Subject : Thermodynamic II	موضوع : ديناميك الحرارة II
Weekly Hours : Theoretical: 2 UNITS: 5	الساعات الأسبوعية : نظري :2 الوحدات ؛5
Tutorial: 1	مناقشة : 1
Experimental : 1	عملي 1:

week	Contents	المحتويات	الأسبوع
1.	Definitions , phase change , pure substance	تعريفات / تغيير الطور / المادة النقية	.1
2.	=	=	.2
3.	Steam tables	جداول البخار	.3
4.	T-s , P v diagrams and h- s chart	رسومات T-s ،p-v او مخطط h-s	.4
5.	Steam process	إجراءات البخار	.5
6.	=	=	.6
7.	Dryness fraction measurement	قياس نسبة الجفاف	.7
8.	=	=	.8
9.	Steam cycles Carnot cycle	دورات البخار دورة كارنو	.9
10.	Rankine cycles	دورة رانكن	.10
11.	Ranking cycle with superheat	دورة رانكن ذو التسخين العالي	.11
12.	The effect of steam condition on thermal efficiency	تأثير حالة البخار على الكفاءة الحرارة	.12
13.	Reheat cycle	دورة اعادة التسخين	.13
14.	Regenerative cycle , open feed heater	دورة إعادة التنشيط / مسخن تغذية مفتوح	.14
15.	Regenerative cycle , closed feed heater	دورة إعادة التنشيط / مسخن تغذية مغلق	.15
16.	Air standard cycle ; Otto cycle	دورات الهواء القياسية : دورة اوتو	.16
17.	Air standard cycle ; Diesel cycle	دورات الهواء القياسية : دورة ديزل	.17
18.	Air standard cycle ; Daul cycle	دورات الهواء القياسية : الدورة المزدوجة	.18
19.	Mean effective pressure , and comparison between air standard cycles	متوسط الضغط الفعال والمقارنة بين دورات الهواء القياسية	.19
20.	Simple gas turbine cycle ; Brayton cycle	دورة التوربين الغازي البسيطة دورة براتن	.20
21.	Reciprocating compressors	الضواغط الترددية	.21
22.	The condition for minimum work	شرط الشغل الأدنى	.22
23.	Isothermal efficiency	الكفاءة الحرارية	.23
24.	Effect of clearance volume	تأثير حجم الخلوص	.24
25.	Volumetric efficiency	الكفاءة الحجمية	.25
26.	Actual indicator diagram	المخطط الإرشادي الحقيقي	.26
27.	Multi – stage compressor :Plus inter cooler	الضواغط متعدد المراحل : إضافة المبرد – الداخلي (البيني)	.27
28.	The ideal inter mediate pressure	الضغط المثالي الأوسط	.28
29.	Energy balance for two – stage machine with inter – cooler	معادلة الطاقة لضاغط ذو مرحلتين مع مبرد بيني	.29
30.	The steady flow analysis of the compressor process	تحليل الجريان المستقر لأجراء الانضغاط	.30

University of technology -   Mechanical Engineering Department
Thermodynamic II, 2nd year
Steam engineering
Prepared by: Dr. Arkan Kh. Altayee & Dr. Aabdulkareem A. Almusawi

References:-
1. Applied Thermodynamics for Engineering Technologists, by Estop, T. D. and McConky, A.
2. Engineering Thermodynamics, Work and heat Transfer, by Rogers, C. F. C. and Mayhew, Y. R.
3. Thermodynamics: An Engineering Approach, 3/e, by Yunus A. Cengel and Michael A. Boles

Fundamentals:

Steam: is water vapor at or above the boiling point. Steam has the widest application as working fluid due the following reasonable reasons:-
1. Water is the most widely spread substance in nature.
2. Water and steam possess relatively good thermodynamic properties.
3. Water and steam do not have harmful effect on metal and living organisms.

Evaporation: - is the process of vapor formation that occurs only from the surface of the liquid and it takes place at any temperature.

Boiling: - is the process of vigorous vapor formation throughout the entire mass of the liquid when a certain amount of heat is transferred to the liquid through the walls of the confining vessel.

Condensation: - is the process of transition from vapor to a liquid phase.

Saturation temperature: - is the temperature at which the boiling occurs at a certain given pressure. $(T = T_s , p = p_s)$

Unsaturated water: - water where its temperature is below the saturated temperature. $(T < T_s)$

Saturated steam: - also called "dry steam" is water liquid where it is completely evaporated, changed from liquid to vapor, at $(T = T_s, p = p_s)$.

Wet steam: - it is a mixture of saturated water and saturated steam at $(T = T_s, p = p_s)$.

Superheated steam: - is a steam at temperature higher than the saturated temperature $(T > T_s)$.

The student should learn about the following processes: Fusion (melting), Vaporization, Sublimation and Freezing (Solidification).

Changes of phase of matter of pure substance

Conceder a block of ice at low temperature in a piston-cylinder arrangement, as in figure (01), at constant pressure of (1 atm). As it is successively heated up, it under goes the phase changes shown in figure (02).

STATE 1
$P = 1$ atm
$T = 20°C$
Heat

STATE 2
$P = 1$ atm
$T = 100°C$
Heat

STATE 3
$P = 1$ atm
$T = 100°C$
Heat

STATE 4
$P = 1$ atm
$T = 100°C$
Heat

STATE 5
$P = 1$ atm
$T = 300°C$
Heat
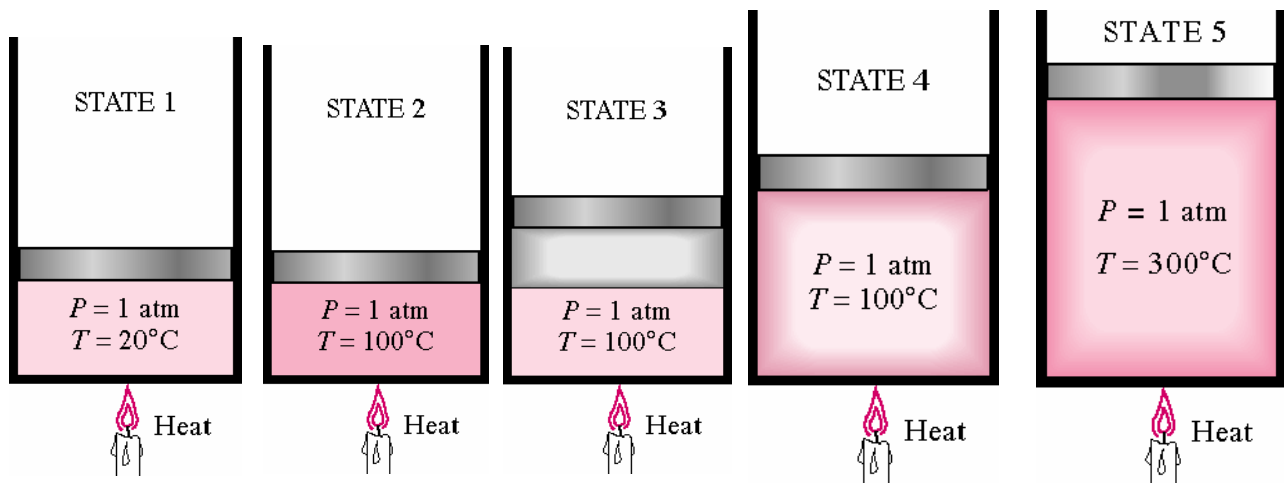
Figure (01) piston-cylinder arrangement
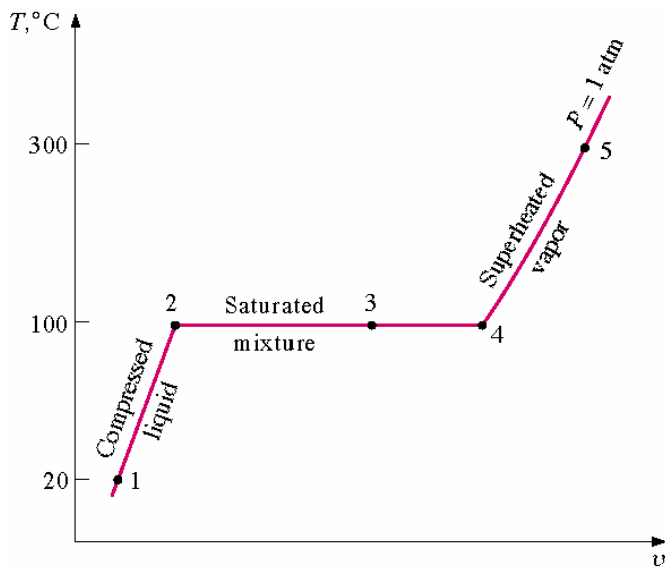
Figure (02) phase change due to heating

For other pure substance other than water, the behavior is the same as in figure (02). However, at low temperature, the behavior of water is different from that for other pure substance, as show in figure (03). Water gains a maximum volume at ($t = 4 \,^{\circ}\text{C}$).

Figure (03): Phase change difference for pure water and other pure substance.
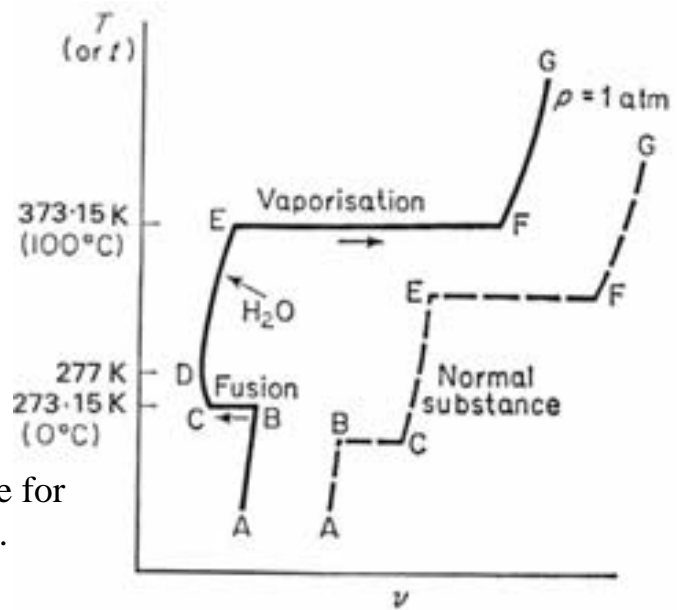
For engineering purpose, figure (02) is more related then figure (03). Heating Process (A-B-C-D-E-F-G), which is under constant pressure, can be repeated for different pressure from a value below triple point ($T_t = 273.16$ K , $p_t = 0.006112$ bar) to a value above critical value ($T_c = 647.3$ K , $p_c = 221.2$ bar, $v_c = 0.00317$ m^3 / kg).

Triple point is…
Critical point is...

3

Figure (04) shows different heating processes under different pressures.

Arrows (1) for boiling, (2) for sublimation and (3) for fusion denote the variation of temperature under different pressures, see figure (05).

Figure (04): Phase change for different heating processes under different pressures for pure water and other pure substance.

(a) Water   (b) Normal substance
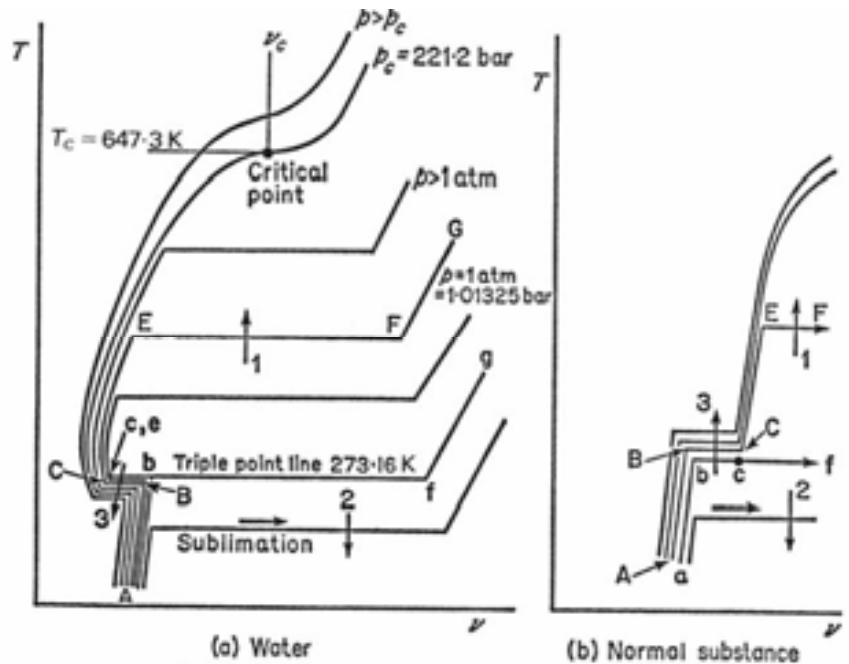
Figure (05): variations of boiling, sublimation and fusion temperatures with pressure for pure water and other pure substance starting from triple point.
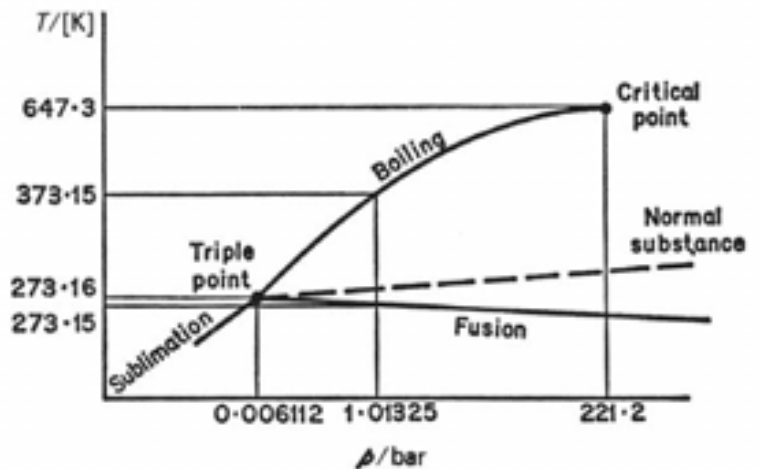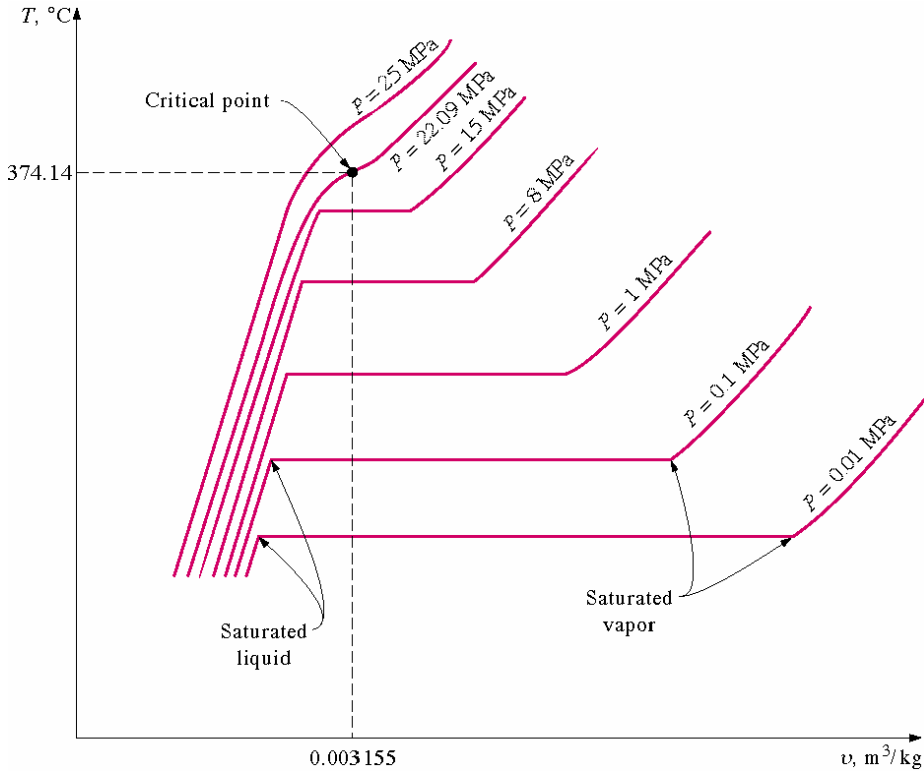
Figure (06): T-v diagram for pure water.

Figures (06) and (07), which are identical to figure (04-a), are used to denote saturation lines. This (T-v) diagram, which is drawn from experiment Results, is used to draw a more useful (T-s) and (p-v) diagrams.
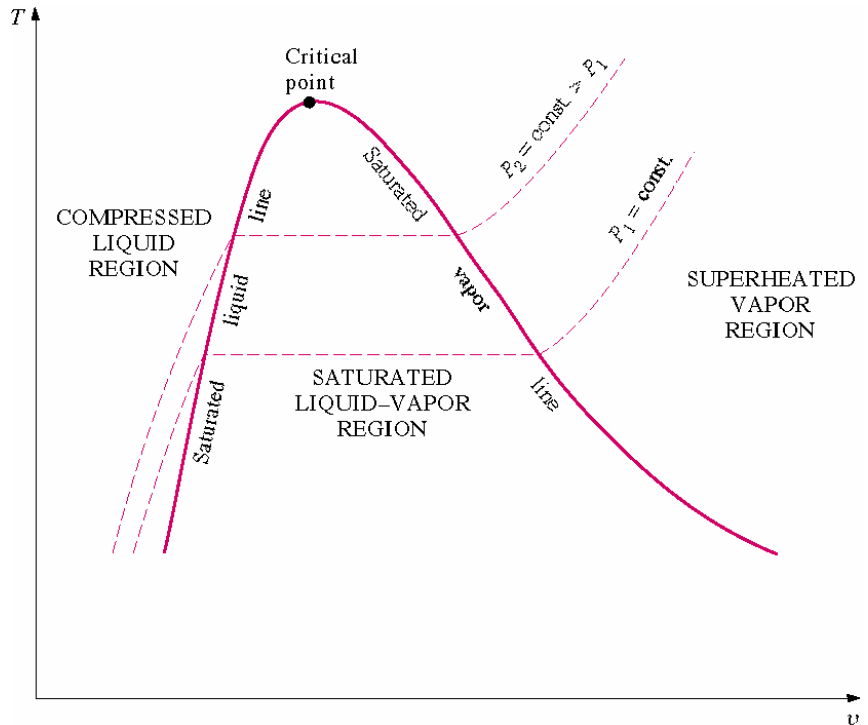
Figure (07): T-v diagram for pure water.

5

T-s and p-v diagrams:

T-v diagram can be transposed into a more useful (T-s) and (p-v) diagrams.

Figure (08): **p-v** diagram, where unsaturated water (liquid), wet steam (Liquid-vapour mixture) and superheated steam regions are denoted. In addition, saturated liquid line and saturated (dry) steam line are presented. Critical point and triple point line are also clear.
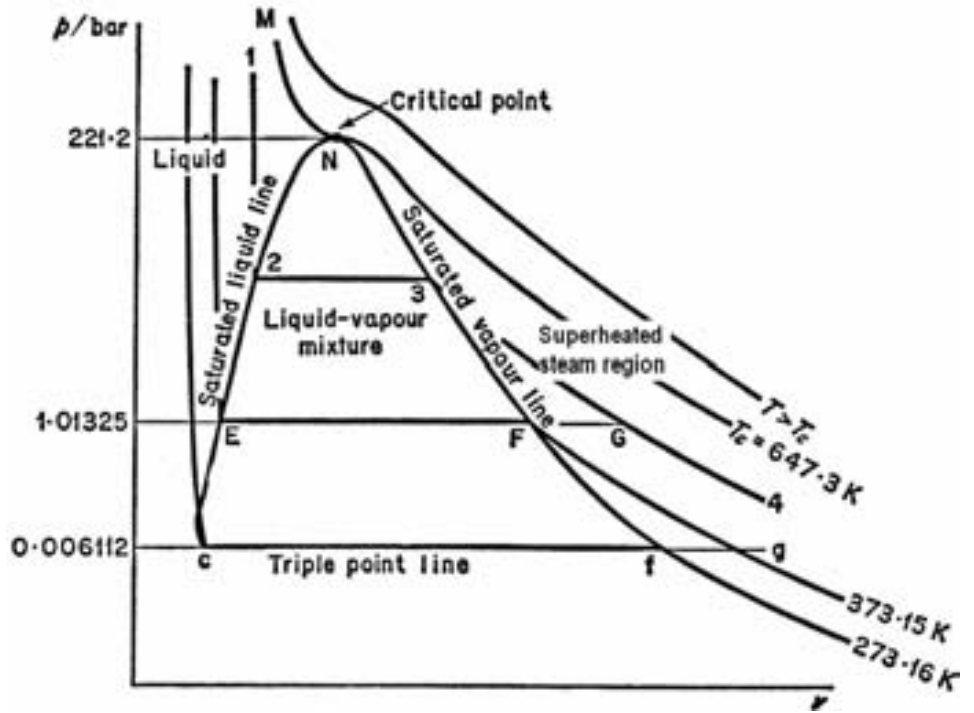
Figure (09): Temperature-entropy diagram for steam. This diagram is useful to represent different steam cycle processes. Constant pressure lines and constant volume lines are represented.
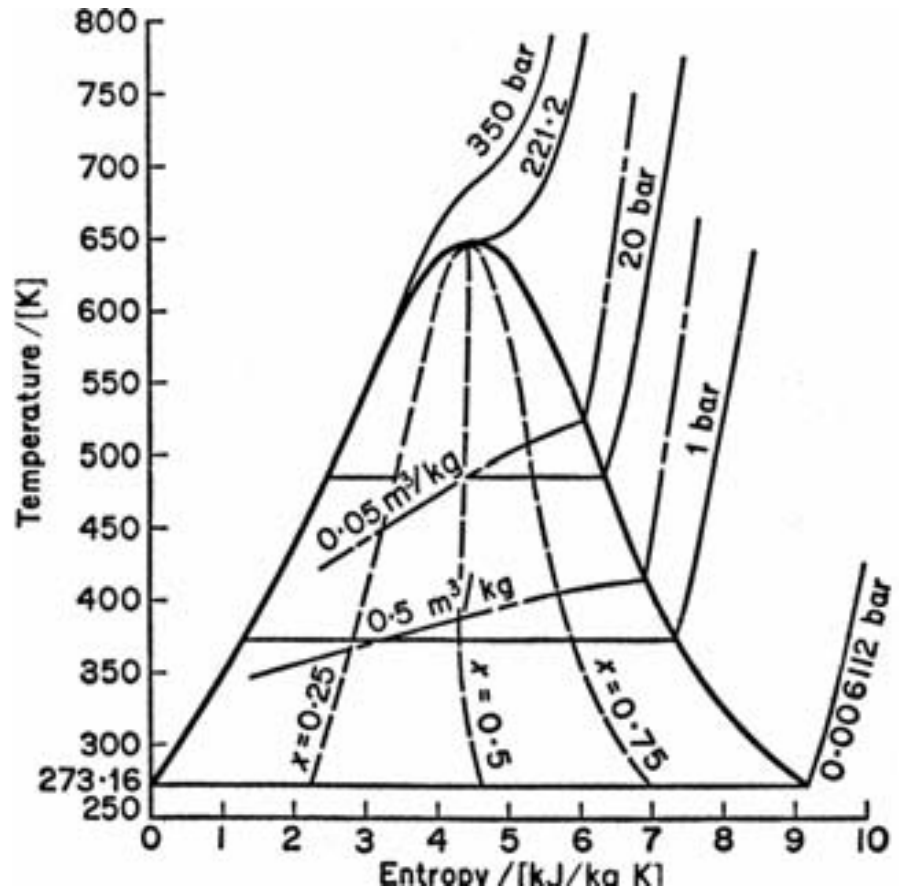
Determining Parameter of State of Steam and Basic Relations

<u>Steam table:-</u> the one used here is arranged by G.F.C. Rogers and Y.R. Mayhew

1) Unsaturated water region: - All the specific properties of saturated liquid are given in page (10), or simply can be approximated by saturated water properties.

2) Saturated liquid: - All the specific properties of saturated liquid are given as (v_f, u_f, h_f, s_f) at page (2) where saturated temperature (t_s) is the known value and pages (3, 4 and 5) where saturated pressure is the known value.

3) Saturated vapour: - All the specific properties of saturated vapour are given as (v_g, u_g, h_g, s_g) at page (2) where saturated pressure (p_s) is the known value and pages (3, 4 and 5) where saturated pressure is the known value.

4) Superheated steam region: - All the specific properties are given as (v_g, u_g, h_g, s_g) at pages (6, 7, 8 and 9) where two properties should be known. Usually properties are presented for different temperatures at each pressure. For values of pressure and temperatures that are not presented, one should interpolate the needed values from the existed values.

5) Wet steam region: - also called two phase (liquid-vapour) region. Properties of wet steam at this are have intermediate values between saturated liquid properties and saturated vapour properties. The dominant properties depend on the position of the point in this region to which line it is closer, i.e. depend on the ratio of dry steam amount to the total steam. This ratio is called dryness ratio or simply quality (x).

At this region, lines of constant pressures and constant temperatures are coincided. Therefore, they are not independent on each other. And for this reason they are regarded as a single property and anther property is needed to extract other properties.

Dryness ratio, $x = \dfrac{\text{dry steam mass}}{\text{total steam mass}}$

At point (1) mixture volume (V_1) is: -

$$V_1 = V_{\text{Liquid}} + V_{\text{Vapour}} \qquad \ldots (01)$$

$$V_1 = m_1 v_1 = m_f v_f + m_g v_g \qquad \ldots (02)$$

$$v_1 = \frac{m_f}{m_1} v_f + \frac{m_g}{m_1} v_g \qquad \ldots (03)$$

$$m_1 = m_f + m_g \qquad \ldots (04)$$

$$v_1 = \left(1 - \frac{m_g}{m_1}\right) v_f + \frac{m_g}{m_1} v_g \qquad \ldots (05)$$

$$v_1 = (1 - x_1) v_f + x_1 v_g \qquad \ldots (06)$$

$$v_1 = v_f + x_1 (v_g - v_f) \qquad \ldots (07)$$
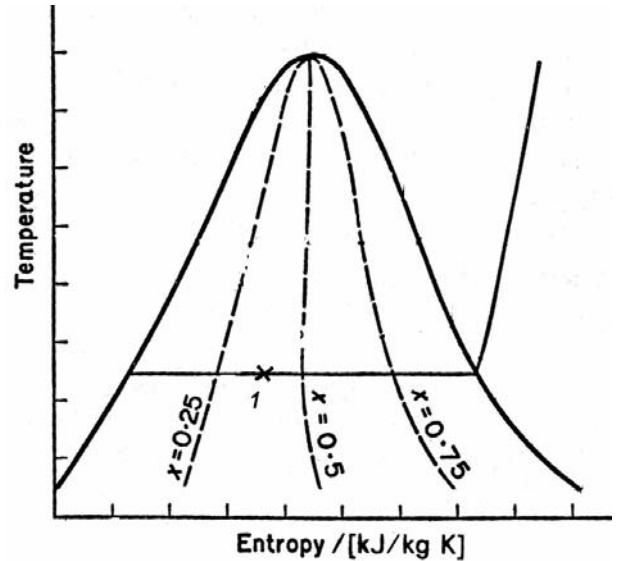
Figure (09): T-s diagram for steam. Evaluation of wet steam properties

For moderate pressures, the specific volume of liquid is much smaller than the specific volume of vapour,($v_g \gg v_f$), than:-

$$v_1 = x_1 v_g \qquad \ldots (08)$$

Similarly for other properties:-

$$u_1 = u_f + x_1 u_{fg} \qquad \ldots (09)$$

$$h_1 = h_f + x_1 h_{fg} \qquad \ldots (10)$$

$$s_1 = s_f + x_1 s_{fg} \qquad \ldots (11)$$

Extracting steam properties by interpolation:-

Steam table cannot contain infinite data, then the between values are evaluated by interpolation from previous and next values.

Steam basic relation: -

Steam properties can be evaluated from the following simple relations.

Unsaturated water:

For specific enthalpy
$$h_w = h_o + q_{senseble} ;$$
(h_o) is the enthalpy at ($t = 0.01 \ ^\circ C$, triple point) which is chosen to be zero. In addition, also ($u_o = 0, s_o = 0$). Where ($q_{sensable}$) is the sensible heat needed to raise water temperature. So,
$$h_w = q_{sens.} = \int_0^t c_{w,p} dt = [c_{w,p}]_0^t (t_w - 0) , (t \ in \ ^\circ C)$$
For specific internal energy:

$$u_w = h_w$$

$$[c_{w,p}]_0^t = 4.187 \ kJ/kg.K \quad as \ a \ mean \ value$$
For liquids the difference between (c_p) and (c_v) is very small. Moreover, the variation of is small for normal pressures and temperatures. For practical purpose, this variation is negligible and a mean value is used.

For entropy:

$$\Delta s = s_w - s_o$$
$$dq = Tds \Rightarrow ds = \frac{dq}{T} = \frac{1}{T} \int_0^T c_{w,p} dT = 4.168 \int_0^T \frac{dT}{T}$$
$$s_w = 4.187 \, Ln \frac{T_w}{273.3} , where \ (T) \ in \ K.$$

Saturated water:

$$h_f = h_w \ ; \ u_f = u_w \ ; \ s_f = s_w \quad at \ saturation \ temperature.$$

Wet steam:

$$h = h_f + x \, h_{fg}$$
$$h_{fg} = h_g - h_f$$

$$u = u_f + x\, u_{fg}$$

$$u_{fg} = u_g - u_f$$

$$u_g = h_g - (p_s v_g)$$

$$s = s_f + x\, s_{fg}$$

$$s_{fg} = \frac{h_{fg}}{T_s}$$

(h_{fg}) is the latent heat for evaporation. At wet steam region, the specific heat at constant pressure is variable due to gradual phase change. It sound that (h_g) should be evaluated experimentally.

Saturated steam:

$$h = h_f + h_{fg}$$

$$u = u_f + u_{fg}$$

$$s = s_f + s_{fg}$$

Super heated steam:

$$h = h_f + h_{fg} + h_{sup}$$

$$h_{sup} = \int_{t_s}^{t_{sup}} c_{sup,p}\, dt = [c_{sup,p}]_{t_s}^{t_{sup}} \times t_{sup}$$

$$u = u_f + u_{fg} + u_{sup}$$

$$u_{sup} = h_{sup} - p_s v_{sup}$$

$$s = s_f + s_{fg} + s_{sup}$$

$$s_{sup} = \frac{h_{sup}}{T_{sup}}$$

(h-s) diagram, also called mollier chart:-

This chart is useful for solving problems in thermodynamics. The chart has constant pressure lines, constant temperature lines and constant quality lines.

It is useful to analyze isentropic, steady state processes such as isentropic compressions or expansions processes that occurred in nozzles, diffusers, turbines and compressors. Where these processes are represented by vertical line.

It is also useful to represent throttling processes (constant enthalpy processes) which are occurred for example in throttling valves, expansion valves. These processes are represented by horizontal line.
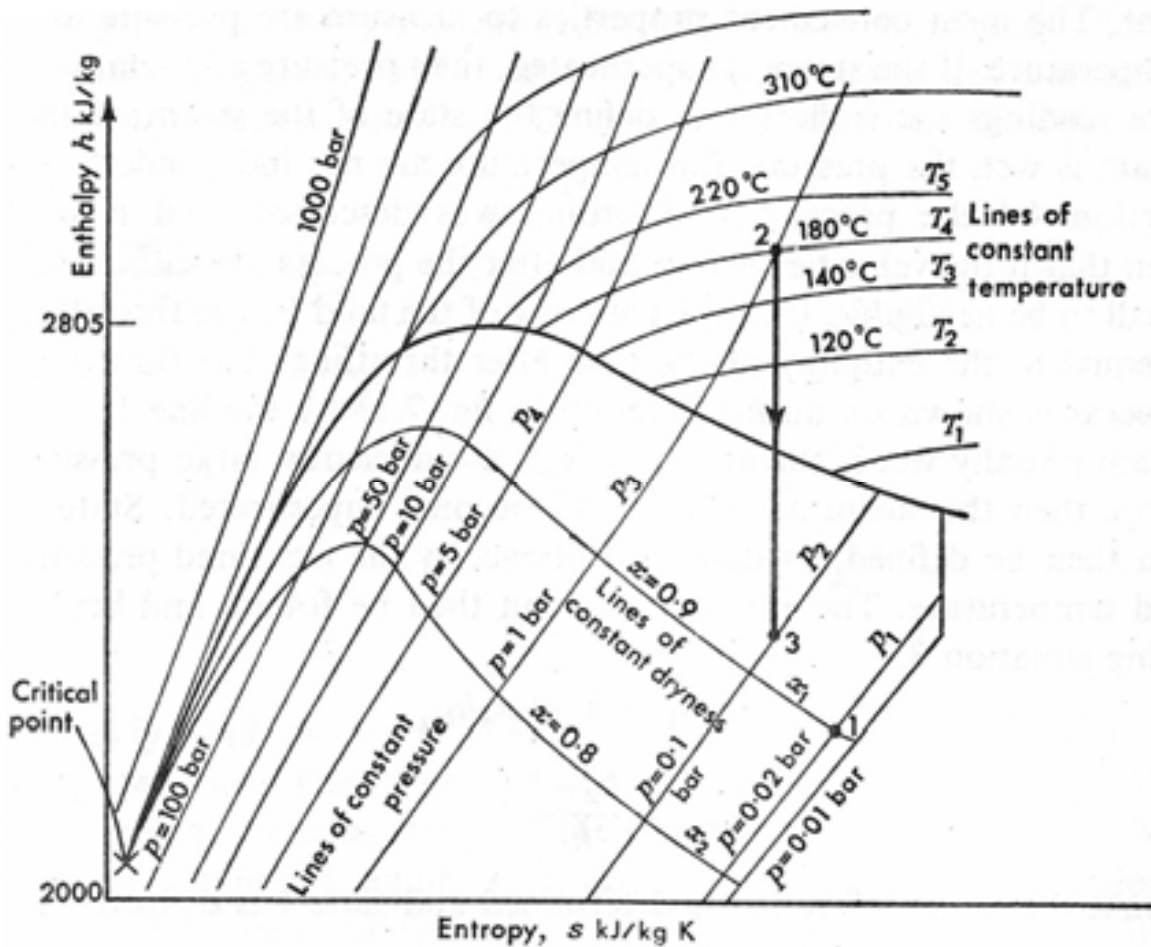
Figure (10): h-s (mollier) diagram for steam.

Reversible non-flow steam processes:-

1) constant volume processes: -

$$q_{12} - w_{12} = \Delta u$$
$$dq = du + pdv$$
$$q_{12} = u_2 - u_1$$
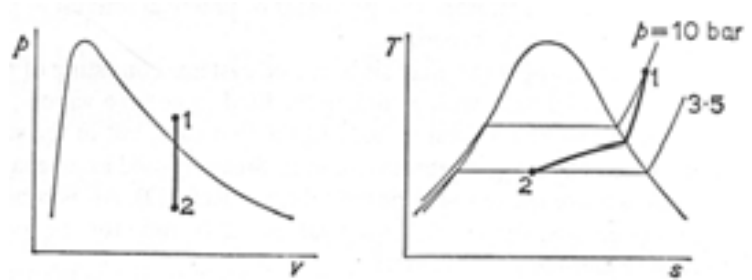
Figure (11): constant volume process

2) constant pressure processes: -

$$q_{12} - w_{12} = u_2 - u_1$$
$$w_{12} = p\Delta v = pv_2 - pv_1$$

$$\therefore q_{12} = h_2 - h_1$$
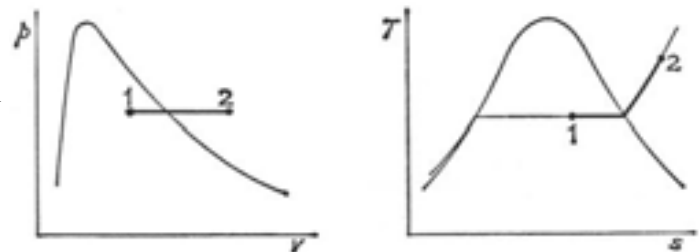
Figure (12): constant volume process

3) constant temperature (isothermal) processes: -

$$dq = Tds$$
$$q_{12} = T(s_2 - s_1)$$
$$w_{12} = q_{12} - \Delta u$$
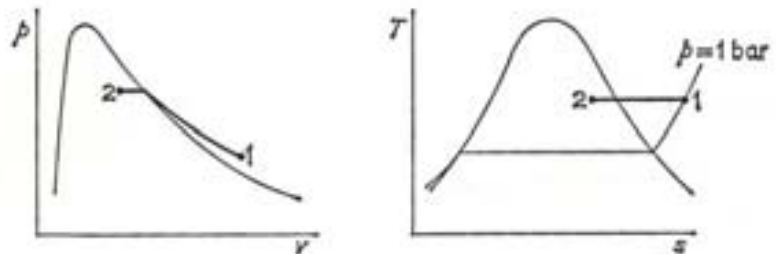
Figure (13): isothermal process

4) isentropic (adiabatic ,reversible) processes: -

$$dq = 0$$
$$ds = 0 \Rightarrow s_1 = s_2$$
$$w_{12} = \Delta u = u_2 - u_1$$

also

$$dw = pdv$$

$$p_1 v_1^\gamma = p_2 v_2^\gamma$$

Figure (14): isotropic process

1
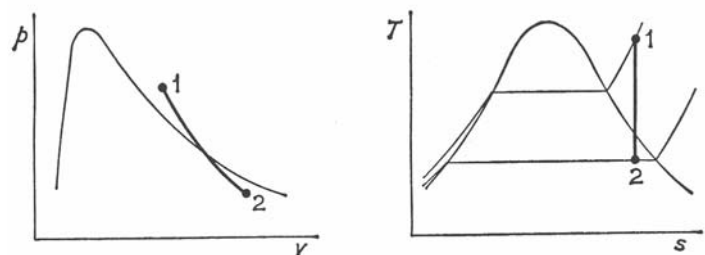
$$w_{12} = \frac{p_1 v_1 - p_2 v_2}{\gamma - 1}$$

At wet region ($\gamma \approx 1.125$) and at superheated region ($\gamma \approx 1.3$).

5) Polytrophic processes: -

$$p_1 v_1^n = p_2 v_2^n$$

$$w_{12} = \frac{p_1 v_1 - p_2 v_2}{n - 1}$$

$$q_{12} = (u_2 - u_1) + \frac{p_1 v_1 - p_2 v_2}{n - 1}$$
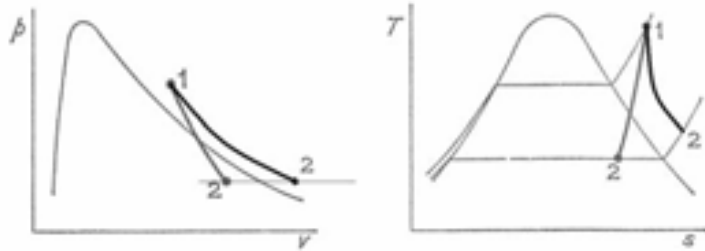
Figure (15): polytropic process

Throttling process:-

Flow process within several devices in thermodynamics causes no heat and work exchanging, and change of kinetic and potential energies is considered negligible, see fig (16). This process is called throttling process.

(a) An adjustable valve

(b) A porous plug

(c) A capillary tube

Throttling orifice
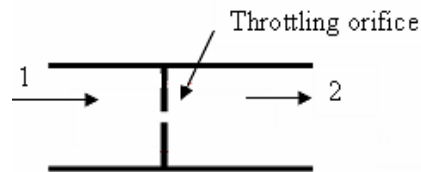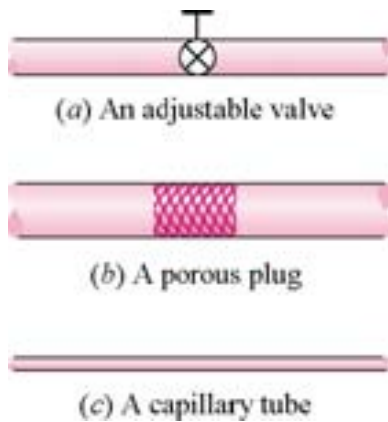
Figure (16): throttling devices

2

Throttling process is highly irreversible due friction loses. From steady flow energy equation (SFEE)

$$q - w = \Delta h + \Delta p.e. + \Delta k.e.$$
$$q = 0 \; ; \; w = 0$$
$$\Delta p.e. = g(z_2 - z_1) = 0$$
$$\Delta k.e. = 0.5(C_2^2 - C_1^2) = 0$$
$$\therefore \Delta h = 0$$

i.e. $h_2 = h_1$

Throttling process is a constant enthalpy process, which is an adiabatic irreversible process.

Throttling process is represented in p-v, T-s and h-s diagrams as shown in figures (18), (19) and (20).
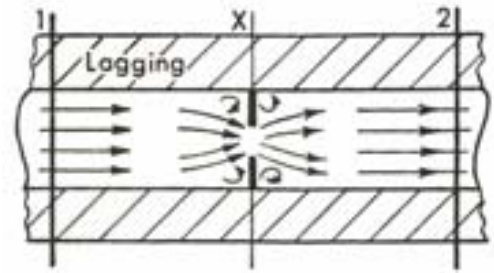
Figure 17: flow in throttling valve
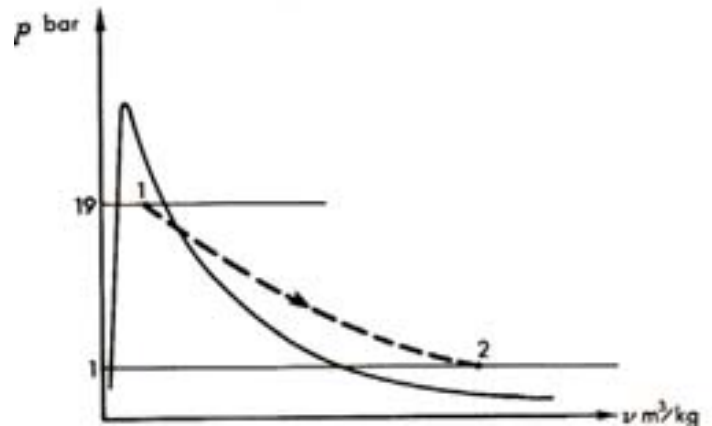
Figure (18): p-v diagram
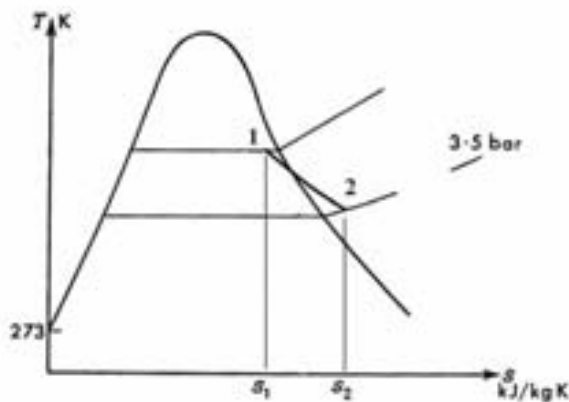
Figure (19): T-s diagram
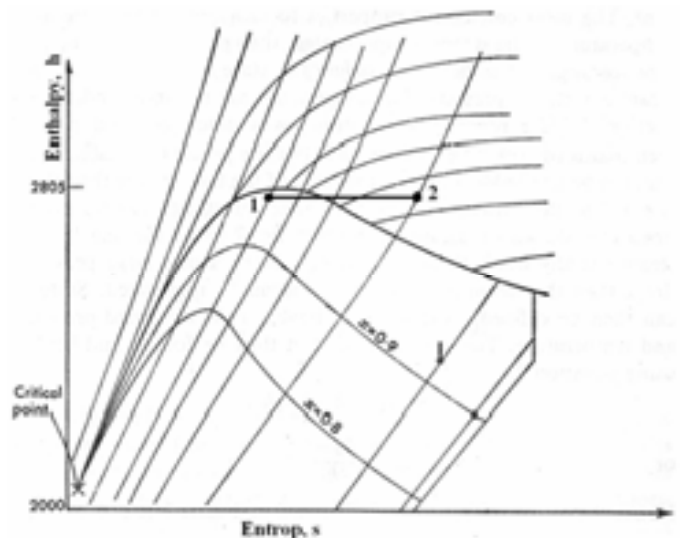
Figure (20): h-s diagram

3

Measurement of dryness fraction of wet steam

Throttling calorimeter:
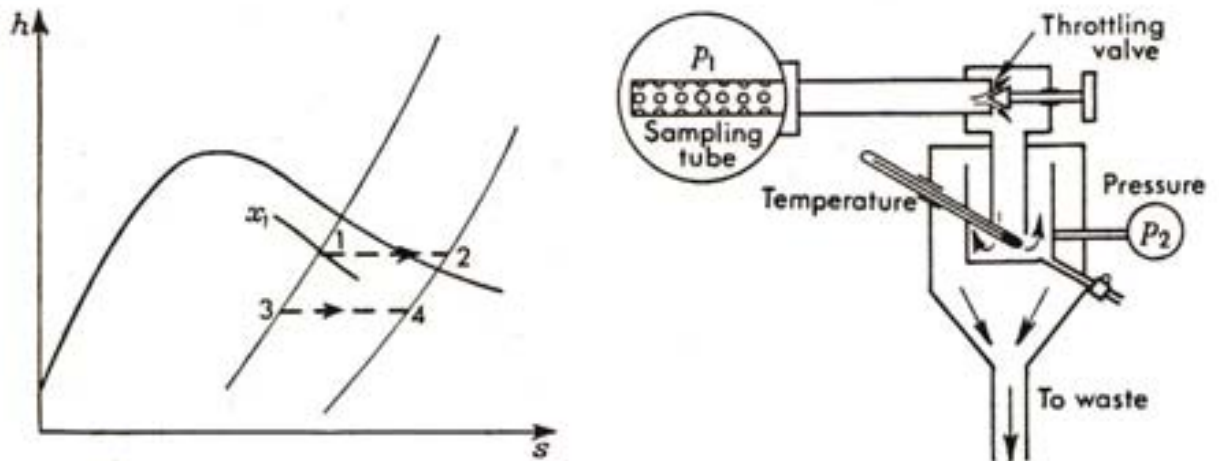
Figure 21: throttling calorimeter, where throttling process is drawn on h-s diagram

A sample of wet steam at (p_1) is taken from the main through sampling tube. It is throttled by throttling valve to (p_2) and temperature (t_2), so that the steam leavening the throttling valve is superheated steam.

$$h_1 = h_2$$
$$h_1 = h_{f1} + x_1 h_{fg1}$$
$$x_1 = \frac{h_2 - h_{f1}}{h_{fg1}}, \text{ which is the dryness fraction of wert steam in the main.}$$

This method can be used only if $(x_1 \geq 0.95)$, otherwise, the throttling process is represented by process (3-4), which is not useful.

Separating and throttling calorimeter:-

In this calorimeter, the moisture is first extracted from the steam sample in separating calorimeter to increase dryness fraction above (0.95). Pressure and temperature are remaining constant. The sample is then passes through throttling calorimeter where it expands to superheated steam region to (p_3) and (t_3).the value of (h_3) is read then from superheated steam table.
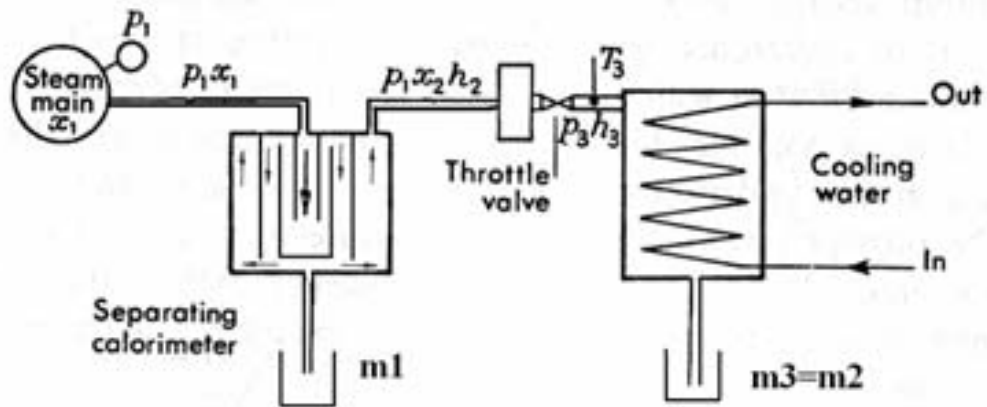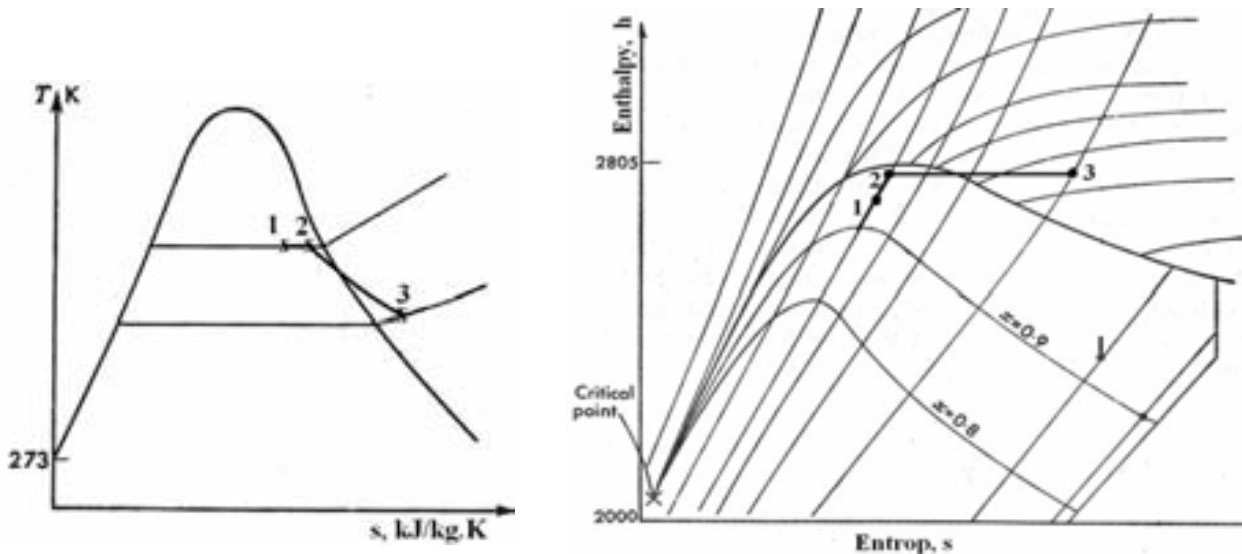

Figure 22: separation-throttling calorimeter.

Figure 23: representation of two processes of separation-throttling calorimeter on T-s and h-s diagrams.

5

$p_1 = p_2$; And steam at (2) is still wet after separation but with less moisture.

$$h_2 = h_3 = h_{f1} + x_2 h_{fg1}$$

$$x_2 = \frac{h_3 - h_{f1}}{h_{fg1}}$$

$$m_{total} = m_1 + m_2$$

$$m_2 = m_3$$

$$x_1 = \frac{\text{mass of dry steam}}{\text{total mass}}$$

$$x_2 = \frac{\text{mass of dry steam}}{m_2}$$

$$\therefore \text{mass of dry steam} = x_2 m_2$$

$$x_1 = \frac{x_2 m_2}{m_1 + m_2}$$

Also from heat balance where the calorimeter is treated as a system, then:

$$m_{total} * h_1 = m_1 h_{f1} + m_2 h_2$$

$$h_1 = \frac{m_1 h_{f1} + m_2 h_2}{m_1 + m_2}$$

$$x_2 = \frac{h_1 - h_{f1}}{h_{fg1}}$$

<u>Non-steady flow energy equation:-</u>

$\dot{m}_1, u_1$
c_1, p_1 →
v_1, z_1

$\dot{W}_{in}$

m', u'
m'', u''

$\dot{Q}_{in}$

$\dot{m}_2, u_2$
c_2, p_2
v_2, z_2
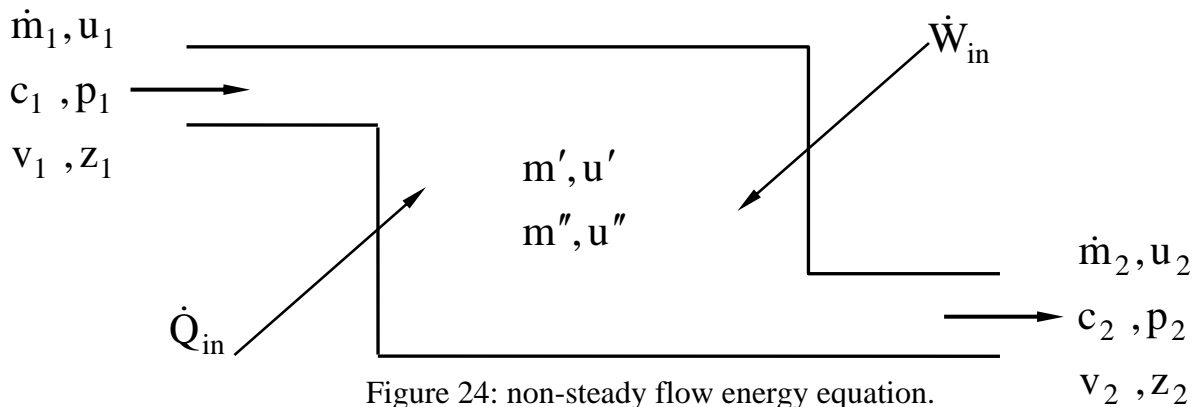
Figure 24: non-steady flow energy equation.

For the open system shown, the flow is non-steady. This means that the entered mass of worked fluid and the entered energies to the open system do not equal to exit mass and energies with time. The difference causes the mass and the energy of the system to change.

$$\text{Energy Entering the System} = \dot{Q}_{in} + \dot{m}_1\left(u_1 + \frac{c_1}{2} + gz_1 + p_1 v_1\right)$$

$$\text{Energy Leaving the System} = \dot{W}_{in} + \dot{m}_2\left(u_2 + \frac{c_2}{2} + gz_2 + p_2 v_2\right)$$

$$\text{Increase of the Energy of System, } E = E'' - E' = m''u'' - m'u'$$

$$\text{Energy Entering the System} - \text{Energy Leaving the System} =$$
$$\text{Increase of the Energy of System}$$

$$\left[\dot{Q}_{in} + \dot{m}_1\left(u_1 + \frac{c_1}{2} + gz_1 + p_1 v_1\right)\right] - \left[\dot{W}_{in} + \dot{m}_2\left(u_2 + \frac{c_2}{2} + gz_2 + p_2 v_2\right)\right] =$$
$$\left[\dot{m}''u'' - \dot{m}'u'\right]$$

This equation is satisfying the conservation of energy. For conservation of mass:-

$$\dot{m}_1 - \dot{m}_2 = \dot{m}'' - \dot{m}'$$

Application of steady flow energy equation:-

The steady flow energy equation has the following form.

$$\left[\dot{Q}_{in} + \dot{m}_1\left(u_1 + \frac{c_1}{2} + gz_1 + p_1v_1\right)\right] - \left[\dot{W}_{in} + \dot{m}_2\left(u_2 + \frac{c_2}{2} + gz_2 + p_2v_2\right)\right] = 0$$
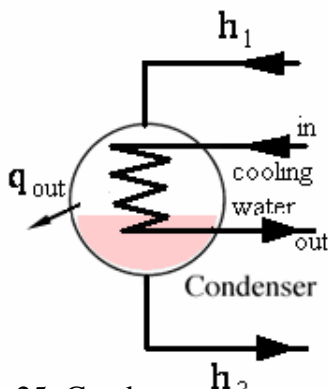
a) Boilers and condensers:-

Figure 25: Condenser.

Figure 26: Boiler.

For condensers; work exchange is zero, ΔPE and ΔKE is also zero, then:-

$$q_{out} = h_1 - h_2$$
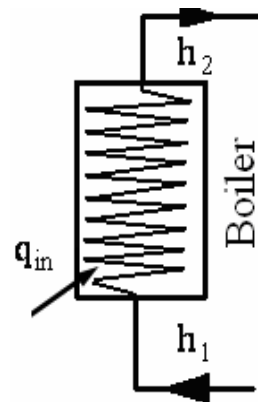
For boilers; work exchange is zero, ΔPE and ΔKE is also zero, then:-

$$q_{in} = h_2 - h_1$$

b) Turbines and compressors:-

Figure 27: Turbine.
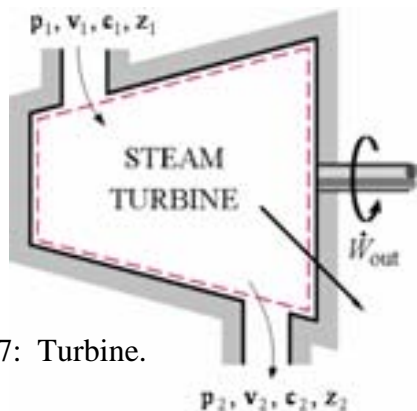
Figure 28: Compressor.

Turbines; for isentropic expansion heat transfer is zero. Change of potential and kinetic energies is zero also, then;

$$w_{out} = h_1 - h_2$$
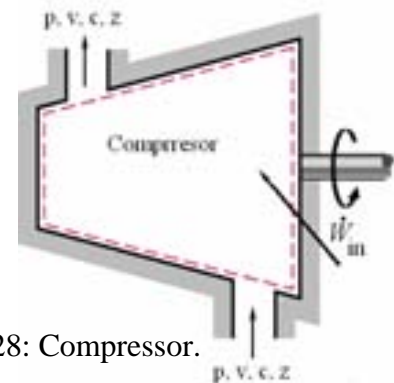
Compressors; for isentropic expansion heat transfer is zero. Change of potential and kinetic energies is zero also, then;

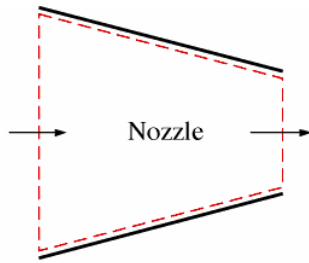$$w_{in} = h_2 - h_1$$
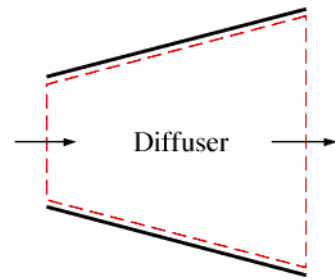
8

c) Nozzles and diffusers:-

Figure 29: Nozzle.

Figure 30: Diffuser.

In nozzles, the expansion is assumed isentropic to obtain maximum velocity. There is no work and heat transfer and change of potential energy is negligible, then:-

$$\frac{1}{2}\left(c_2^2 - c_1^2\right) = h_1 - h_2$$

And nozzle efficiency is:

$$\eta_N = \frac{c_2^2}{c'^2_2}$$

c_2 is the final velocity due to actual process.

c'_2 is the final velocity due to isentropic process

In diffusers, the expansion is assumed isentropic to obtain maximum pressure. There is no work and heat transfer and change of potential energy is negligible, then:-

$$\frac{1}{2}\left(c_1^2 - c_2^2\right) = h_2 - h_1$$

And diffuser efficiency is:

$$\eta_D = \frac{p_2 - p_1}{p_2 - p_1}$$

$p_2 - p_1$ is pressure rise due to acual process.

$p'_2 - p_1$ is pressure rise due to isentropic process.

Isentropic efficiency:-

The isentropic efficiency is very important tool to relate isentropic expansion to actual expansion and isentropic compression to actual compression.

$$\eta_{ise,comp.} = \frac{h'_2 - h_1}{h_2 - h_1} \text{ to find } (h_2).$$

$$\eta_{ise,exp.} = \frac{h_3 - h_4}{h_3 - h'_4} \text{ to find } (h_4).$$
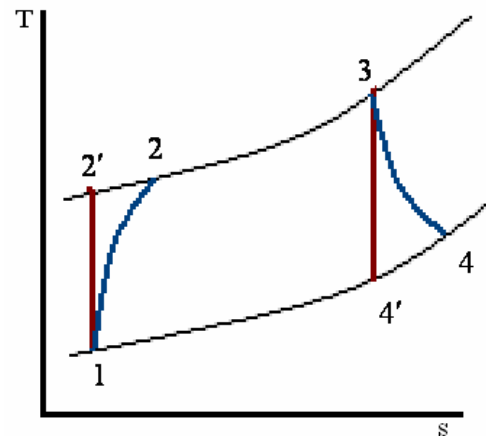
<u>Steam cycles:-</u>

<u>Carnot cycle:-</u>

      This cycle is hypothetical cycle, which is used in steam field for explanation and comparison purpose only. The cycle is composed from the following processes:-

(1-2): is an isothermal processes where heat is added through a boiler at maximum temperature.
(2-3):is isentropic process where work is extracted through turbine.

(3-4): is an isothermal process where heat is rejected through a condenser at minimum temperature.
(2-3): is isentropic process where work is added through water pump.
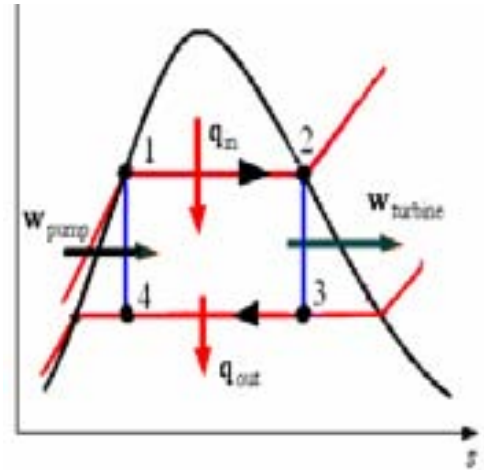
Figure 31: Carnot cycle.

      Although Carnot cycle has the maximum possible efficiency, but its work ratio is low. Furthermore the pump needed to pump water-steam mixture from (4) to (1) is complex, expensive and consumes high work., so it is impractical for steam power plants.

$$\eta_{carnot} = \frac{workout\ put}{heat\ added}$$

$$= \frac{W_{net}}{q_{in}}$$

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

$$work\ ratio = \frac{W_{net}}{W_{out}}$$

<u>Rankin cycle:-</u>

It is the ideal steam cycle, where steam is condensed completely into water at (3). Therefore, pump needed to pump saturated water from (3) to (4) is simple ordinary one and the woke needed is also small, which means that it has a high work ratio.

(1-2): isentropic expansion through the turbine.

(2-3) Heat is rejected at constant pressure and steam is condensed. Point (2) may lies on dry steam line or to the lift or to the right from this line along ($p_c = c$) line.

(3-4) isentropic compression, where water pump exerts work to lift water from condenser pressure to boiler pressure.

(4-1) heat addition at constant pressure.

Figure 32: simple ideal Rankine cycle.

Work out put:
$$w_{out} = h_1 - h_2$$

Heat added:
$$q_{in} = h_1 - h_4$$

Pump work:
$$w_{pump} = h_4 - h_3$$

Heat rejected:
$$q_{out} = h_2 - h_3$$
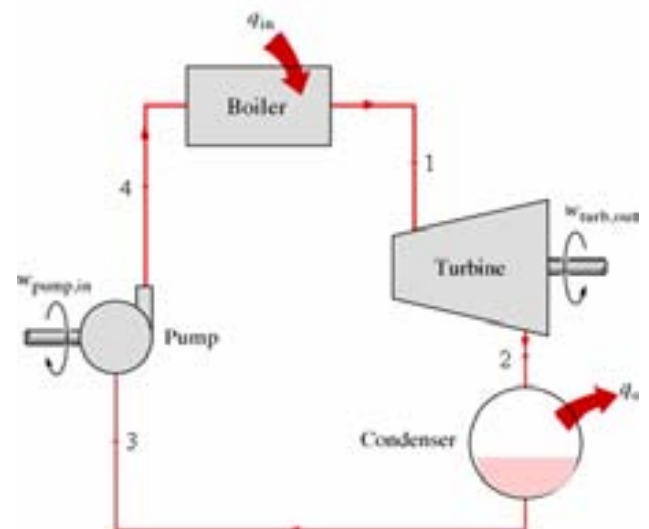
Figure 33: Steam power plant based on simple and ideal Rankine cycle.

Net work out put:

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

Rankine thermal efficiency:

$$\eta_R = \frac{Net\ work\ output}{heat added} = \frac{W_{out} - W_{in}}{q_{in}}$$
$$= \frac{(h_1 - h_2) - (h_3 - h_4)}{h_1 - h_3}$$

In many cases, $W_{pump} << W_{turbine}$, then:

$$\eta_R = \frac{h_1 - h_2}{h_1 - h_3}$$

Pump work is also being expressed in terms of boiler and condenser pressures. Taking into consideration that there is no heat transfer then:

$$dq = du + pdv = 0$$
$$dh = du + pdv + vdp$$

Liquids incompressible fluids at moderated pressures i.e. have a constant volume, then:

$$\int dh = \int vdp \Rightarrow \Delta h = v\Delta p$$
$$W_{pump} = v_{water}(p_b - p_c)$$

Work ratio:

It is another comparison criterion of performance in steam power plants. It is define as the ratio of net output work to the total produced work. The higher the value the best the plant for the same output power.

$$work\ ratio = \frac{W_{net}}{W_{out}}$$

3

Specific steam consumption:

It is the amount of steam per unit time to produce one unit power. (s.s.c.) is also a comparison criterion in steam power plants. The lower the value the best the plant for the same output power. It gives a direct indication to the relative size of the plant components.

$$\text{s.s.c.} = \frac{\dot{m}_s}{P_{out}} \frac{kg/h}{kW}$$

($\dot{m}_s$) is steam mass flow rate in kilogram per hour required. However, steam mass flow rate is usually measured in kilogram per second so it should be multiply by (3600).
(P_{out}) is output power in kilowatt.

$$\text{s.s.c.} = \frac{\dot{m}_s * 3600}{\dot{m}_s w_{net}}$$

$$= \frac{3600}{w_{net}} \quad kg / kW.h$$

Influence of Steam Condition on Power Plant Performance:-

The effect of the inlet and outlet steam conditions in turbine has a great effect on cycle thermal efficiency, work ratio and s.s.c. by comparison with Carnot steam cycle that the efficiency is increased as the mean temperature at which heat is added is increased or as the temperature at which heat is rejected is decreased. In addition, efficiency is improved by increasing boiler pressure or decreasing condenser pressure.
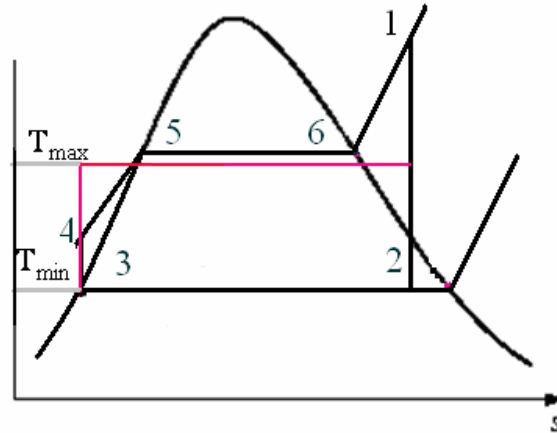
Increasing boiler temperature or pressure is limited, either by thermal stresses or hoop stresses due to metallurgical limitation.

Decreasing condenser temperature or pressure is limited by cooling water temperature (river water temperature) or pumping functionality of building vacuum pressure at condenser side.

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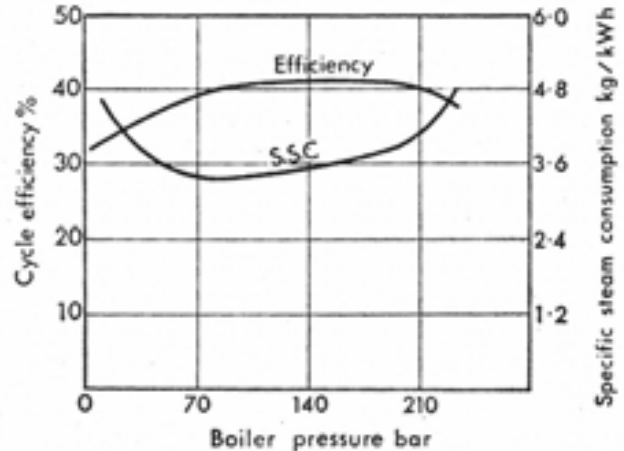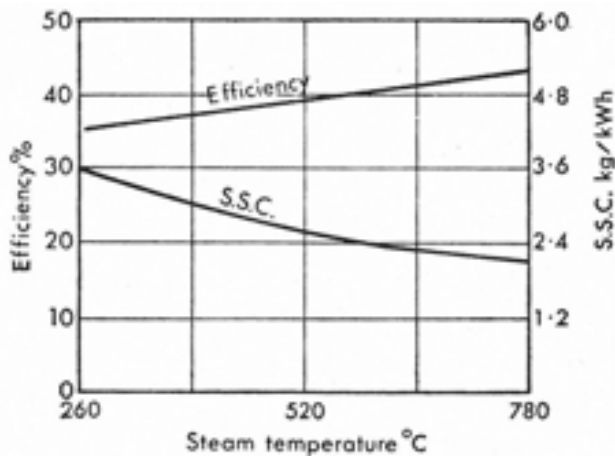
Rankine cycle with superheating:

In order to increase cycle thermal efficiency, the mean temperature at which heat is added should be increased. One method to do so is by increasing heating temperature so that the produced steam becomes super-heated.

This method has the advantage of moving point (2) closer to dry steam line where. This means that either the steam leaving the turbine is of higher dryness fraction or dry or even a bit little superheated, i.e. very low moisture contain. This moisture contain at the final stages of turbine could harm the turbine blade.
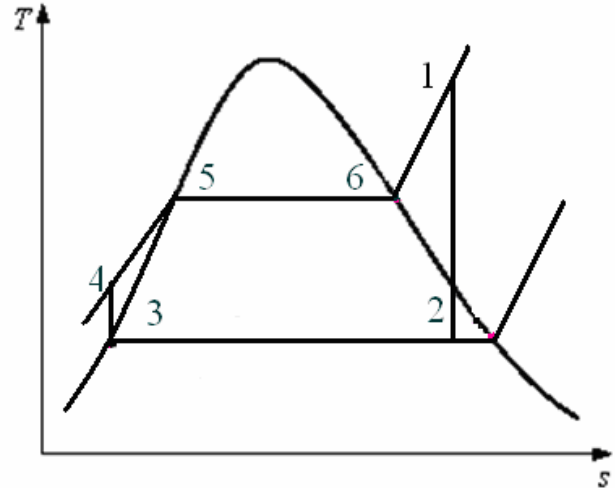
$$\eta_R = \frac{h_1 - h_2}{h_1 - h_4}$$

$$\text{work ratio} = \frac{W_{net}}{W_{out}} = \frac{(h_1 - h_2) - (h_4 - h_3)}{h_1 - h_4}$$

$$\text{s.s.c.} = \frac{3600}{W_{net}} \quad {}^{kg}\!\!\big/\!\!_{kW.h}$$

6

Rankine cycle with reheater:

There is a limit for increasing boiler temperature. Moreover, one long expansion process may result in a wet steam of low quality. Besides one heavy, high number of stages is not preferable. Instead of that two-turbines (may be three) are used. The first one is high-pressure turbine (HPT) and the second one is low-pressure turbine (LPT). The expansion process is subdivided into two shorter processes. The steam leaving the (HPT) is heated again to energize it to higher enthalpy. The heating system is called reheating. The reheating is accomplished either in separate heating system near the turbines or at the main furnace of the boiler. The steam condition at (LPT) exit is actually very close to saturated steam line.
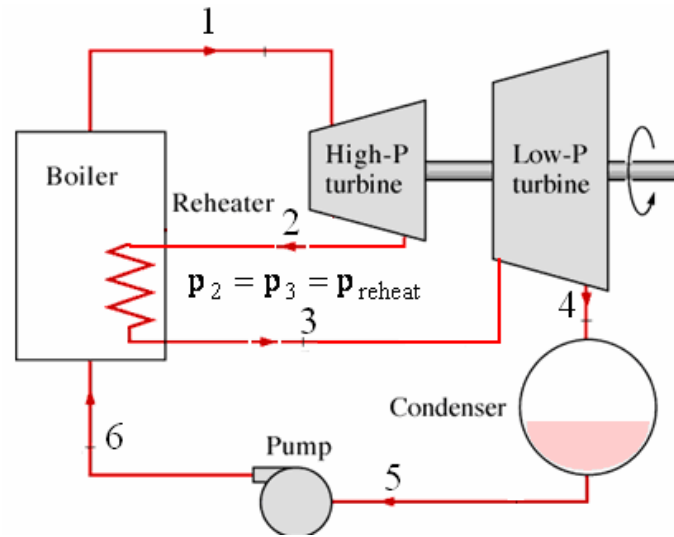
For perfect reheat: $p_3 = p_{boiler}/4$; $T_3 = T_{boiler}$

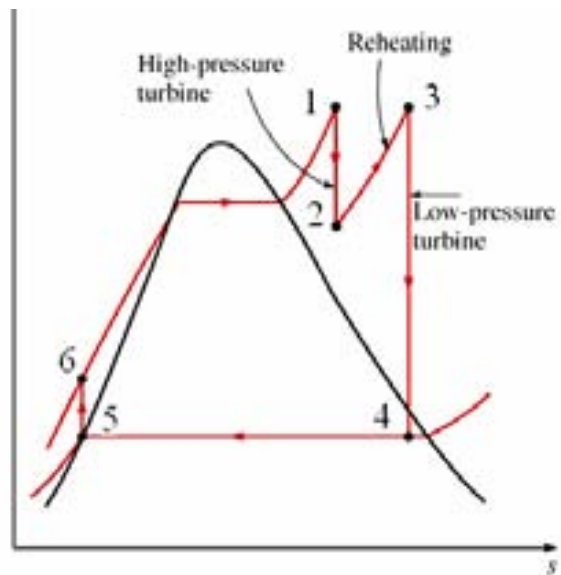$$w_{out} = (h_1 - h_2) + (h_3 - h_4)$$
$$w_{in} = (h_6 - h_5)$$
$$q_{in} = (h_1 - h_6) + (h_3 - h_4)$$
$$q_{out} = (h_4 - h_5)$$
$$\eta_R = \frac{w_{net}}{q_{in}} = \frac{w_{out} - w_{in}}{q_{in}}$$
$$\text{work ratio} = \frac{w_{net}}{w_{out}}$$
$$\text{s.s.c.} = \frac{3600}{w_{net}} \quad kg/kW.h$$

Regenerative Rankine Cycle:

Increasing cycle efficiency further is possible by increasing the temperature of the feed water that enters the boiler, which led to the increase of mean temperature. It is impractical to take feed water to pass through the turbine, counter clockwise, in order to gain higher temperature by means of a heat exchanger. However, it is practical to take some hot steam from the turbine to heat the feed water. Increasing feed water temperature before boiler called regeneration, i.e. increase its enthalpy.

The process of taking some hot steam from the turbine called turbine bleeding. That is why some pointed mined called this cycle as "bleeding cycle".

The Area shaded ($7' - 7 - 2 - 3 - 4 - 4'$) which is heat extracted from turbine, is equal to the area shaded ($8' - 8 - 1 - 6 - 5 - 5'$) which is the heat gained by feed water to raise its temperature to saturated temperature at boiler pressure.

For optimum cycle performance, the value of pressure along the turbine at which the stem is bled is important. Almost that the bleeding pressure value is at a temperature ($t_{s,bleed}$), which is the mean temperature between ($t_{s,boiler}$) at boiler pressure and ($t_{s,condenser}$) at condenser pressure. The interval (Δt_s) temperature is evaluated as:

$$\Delta t_s = \frac{t_{s,B} - t_{s,C}}{n + 1}; \text{ Where n is the number of}$$

heaters. For one feed-water heaters:
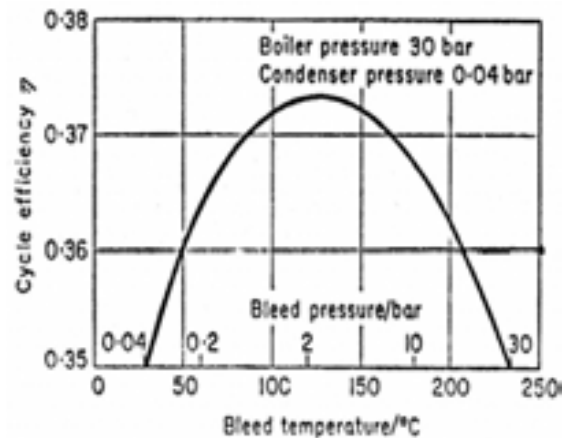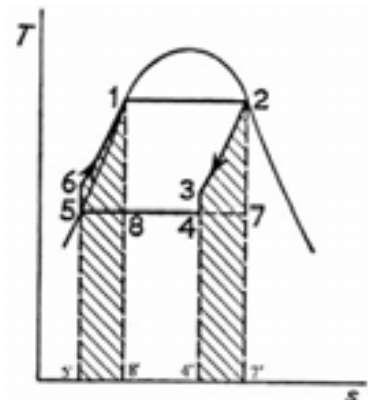
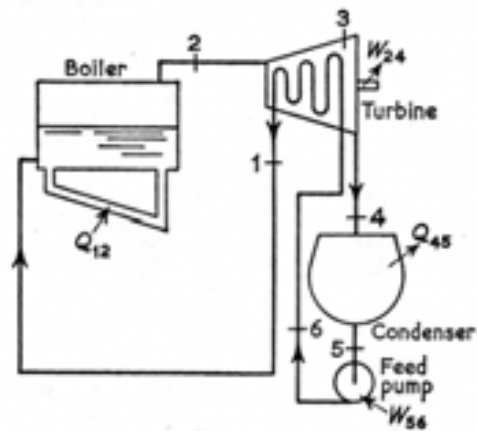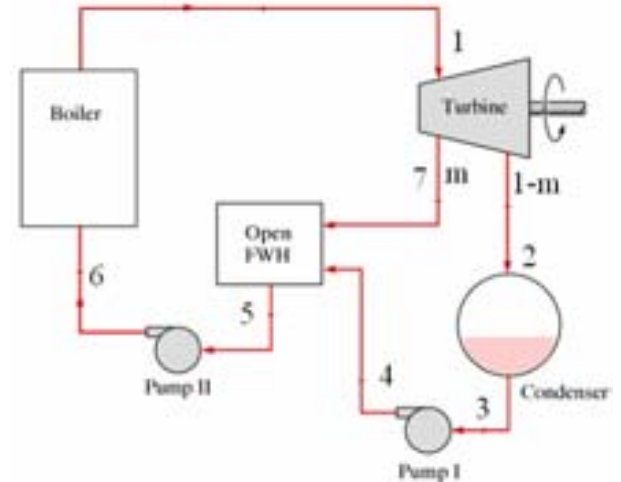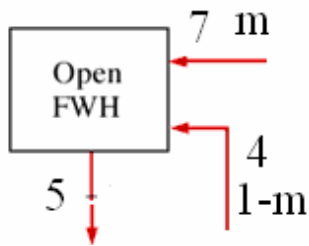$$t_{s,bleed} \frac{t_{s,B} + t_{s,C}}{2}$$

Fig. 11.18 Effect of bleed temperature on the cycle efficiency

Regenerative Rankine Cycle with Open Feed Water Heater:

Hot steam at (p_7) from turbine of mass fraction (m) is mixed with condensate water at (p_4) of mass fraction (1-m) in an open feed water heater chamber.

From heat balance

$$h_5 = mh_7 + (1-m)h_4$$

$$m = \frac{h_5 - h_4}{h_7 - h_4}$$

$$w_{out} = (h_1 - h_7) + (1-m)(h_7 - h_2)$$
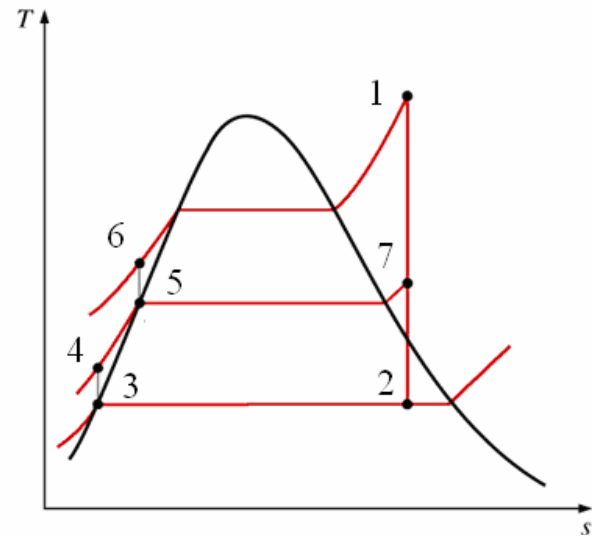
$$w_{in} = (h_4 - h_3) + (h_6 - h_5)$$

$$q_{in} = (h_1 - h_6)$$

$$q_{out} = (h_2 - h_{3)}$$

$$\eta_R = \frac{w_{net}}{q_{in}} = \frac{w_{out} - w_{in}}{q_{in}}$$

$$\text{work ratio} = \frac{w_{net}}{w_{out}}$$

$$\text{s.s.c.} = \frac{3600}{w_{net}} \quad kg\!\!\Big/\!\!kW.h$$

Number of (OFWH) is up to five or more in order to increase cycle efficiency, maintaining work ratio but with some penalty in (s.s.c.). The disadvantage of such method is that each OFWH needs feed pump and for further number of heaters the rise of cycle efficiency is dispensed with.

Regenerative Rankine Cycle with Closed Feed Water Heater:

For this cycle, no extra fee water pumps are needed. Moreover bled steam pressure may not equal to incoming water since the open type is a heat exchanger system and no mixing is occurred. There are two type of (CFWH). They are :

1. Cascaded backward closed feed water heater.
2. Cascaded forward ward closed feed water heater.

Cascaded back ward closed feed water heater:

This is the usual type used. Liquid at (p_7) of high pressure is throttling to lower pressure, for example (p_8).

$p_6 = p_7 = p_{bleed}$

$p_2 = p_3 = p_8$

$t_5 = t_7$

$h_7 = h_8 \approx h_5$

$h_4 = h_3 + w_{34}$
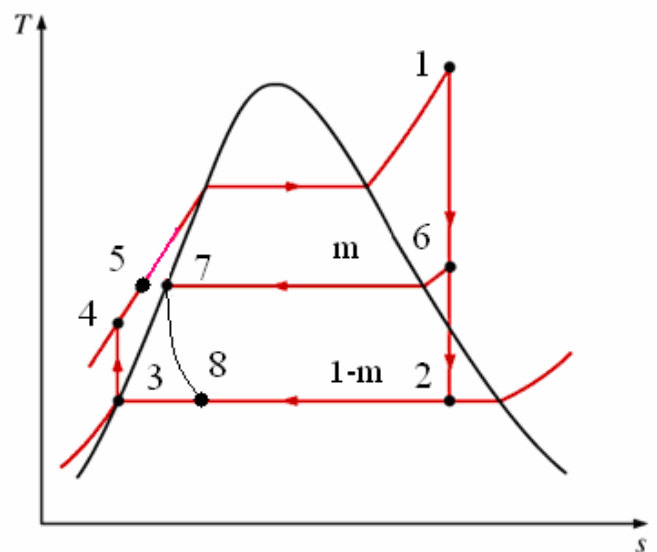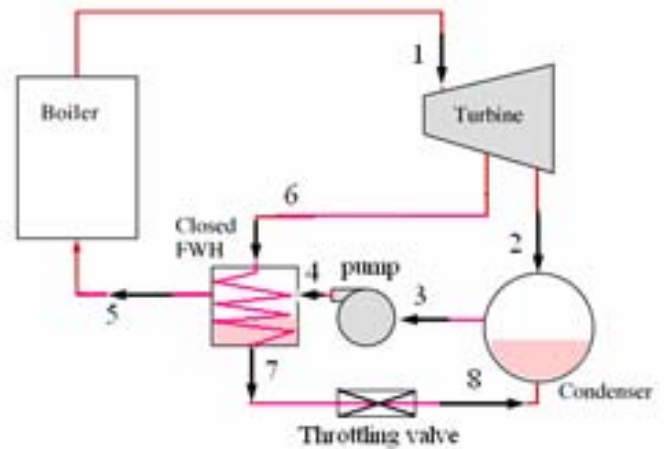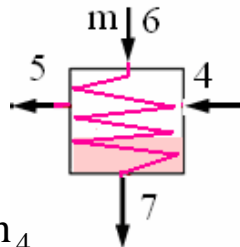
From heat balance:

$h_5 + mh_7 = mh_6 + h_4$

$m = \dfrac{h_5 - h_4}{h_6 - h_7}$

$w_{out} = (h_1 - h_6) + (1-m)(h_6 - h_2)$

$w_{in} = (h_4 - h_3)$

$q_{in} = (h_1 - h_5)$

$q_{out} = (1-m)(h_2 - h_8) + (h_8 - h_3)$

Cascaded back ward closed feed water heater:

This type demands a mixing chamber and one extra feedwater pump.
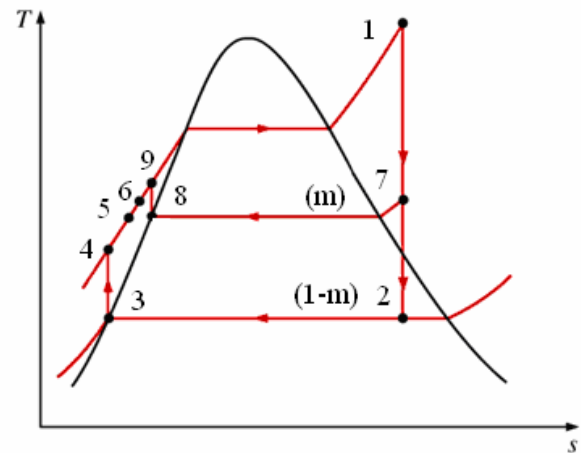
$p_7 = p_8 = p_{bleed}$
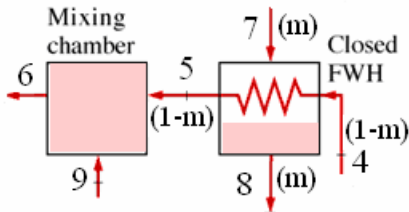
h_7 from $(h - s)$ chart.

$h_4 = h_3 + w_{34}$

$t_5 = t_8$

$h_8 \approx h_5$

$h_9 = h_8 + w_{89}$

From heat balance:

Closed feed-water heater:

$$(1 - m)h_5 + mh_8 = mh_7 + (1 - m)h_4$$

$$m = \frac{h_5 - h_4}{h_7 + h_5 - h_8 - h_4}$$
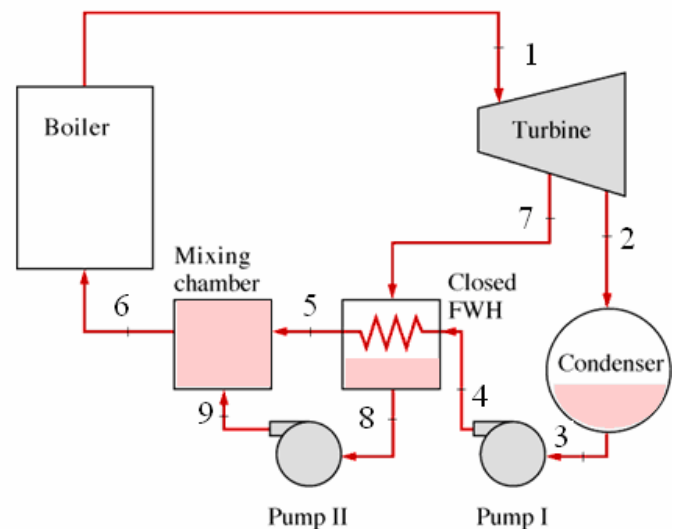
Mixing camber: to evaluate (h_6):

$$h_6 = mh_9 + (1 - m)h_5$$

$$w_{out} = (h_1 - h_6) + (1 - m)(h_6 - h_2)$$

$$w_{in} = (h_4 - h_3)$$

$$q_{in} = (h_1 - h_5)$$

$$q_{out} = (1 - m)(h_2 - h_8) + (h_8 - h_3)$$

11

Example:

The thermal power plant shown operates between $(10 \text{ MPa}, 450 \, ^\circ\text{C})$ and (0.035 bar). Draw (T-s) diagram and find plant performance.