° h_f = 419 kJ/kg Δt = (238 - 100) P = 101.3 kPa,

Cp_w * temp. of boiler space = 4.2*24 = h_w = 0.02[419.1+2(238-100) -(24*4.2)]

= 12 kJ/kg of fuel

Heat loss due to the exhaust gases = $\dot{m}C_p(\Delta t)$ = 30*1(238-24) = 6420 kJ/kg

Heat loss due to carbon not fired in ash= $\dot{m}$*G.H.V=0.1* 0.2* 33900 =678 kJ/kg

Type of heat	inlet (kJ/kg)	outlet (kJ/kg)
Heat of fuel	33222	-----
Heat of steam		22244
Heat loss due to moisture		12
Heat loss due to the exhaust gases		6420
Heat loss due to carbon not fired		678
		29354

Unaccounted values of heat losses (such as radiation losses) = 33222 - 29354

= 3868 kJ/kg

$$EE = \frac{22244}{2257} = 9.85\,\text{kg of steam/kg of fuel} \quad \text{(ans.)}$$