Properties of Pure Substances

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What is a pure substance?

A pure substance is a system which is

(i) ***Homogeneous in composition*** *- means that the composition of each part of the system is the same as the composition of every other part.*

(ii) ***Homogeneous in chemical aggregation*** *- means that the chemical elements must be combined chemically in the same way in all parts of the system.*

(iii) ***Invariable in chemical aggregation*** *- means that the state of chemical combination of the system does not change with time*

for example, Water, nitrogen, helium, and carbon dioxide, are all pure substances.

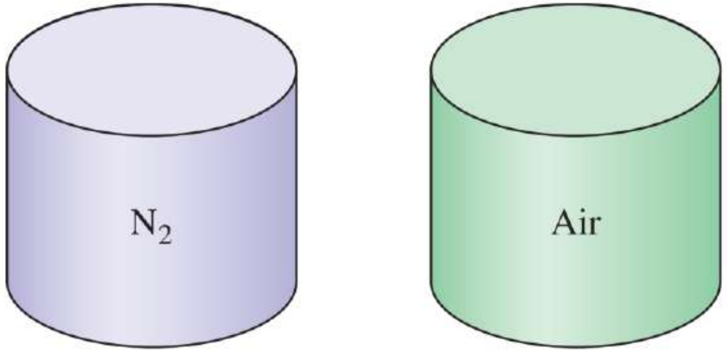
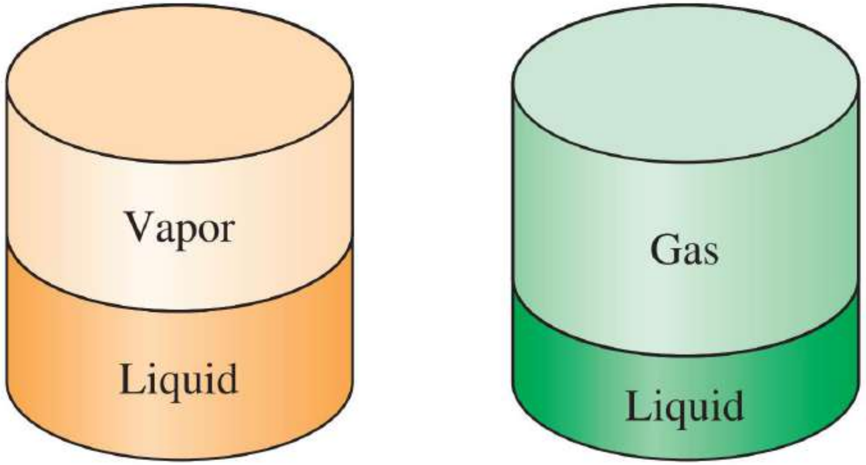


FIGURE
Nitrogen and gaseous air are pure substances.



(*a*) H_2O (*b*) Air

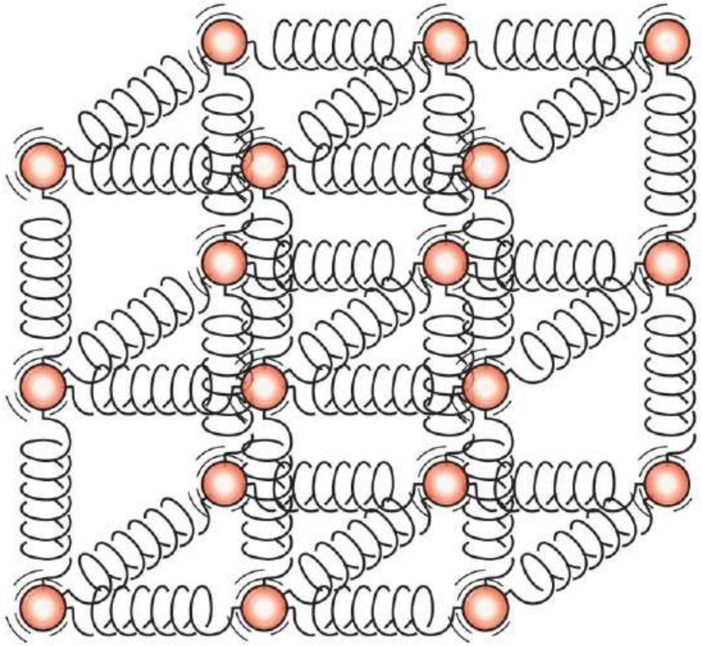
FIGURE
A mixture of liquid and gaseous water is a pure substance, but a mixture of liquid and gaseous air is not.

Phases of a pure substance

A phase is identified as having a distinct molecular arrangement that is homogeneous throughout and separated from the others by easily identifiable boundary surfaces.

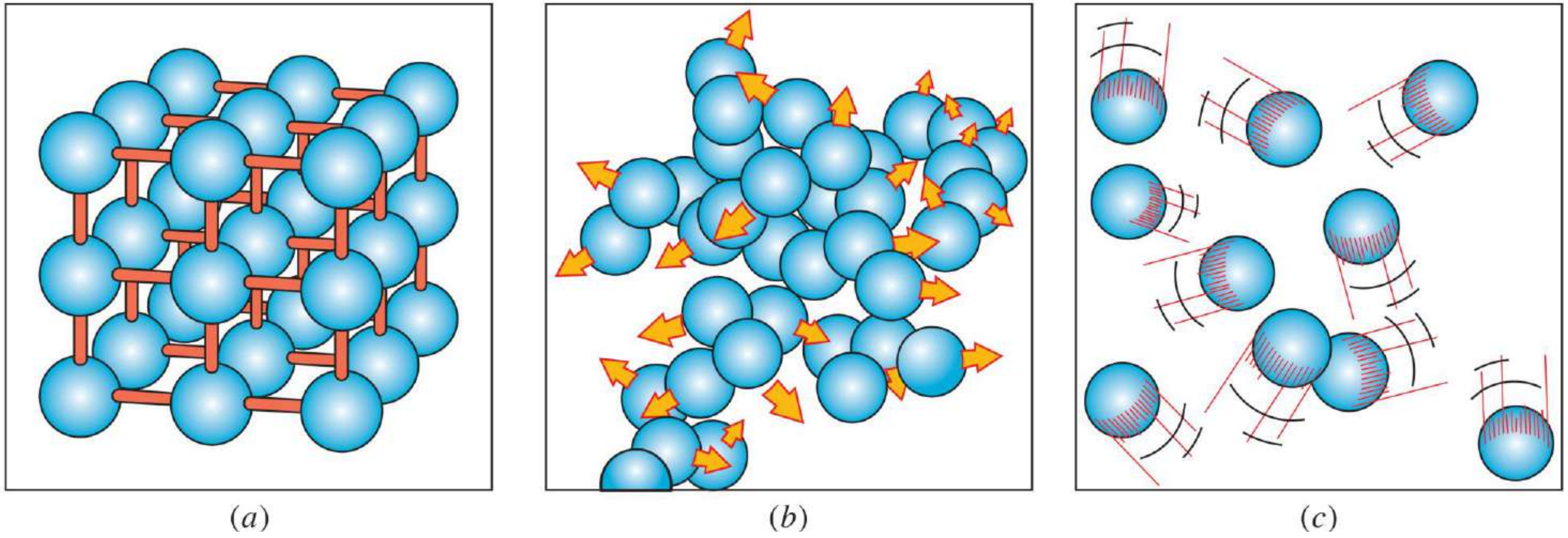
Even though there are three principal phases—***solid***, ***liquid***, and ***gas***—a substance may have several phases within a principal phase, each with a different molecular structure. Carbon, for example, may exist as graphite or diamond in the solid phase. Helium has two liquid phases; iron has three solid phases. Ice may exist at seven different phases at high pressures.

Intermolecular bonds are strongest in solids and weakest in gases. This is why molecules in solids are closely packed together, whereas in gases they are separated by relatively large distances. The molecular spacing in the liquid phase is not much different from that of the solid phase, except the molecules are no longer at fixed positions relative to each other and they can rotate and translate freely.



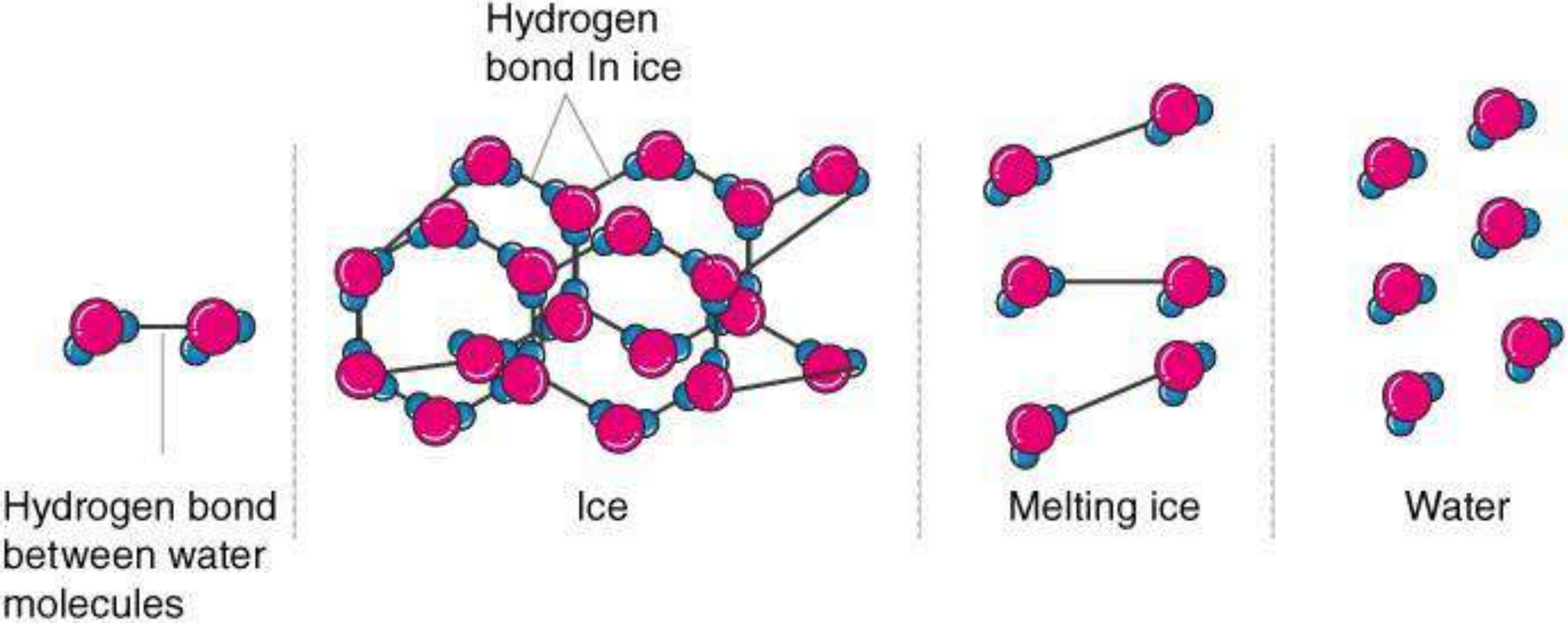
FIGURE

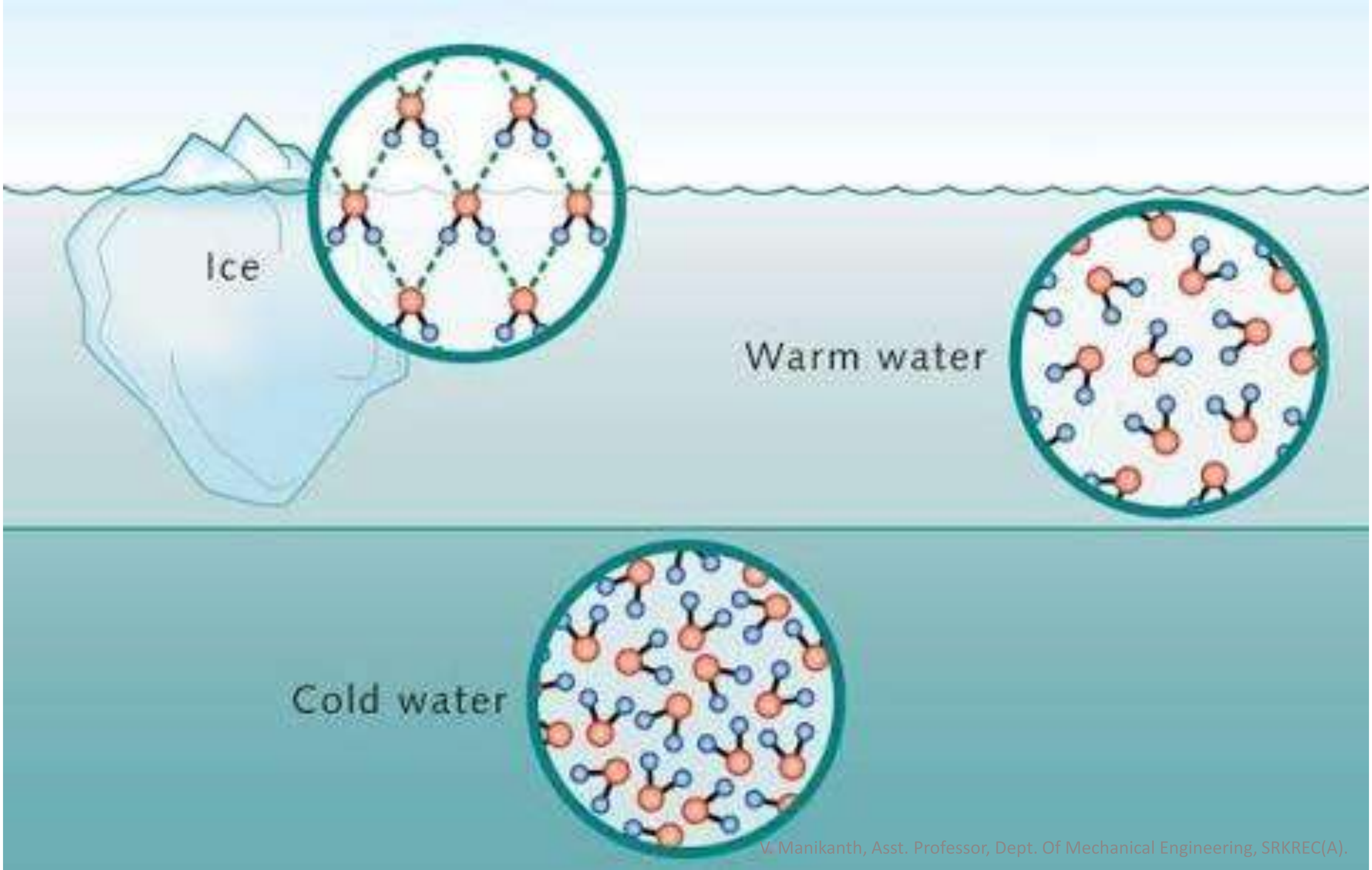
The molecules in a solid are kept at their positions by the large springlike intermolecular forces.



FIGURE

The arrangement of atoms in different phases: (*a*) molecules are at relatively fixed positions in a solid, (*b*) groups of molecules move about each other in the liquid phase, and (*c*) molecules move about at random in the gas phase.


Hydrogen bond between water molecules
Hydrogen bond In ice
Ice
Melting ice
Water


Ice
Warm water
Cold water

Sensible heating: It refers to the heating of substance in single phase. It causes rise in temperature of substance. In case of cooling in above conditions it shall be called sensible cooling.

It takes a large amount of energy to melt a solid or vaporize a liquid. The amount of energy absorbed or released during a phase-change process is called the ***latent heat***. More specifically, the amount of energy absorbed during melting is called the *latent heat of fusion* and is equivalent to the amount of energy released during freezing. Similarly, the amount of energy absorbed during vaporization is called the *latent heat of vaporization* and is equivalent to the energy released during condensation.

Saturation states: Saturation state of a substance refers to the state at which its phase transformation takes place without any change in pressure and temperature. These can be saturated solid state, saturated liquid state and saturated vapour state.

Normal boiling point: It is the temperature at which vapour pressure equals to atmospheric pressure and at this temperature phase change from liquid to gas begins.

PHASE-CHANGE PROCESSES OF PURE SUBSTANCES

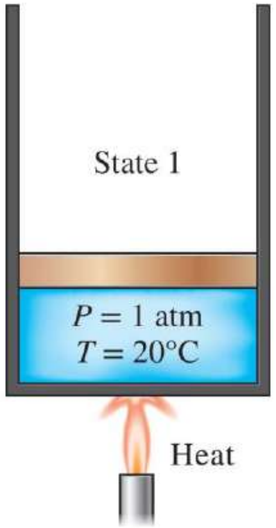


FIGURE 3–5
At 1 atm and 20°C, water exists in the liquid phase (*compressed liquid*).

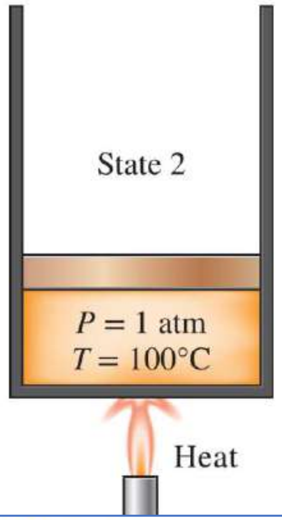


FIGURE 3–6
At 1 atm pressure and 100°C, water exists as a liquid that is ready to vaporize (*saturated liquid*).

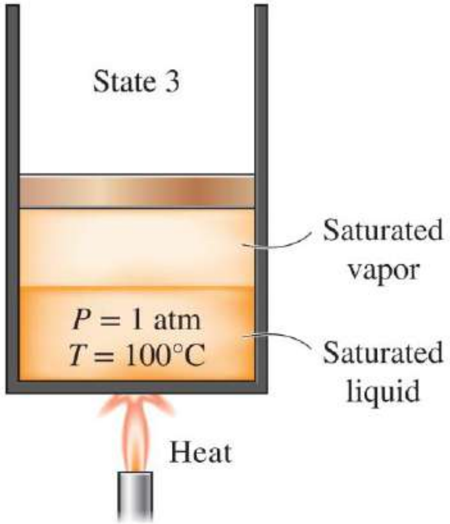


FIGURE 3–7
As more heat is transferred, part of the saturated liquid vaporizes (*saturated liquid–vapor mixture*).

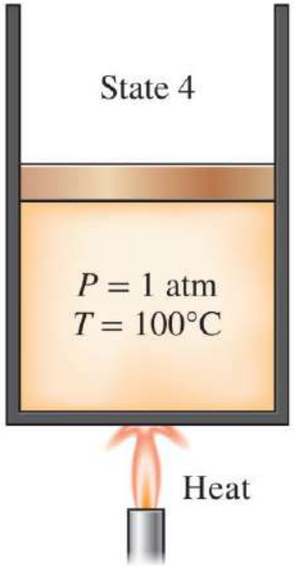


FIGURE 3–8
At 1 atm pressure, the temperature remains constant at 100°C until the last drop of liquid is vaporized (*saturated vapor*).

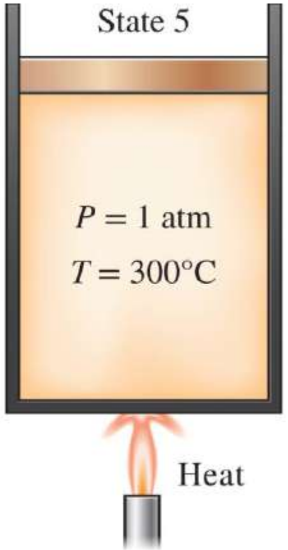


FIGURE 3–9
As more heat is transferred, the temperature of the vapor starts to rise (*superheated vapor*).

Subcooled liquid/Compressed liquid: if the temperature of the liquid water on cooling becomes lower than the saturation temperature for the given pressure, the liquid water is called a subcooled liquid(ref. state 1). The term compressed liquid or sub-cooled liquid is used to distinguish it from saturated liquid.

Saturated liquid: A liquid that is about to vaporize is called a saturated liquid(*ref. state 2*).

Saturated vapor: A vapor that is about to condense is called a saturated vapor(*ref. state 4*). Any heat loss from this vapor will cause some of the vapor to condense (phase change from vapor to liquid).

A substance at states between 2 and 4 is referred to as a ***saturated liquid–vapor mixture*** since the liquid and vapor phases coexist in equilibrium at these states.

A vapor that is not about to condense (i.e., not a saturated vapor) is called a superheated vapor(*ref. state 5*). Therefore, water at state 5 is a ***superheated vapor.***

At a given pressure, the temperature at which a pure substance changes phase is called the ***saturation temperature*** T_{sat}. Likewise, at a given temperature, the pressure at which a pure substance changes phase is called the ***saturation pressure*** P_{sat}.

TABLE

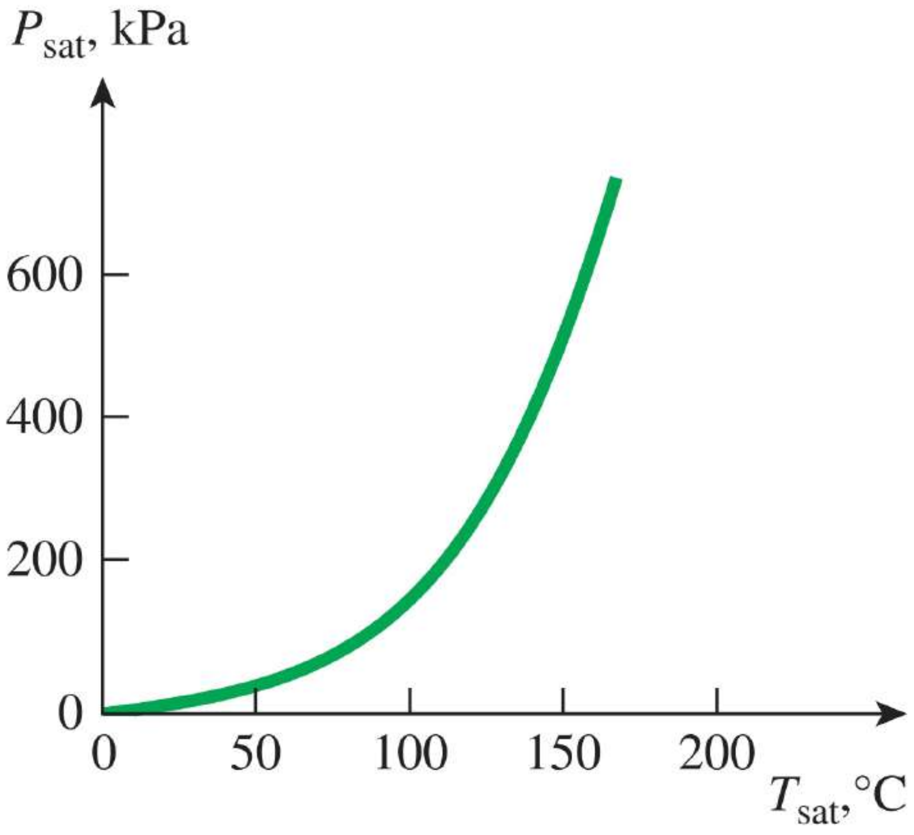
Saturation (or vapor) pressure of water at various temperatures

Temperature T, °C	Saturation pressure P_{sat}, kPa
–10	0.260
–5	0.403
0	0.611
5	0.872
10	1.23
15	1.71
20	2.34
25	3.17
30	4.25
40	7.38
50	12.35
100	101.3 (1 atm)
150	475.8
200	1554
250	3973
300	8581

TABLE

Variation of the standard atmospheric pressure and the boiling (saturation) temperature of water with altitude

Elevation, m	Atmospheric pressure, kPa	Boiling temperature, °C
0	101.33	100.0
1,000	89.55	96.5
2,000	79.50	93.3
5,000	54.05	83.3
10,000	26.50	66.3
20,000	5.53	34.7



FIGURE

The liquid–vapor saturation curve of a pure substance (numerical values are for water).

Water boils at ?

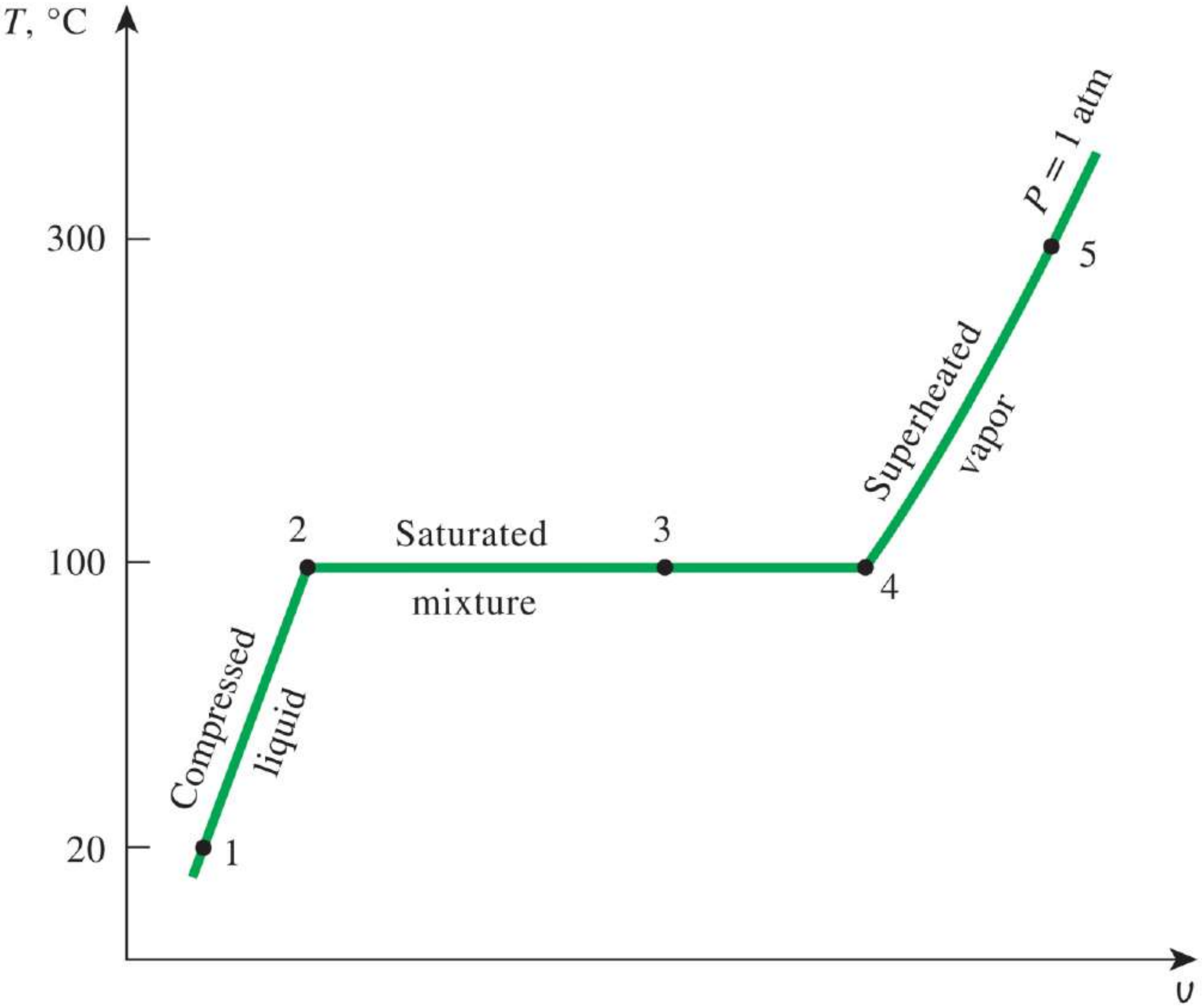
The variations of properties during phase-change processes are best studied and understood with the help of property diagrams. Next, we develop and discuss the T-v, P-v, and P-T diagrams for pure substances.

“water boils at 100°C”

“water boils at 100°C at 1 atm pressure.” ✓

The temperature at which water starts boiling depends on the pressure; therefore, if the pressure is fixed, so is the boiling temperature.

PROPERTY DIAGRAMS FOR PHASE-CHANGE PROCESSES

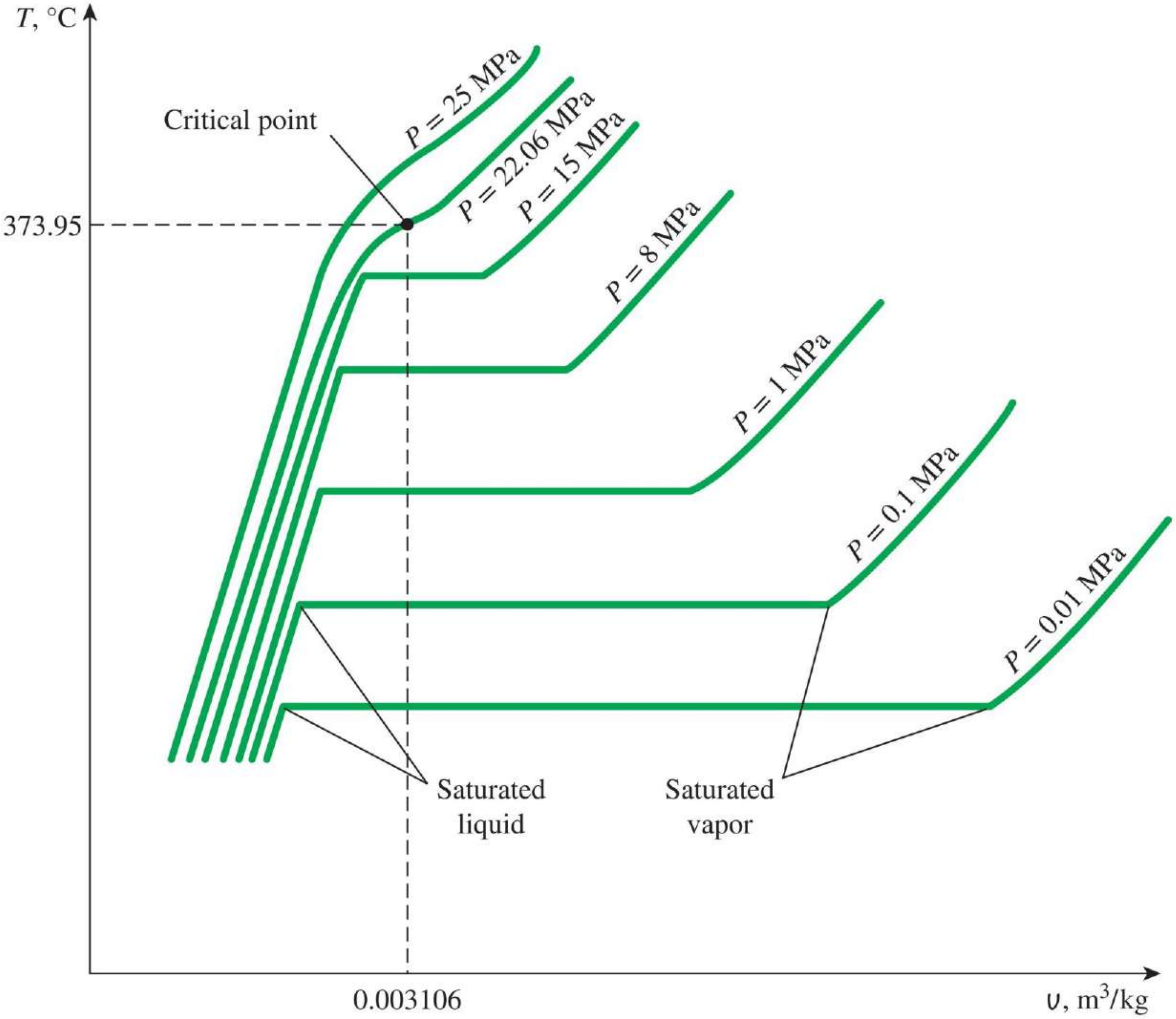


The variations of properties during phase-change processes are best studied and understood with the help of ***property diagrams***. To understand this we develop and discuss the T-v, P-v, and P-T & h-s diagrams for pure substances.

FIGURE

T-*υ* diagram for the heating process of water at constant pressure.

T-υ Diagram



Critical point is defined as the point at which the saturated liquid and saturated vapor states are identical.

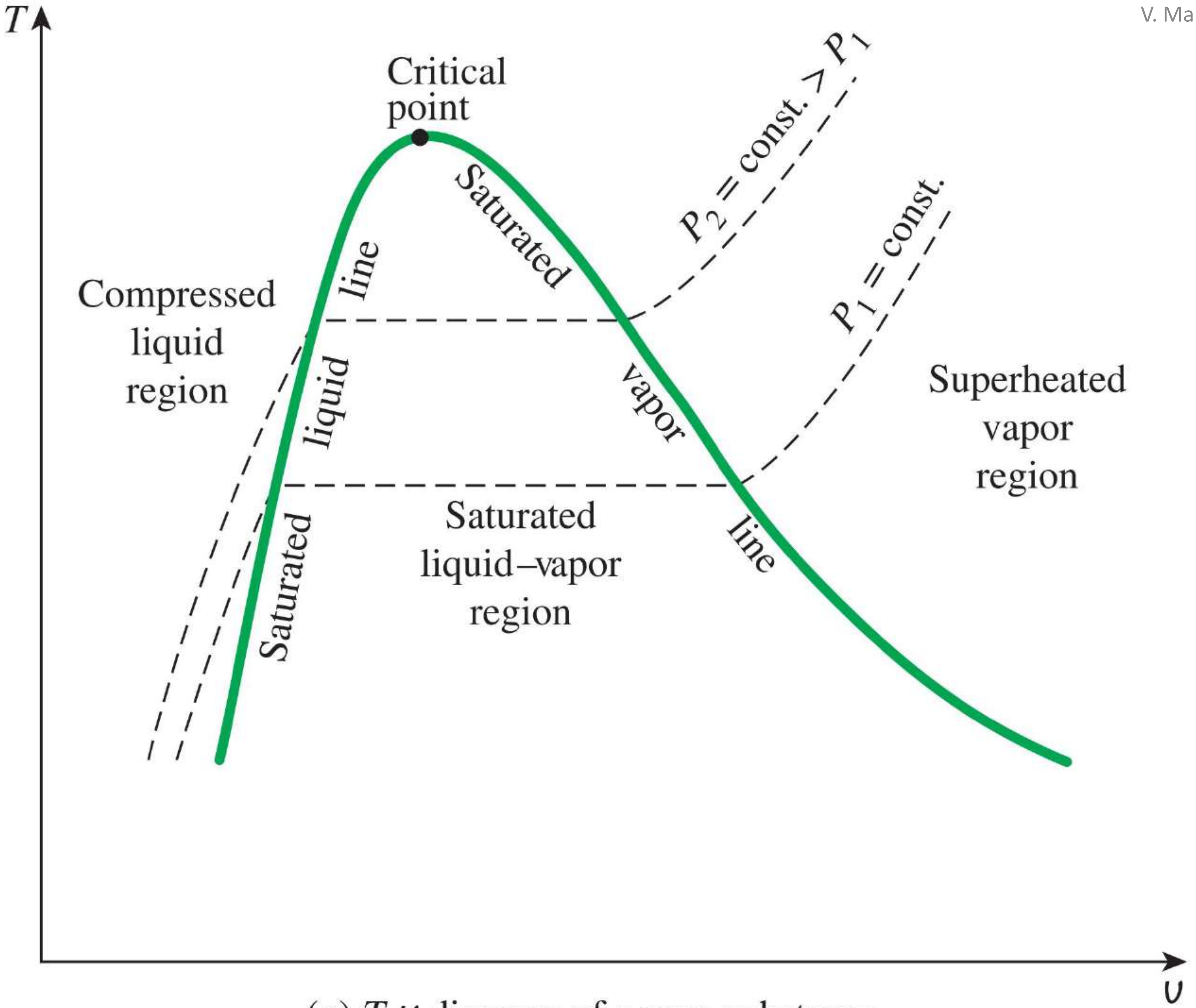
The critical-point properties of water are
Pcr = 22.06 MPa,
Tcr = 373.95°C, and
Vcr = 0.003106 m3/kg.

FIGURE

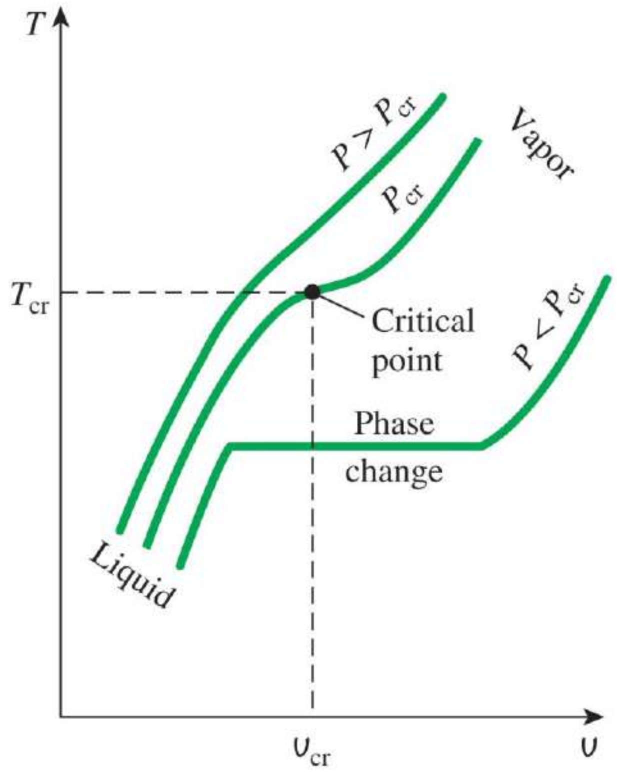
T-υ diagram of constant-pressure phase-change processes of a pure substance at various pressures (numerical values are for water).

The saturated liquid states in Fig. can be connected by a line called the *saturated liquid line*, and saturated vapor states in the same figure can be connected by another line, called the *saturated vapor line.* These two lines meet at the *critical point*, forming a dome. All the compressed liquid states are located in the region to the left of the saturated liquid line, called the *compressed liquid region*. All the superheated vapor states are located to the right of the saturated vapor line, called the *superheated vapor region*. In these two regions, the substance exists in a single phase, a liquid or a vapor. All the states that involve both phases in equilibrium are located under the dome, called the ***saturated liquid–vapor mixture region, or the wet region.***

FIGURE
At triple-point pressure and temperature, a substance exists in three phases in equilibrium.



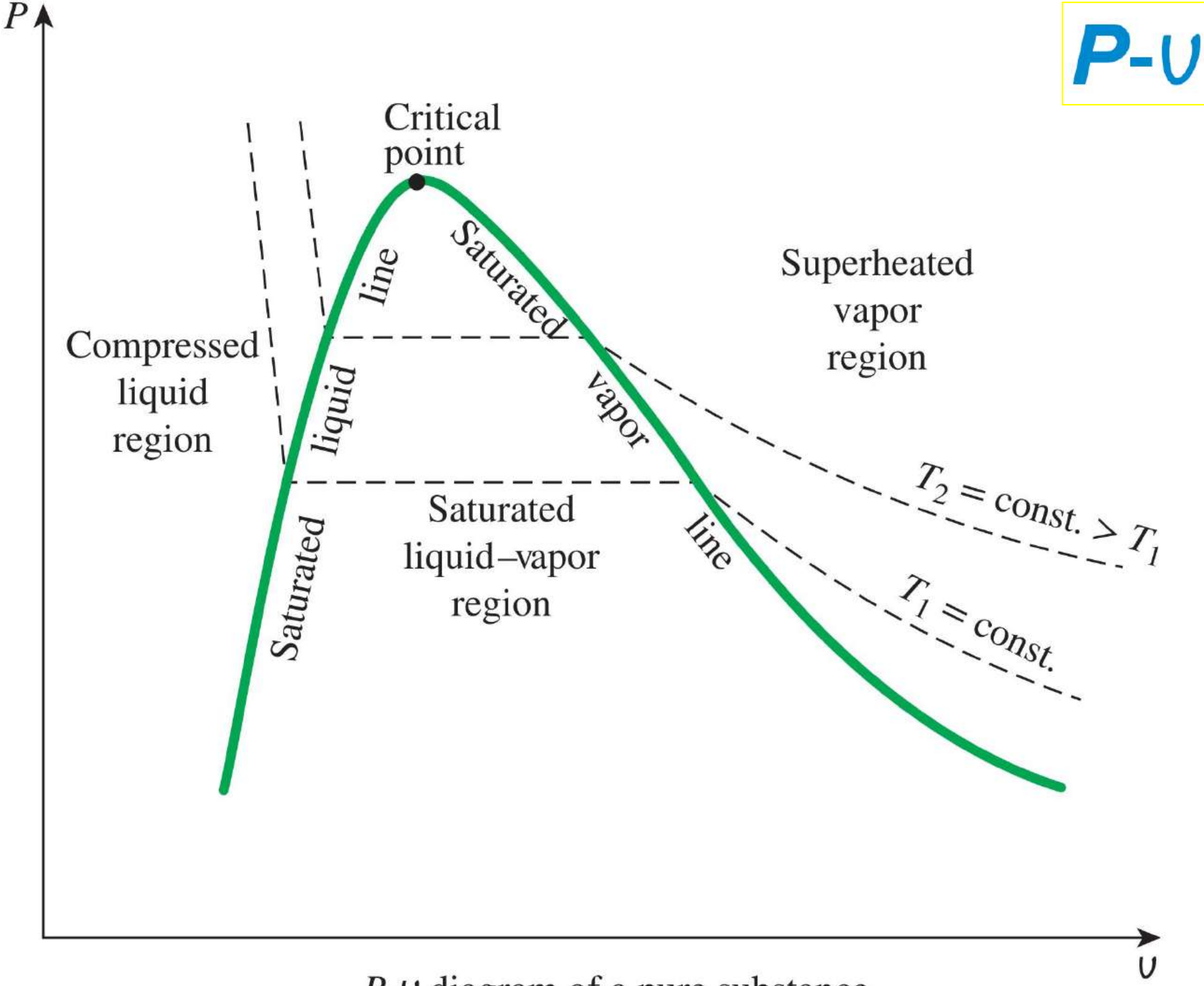
(*a*) T-υ diagram of a pure substance



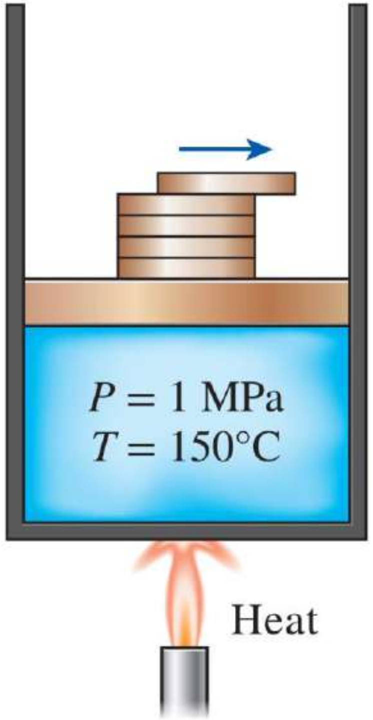
FIGURE

At supercritical pressures ($P > P_{cr}$), there is no distinct phase-change (boiling) process.

P-υ Diagram

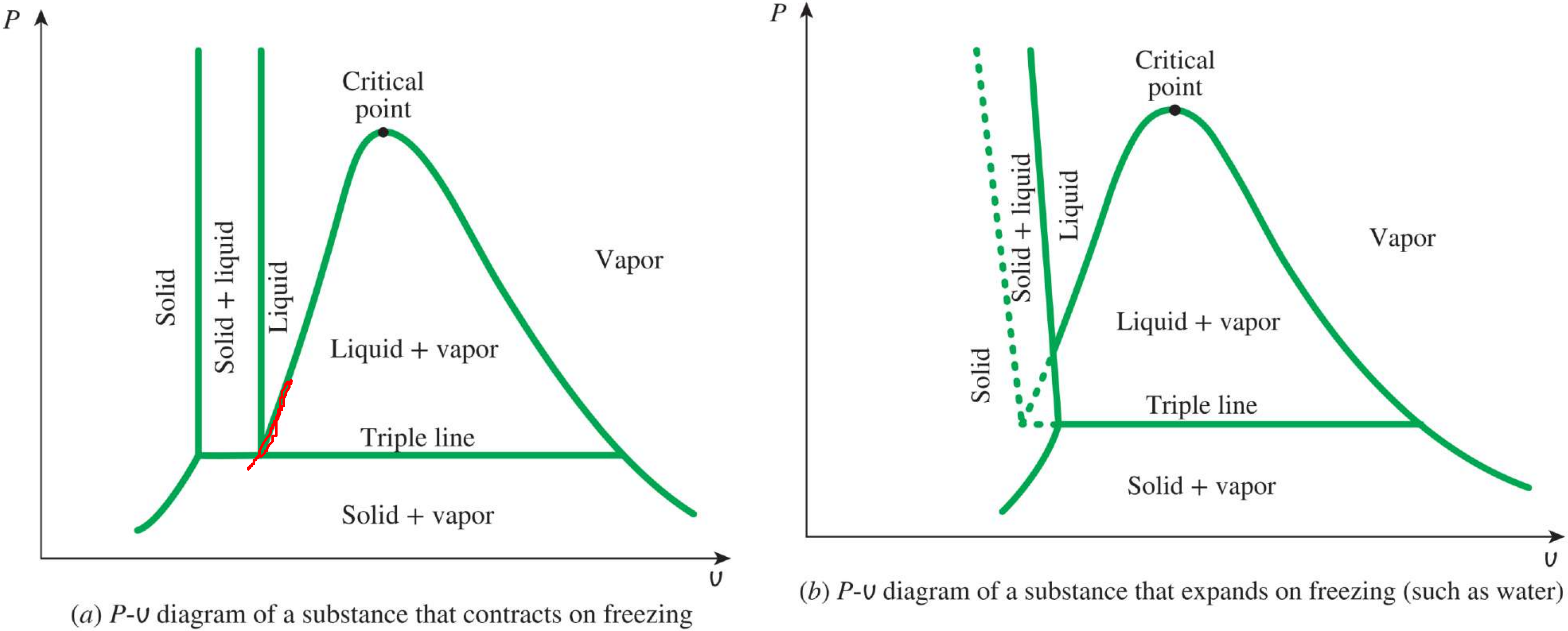


P-υ diagram of a pure substance



FIGURE

The pressure in a piston–cylinder device can be reduced by reducing the weight of the piston.



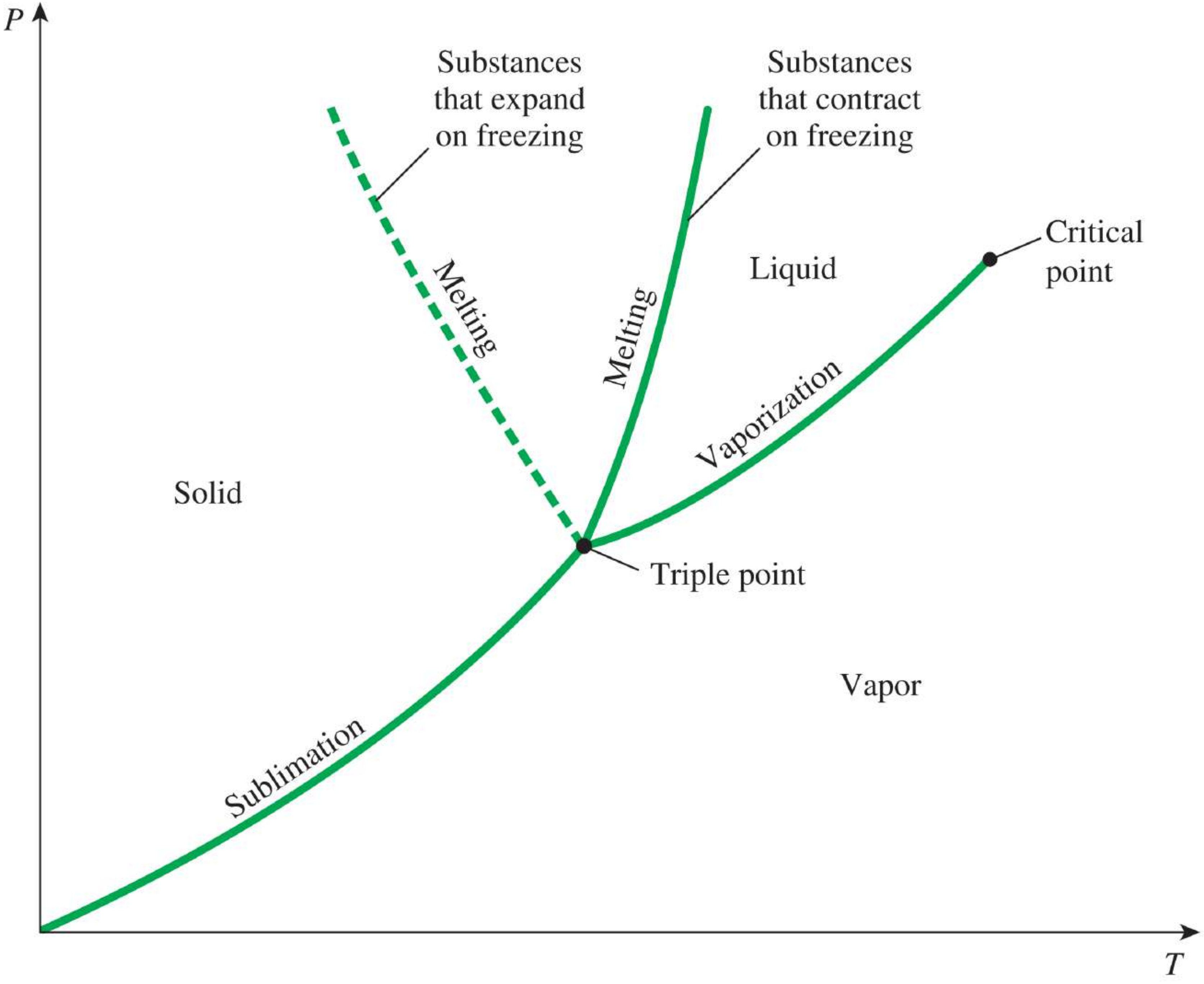
(*a*) *P*-υ diagram of a substance that contracts on freezing

(*b*) *P*-υ diagram of a substance that expands on freezing (such as water)

FIGURE

P-υ diagrams of different substances.

P-T Diagram

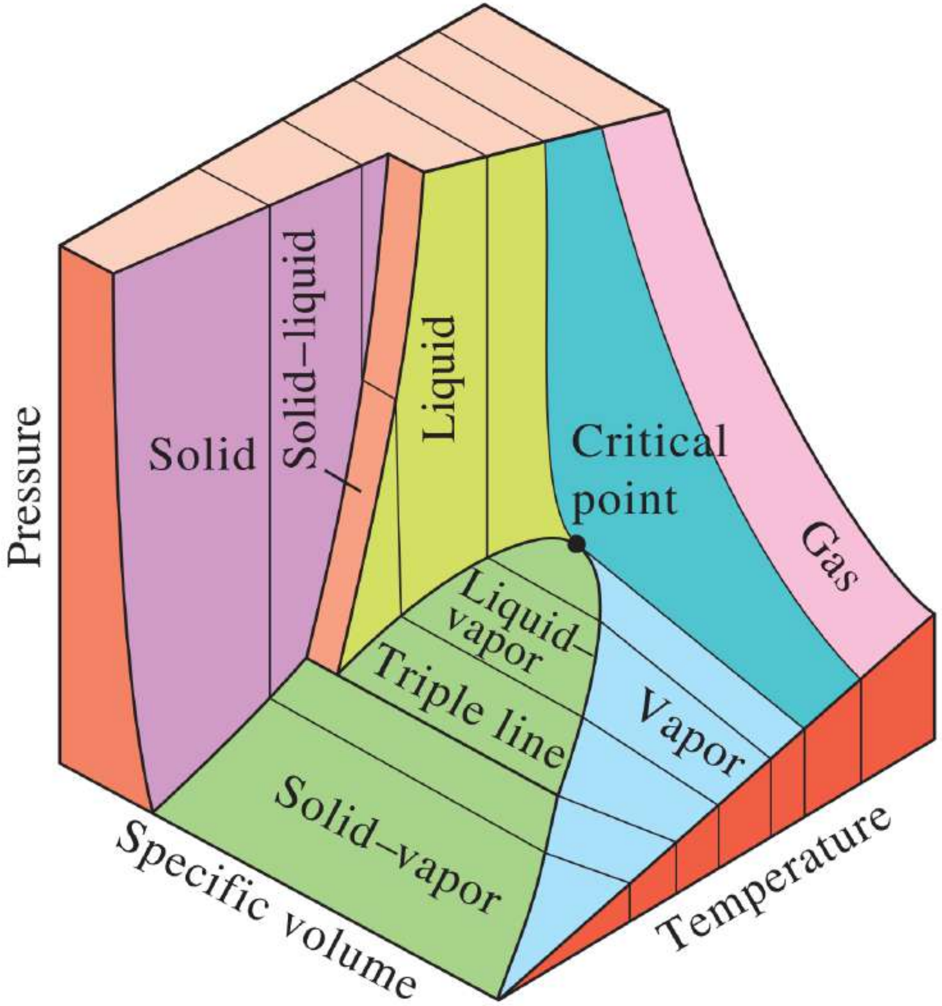


This diagram is often called the ***Phase diagram*** since all three phases are separated from each other by three lines.

FIGURE

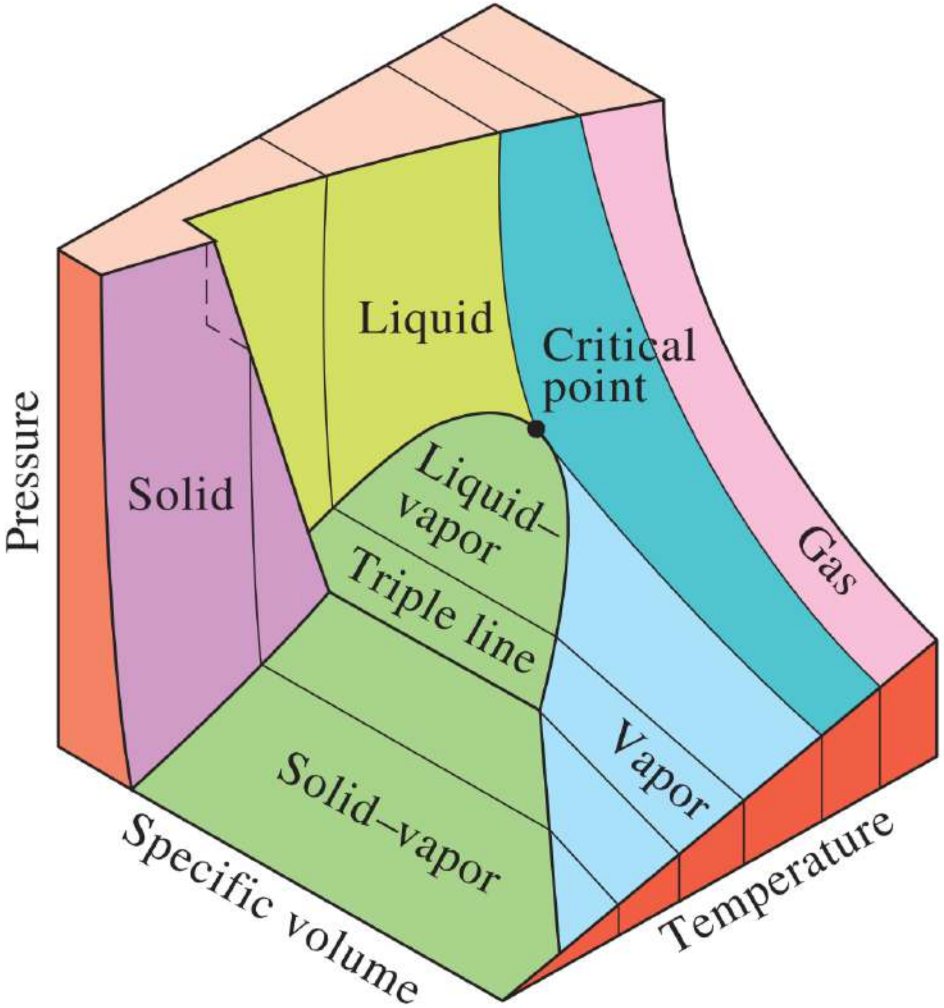
P-T diagram of pure substances.

The P-υ-T Surface



FIGURE

P-υ-*T* surface of a substance that *contracts* on freezing.



FIGURE

P-υ-*T* surface of a substance that expands on freezing (like water).

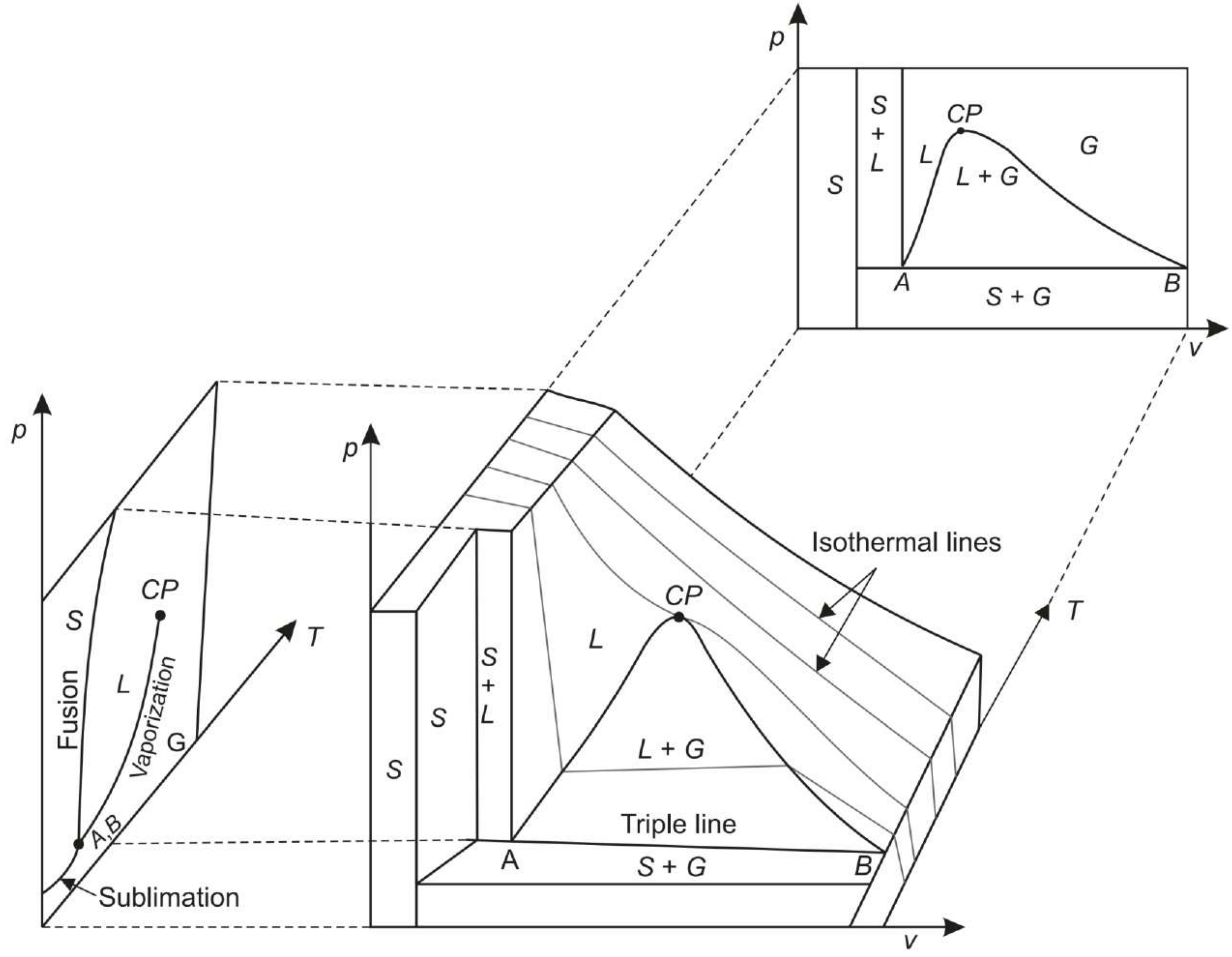


Fig. *p–v-T* surface for a substance which expands on melting or contracts on freezing (like carbon dioxide)

PHASE CHANGE TERMINOLOGY AND DEFINITIONS

Suffices :

Solid	i
Liquid	f
Vapour	g

Phase change	Name	Process	Process suffix
1. Solid-liquid	Fusion	Freezing, melting	if
2. Solid-vapour	Sublimation	Frosting, defrosting	ig
3. Liquid-vapour	Evaporation	Evaporating, Condensing	fg

Saturated vapor: A term including wet and dry vapor.

Dry (saturated) vapor: Vapor which has just completed evaporation. The pressure and temperature of the vapor are the saturation values. Dry vapor is represented by a point g on a diagram.

Wet vapor: The mixture of saturated liquid and dry vapor during the phase change.

Superheated vapor: Vapor whose temperature is greater than the saturation temperature corresponding to the pressure of the vapor.

Degree of superheat: The term used for the numerical amount by which the temperature of a superheated vapor exceeds the saturation temperature.

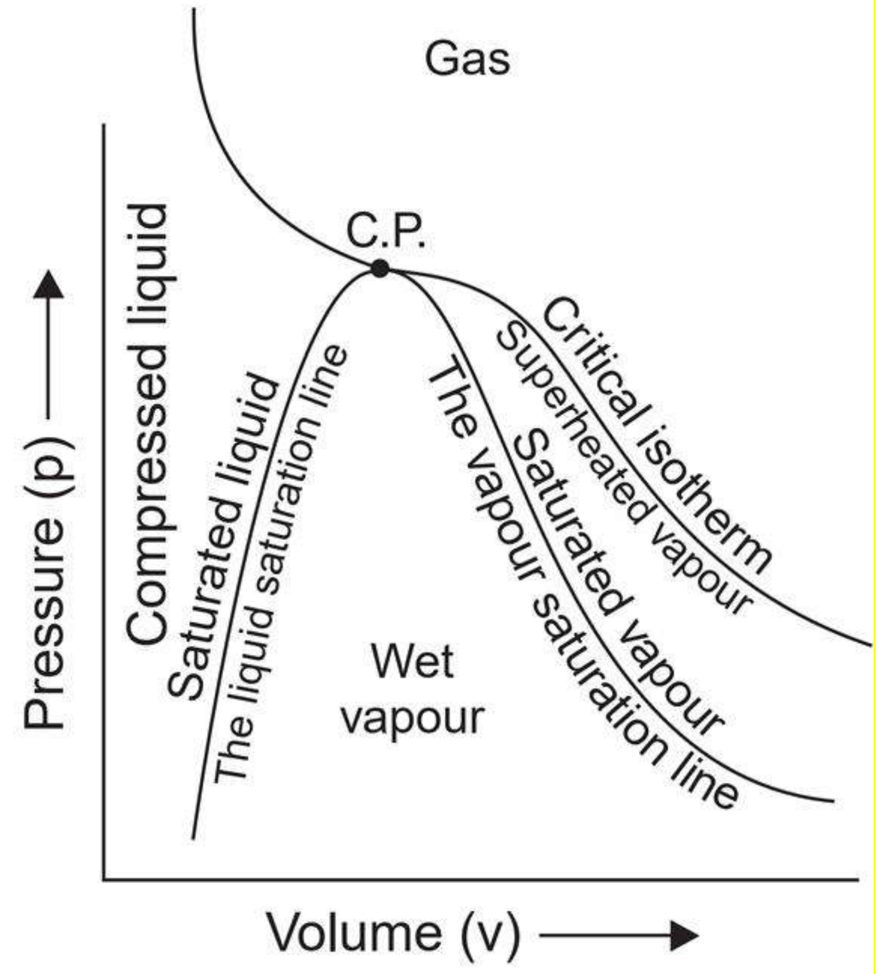


Fig. Phase change terminology.

IMPORTANT TERMS RELATING STEAM FORMATION

1. Sensible heat of water (h_f): It is defined as the quantity of heat absorbed by 1 kg of water when it is heated from 0°C (freezing point) to boiling point.

2. Latent heat or hidden heat (h_{fg}): It is the amount of heat required to convert water at a given temperature and pressure into steam at the same temperature and pressure. The value of latent heat is not constant and varies according to pressure variation.

3. Dryness fraction (x): The term dryness fraction is related with wet steam. It is defined as the ratio of the mass of actual dry steam to the mass of steam containing it. It is usually expressed by the symbol ‘x’

If

m_s = Mass of dry steam contained in steam considered, and

m_w = Weight of water particles in suspension in the steam considered, Then, $x = \dfrac{m_s}{m_s + m_w}$

Thus if in 1 kg of wet steam 0.9 kg is the dry steam and 0.1 kg water particles then $x = 0.9$

4. Total heat or enthalpy of wet steam (h): *It is defined as the quantity of heat required to convert 1 kg of water at 0°C into wet steam at constant pressure*. It is the sum of total heat of water and the latent heat and this sum is also called **enthalpy.**

In other words, $h = h_f + xh_{fg}$

If steam is dry and saturated, then $x = 1$ and $h_g = h_f + h_{fg}$.

5. Superheated steam: When steam is heated after it has become dry and saturated, it is called *superheated steam* and the process of heating is called *superheating*. Superheating is always carried out at constant pressure. The additional amount of heat supplied to the steam during superheating is called as *'Heat of superheat'.*

The total heat of superheated steam is given by

$$h_{sup} = h_f + h_{fg} + c_{ps}\,(T_{sup} - T_s)$$

6. Volume of wet and dry steam. If the steam has dryness fraction of x, then 1 kg of this steam will contain x kg of dry steam and $(1 - x)$ kg of water. If v_f is the volume of 1 kg of water and v_g is the volume of 1 kg of perfect dry steam (also known as specific volume), then volume of 1 kg of wet steam = volume of dry steam + volume of water.

$$= xv_g + (1 - x)v_f$$

Note. The volume of v_f at low pressures is very small and is generally neglected. Thus is general, the volume of 1 kg of wet steam is given by, xv_g and density $\frac{1}{xv_g}$ kg/m³.

$$\begin{aligned}
&= xv_g + v_f - xv_f \\
&= v_f + x(v_g - v_f) \\
&= v_f + xv_{fg} \\
&= v_f + xv_{fg} + v_{fg} - v_{fg} \\
&= (v_f + v_{fg}) - (1 - x)\,v_{fg} \\
&= v_g - (1 - x)v_{fg}
\end{aligned}$$

7. Volume of superheated steam. As superheated steam behaves like a perfect gas its volume can be found out in the same way as the gases.

If, v_g = Specific volume of dry steam at pressure p,

T_s = Saturation temperature in K,

T_{sup} = Temperature of superheated steam in K, and

v_{sup} = Volume of 1 kg of superheated steam at pressure p,

Then,

$$\frac{p \cdot v_g}{T_s} = \frac{p \cdot v_{\text{sup}}}{T_{\text{sup}}}$$

or,

$$v_{sup} = \frac{v_g \, T_{sup}}{T_s}$$

Following are the thermodynamic properties of steam which are tabulated in the form of table :

p = Absolute pressure (bar or kPa) ;
t_s = Saturation temperature (°C) ;
h_f = Enthalpy of saturated liquid (kJ/kg) ;
h_{fg} = Enthalpy or latent heat of vapourisation (kJ/kg) ;
h_g = Enthalpy of saturated vapour (steam) (kJ/kg) ;
s_f = Entropy of saturated liquid (kJ/kg K) ;
s_{fg} = Entropy of vapourisation (kJ/kg K) ;
s_g = Entropy of saturated vapour (steam) (kJ/kg K) ;
v_f = Specific volume of saturated liquid (m^3/kg) ;
v_g = Specific volume of saturated vapour (steam) (m^3/kg).

Also, $h_{fg} = h_g - h_f$ ...... Change of enthalpy during evaporation
$s_{fg} = s_g - s_f$ ...... Change of entropy during evaporation
$v_{fg} = v_g - v_f$ ...... Change of volume during evaporation.

The above mentioned properties at different pressures are tabulated in the form of tables as under :

The internal energy of steam ($u = h - pv$) is also tabulated in some steam tables.

EXTERNAL WORK DONE DURING EVAPORATION

When water is evaporated to form *saturated* steam, its volume increases from v_f to v_g at a constant pressure, and thus external work is done by steam due to increase in volume. The energy for doing the work is obtained during the absorption of latent heat. This work is called **external work of evaporation** and is given by $p(v_g - v_f)$.

i.e., External work of evaporation $= p(v_g - v_f)$

As at low pressure v_f is very small and hence neglected, work of evaporation is

$$p \cdot v_g$$

In case of *wet* steam with dryness fraction x, work of evaporation will be

$$pxv_g$$

INTERNAL LATENT HEAT

The latent heat consists of true latent heat and the work of evaporation. This true latent heat is called the *internal latent heat* and may also be found as follows :

$$\text{Internal latent heat} = h_{fg} - \frac{pv_g}{J}$$

J = 1 in SI units.

Example1. *Calculate the dryness fraction (quality) of steam which has 1.5 kg of water in suspension with 50 kg of steam.*

Solution. Mass of dry steam, $m_s = 50$ kg

Mass of water in suspension, $m_w = 1.5$ kg

$$\therefore \textbf{Dryness fraction, } x = \frac{\text{Mass of dry steam}}{\text{Mass of dry steam + mass of water in suspension}}$$

$$= \frac{m_s}{m_s + m_w} = \frac{50}{50 + 1.5} = \mathbf{0.971.} \quad \textbf{(Ans.)}$$

STEAM TABLES

Absolute pressure	*Temperature* °C	*Specific enthalpy* kJ / kg			*Specific entropy* kJ / kg K			*Specific volume* m^3/kg	
bar, p	t_s	h_f	h_{fg}	h_g	s_f	s_{fg}	s_g	v_f	v_g
1.0	99.6	417.5	2257.9	2675.4	1.3027	6.0571	7.3598	0.001043	1.6934
50.0	263.9	1154.9	1639.7	2794.2	2.9206	3.0529	5.9735	0.001286	0.00394
100.0	311.1	1408.0	1319.7	2727.7	3.3605	2.2593	5.6198	0.001452	0.01811

Table 1.1 Saturated steam on pressure based

Absolute pressure	Saturated temperature	Specific volume	Specific internal energy	Specific enthalpy	Specific entropy
p kP a	T_{sat} °C	v_f v_{fg} v_g m^3/kg	u_f u_g kJ/kg	h_f h_{fg} h_g kJ/kg	s_f s_{fg} s_g kJ/kgK

Table 1.2 Saturated steam on temperature based

Saturated temperature	Absolute pressure	Specific volume	Specific internal energy	Specific enthalpy	Specific entropy
T_{sat} °C	P kPa	v_f v_{fg} v_g m^3/kg	u_f u_g kJ/kg	h_f h_{fg} h_g kJ/kg	s_f s_{fg} s_g kJ/kgK

Table 1.3 Superheated steam table

Absolute pressure: p kPa	Saturated temperature: T_{sat} °C	Specific volume: v Specific entropy: h Specific entropy: s	Temperature in °C				
			200	250	300	350	400
500	151.9	v: m^3/kg h: kJ/kg s: kJ/kgK	0.4249 2855.4 7.059	0.4749 2960.7 7.271	0.5226 3064.2 7.460	0.5701 3167.7 7.633	0.6173 3271.9 7.945

Example Using the steam tables, estimate the saturation temperature and specific volume, specific enthalpy and specific entropy of saturated liquid and vapour at 3 MPa.

Solution From the saturated steam table based on pressure , it is found that saturation temperature of water corresponds to 3 MPa is $t_{\mathrm{sat}} = 233.9\ \mathrm{C}$

Specific volume of saturated liquid is $v_f = 0.001216\ \mathrm{m^3/kg}$

Specific volume of saturated vapour is $v_g = 0.06668\ \mathrm{m^3/kg}$

Specific enthalpy of saturated liquid is $h_f = 1008.4\ \mathrm{kJ/kg}$

Specific enthalpy of saturated vapour $h_g = 2804.1\ \mathrm{kJ/kg}$

Specific entropy of saturated liquid is $s_f = 2.6462\ \mathrm{kJ/kg\text{-}K}$

Specific entropy of saturated vapour is $s_g = 6.1878\ \mathrm{kJ/kg\text{-}K}$

Example Using the steam tables, estimate saturation pressure and the specific volume, specific enthalpy and specific entropy of saturated liquid and vapour at 200°C.

Solution From the saturated steam table based on temperature, it is found that saturation pressure of water corresponds to 200°C is $P_{\text{sat}} = 1.554 \text{ MPa}$.

Specific volume of saturated liquid is $v_f = 0.001156 \text{ m}^3/\text{kg}$

Specific volume of saturated vapour is $v_g = 0.1274 \text{ m}^3/\text{kg}$

Specific enthalpy of saturated liquid is $h_f = 852.4 \text{ kJ/kg}$

Specific enthalpy of saturated vapour $h_g = 2793.2 \text{ kJ/kg}$

Specific entropy of saturated liquid is $s_f = 2.3313 \text{ kJ/kg-K}$

Specific entropy of saturated vapour is $s_g = 6.4331 \text{ kJ/kg-K}$

ENTHALPY-ENTROPY (h-s) CHART or MOLLIER DIAGRAM

Dr. Mollier, in 1904, conceived the idea of plotting total heat against entropy, and his diagram is more widely used than any other entropy diagram, since the work done on vapour cycles can be scaled from this diagram directly as a length ; whereas on T-s diagram it is represented by an area.

Lines of constant pressure are indicated by p_1, p_2 etc., lines of constant temperature by T_1, T_2, etc.

— Any two independent properties which appear on the chart are sufficient to define the state (e.g., p_1 and x_1 define state 1 and h can be read off the vertical axis).

— In the superheat region, pressure and temperature can define the state (e.g., p3 and T4 define the state 2, and h2 can be read off)

— A line of constant entropy between two state points 2 and 3 defines the properties at all points during an *isentropic process* between the two states.

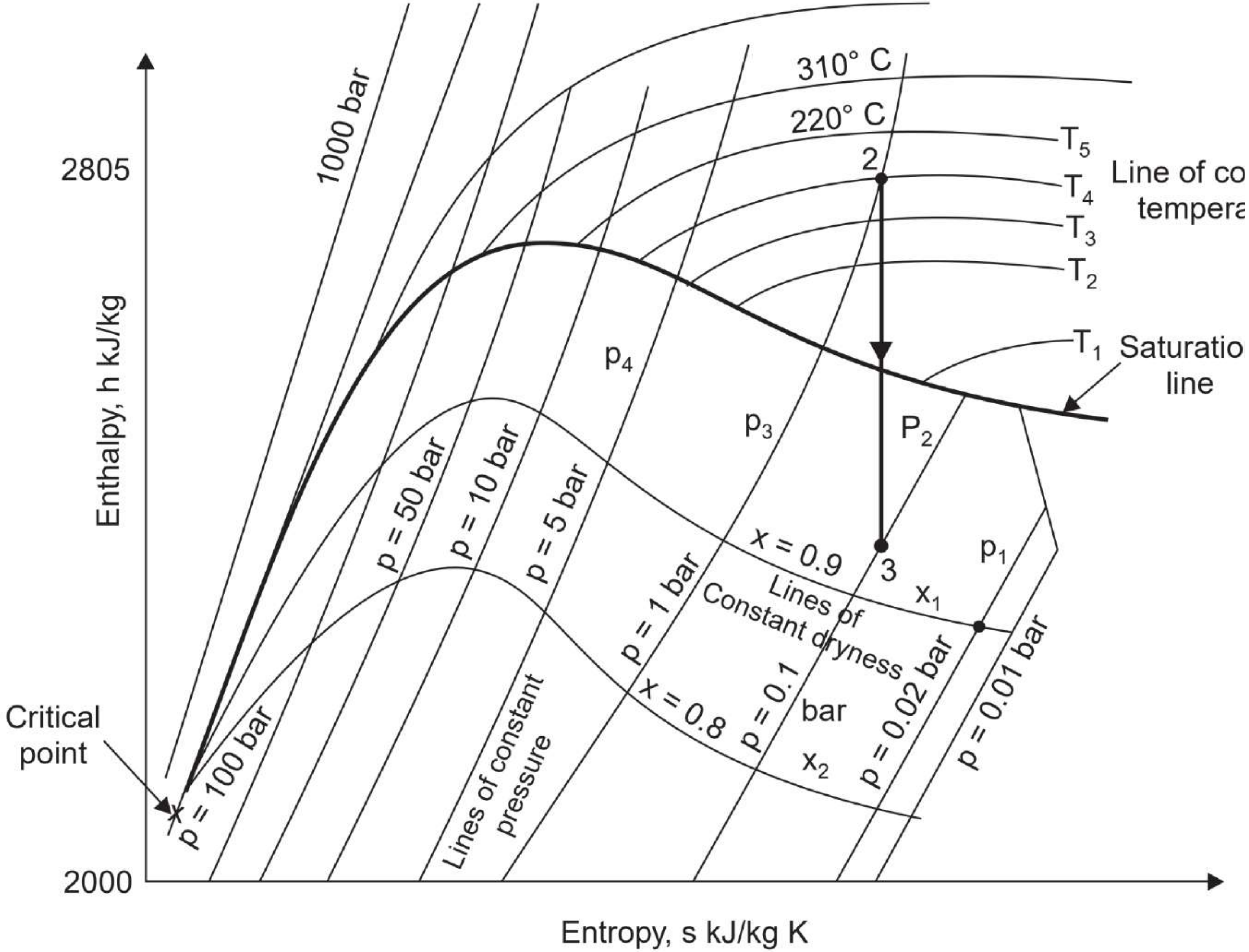


Fig. Enthalpy-entropy (*h*-*s*) chart.

Points to consider before solving the problems

1. The steam table gives the properties on per kg basis. For the different masses they should be multiplied by the given mass.
2. If the value of the independent property of the column 1, i.e., temperature or pressure is not included in steam tables, the properties should be obtained by linear interpolation.
3. A meagre negligible variation in answers is inevitable due to usage of different steam tables.
4. In case the nature of pressure is not specified, it should be assumed absolute pressure.
5. The steam tables give the values of properties above 0°C. Hence if the initial temperature of water is other than 0°C then the initial enthalpy of water can be obtained from the temperature entry steam table at a given temperature. The enthalpy of fluid is then obtained by deducting the initial enthalpy from the total enthalpy of steam.

DETERMINATION OF DRYNESS FRACTION OF STEAM

The dryness fraction of steam can be measured by using the following ***calorimeters*** :

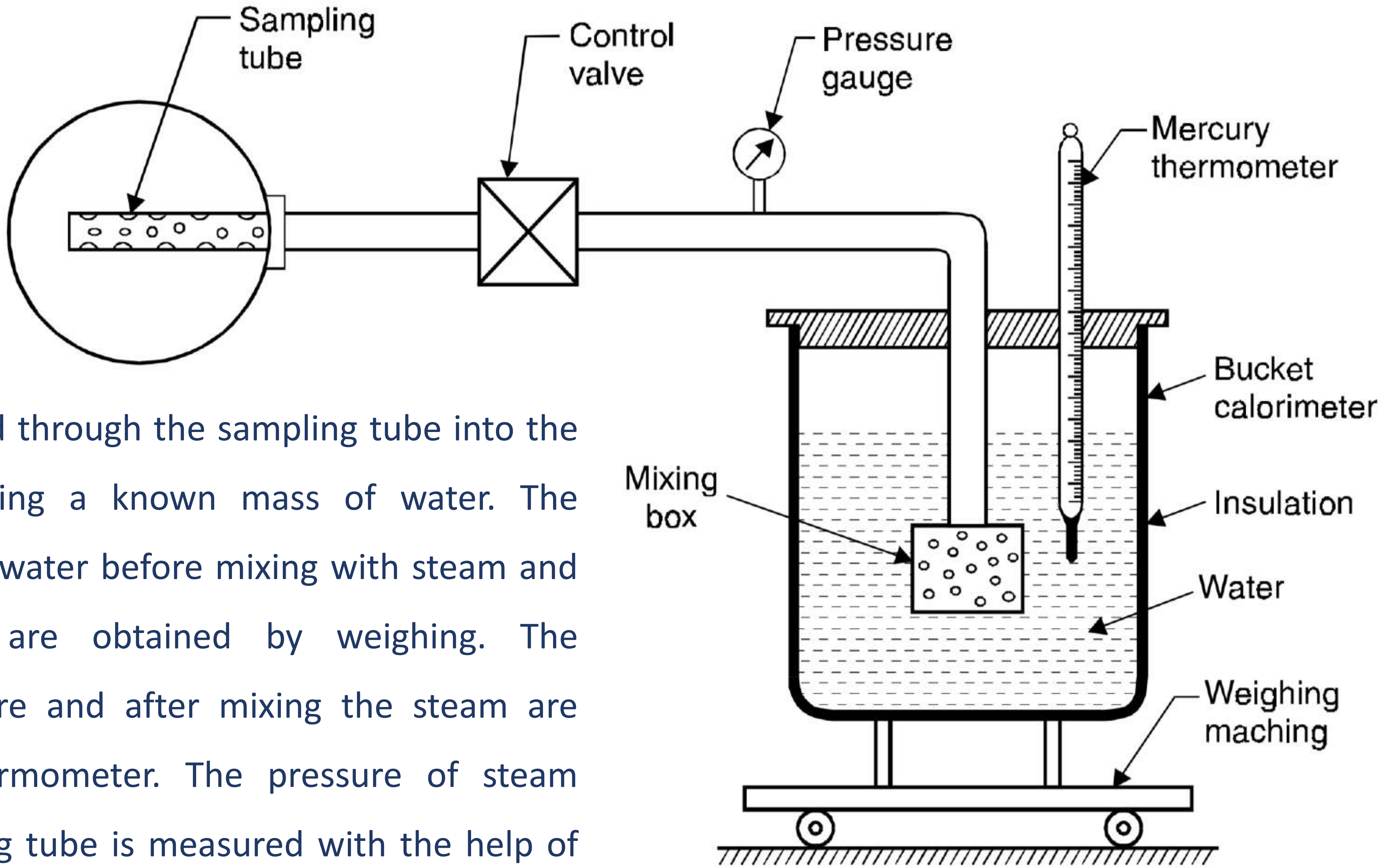
1. Tank or bucket calorimeter
2. Throttling calorimeter
3. Separating and throttling calorimeter.

The working principle of all the calorimeters is to bring the state of the substance from the two-phase region to the single phase region (either compressed liquid or superheated vapour region). This is because in the two-phase region, pressure and temperature are not independent variable, one is independent and the other is dependent. On the other hand, both in compressed liquid and superheated vapour region, both pressure and temperature are independent variables.

Tank or Bucket Calorimeter:

Principle:

A known mass of steam is passed through a known mass of water and steam is completely condensed. The heat lost by steam is equated to heat gained by the water.



Working:

The steam is passed through the sampling tube into the bucket calorimeter containing a known mass of water. The weights of calorimeter with water before mixing with steam and after mixing the steam are obtained by weighing. The temperature of water before and after mixing the steam are measured by mercury thermometer. The pressure of steam passed through the sampling tube is measured with the help of pressure gauge.

Fig. 3.14. Tank or bucket calorimeter.

Let, p_s = Gauge pressure of steam (bar),

p_a = Atmospheric pressure (bar),

t_s = Daturation temperature of steam known from steam table at pressure $(p_s + p_a)$,

h_{fg} = Latent heat of steam,

x = Dryness fraction of steam,

c_{pw} = Specific heat of water,

c_{pc} = Specific heat of calorimeter,

m_c = Mass of calorimeter, kg,

m_{cw} = Mass of calorimeter and water, kg,

$m_w = (m_{cw} - m_{\underline{c}})$ = Mass of water in calorimeter, kg,

m_{cws} = Mass of calorimeter, water and condensed steam, kg,

$m_s = (m_{cws} - m_{cw})$ = Mass of steam condensed in calorimeter, kg,

t_{cw} = Temperature of water and calorimeter before mixing the steam, °C, and

t_{cws} = Temperature of water and calorimeter after mixing the steam, °C.

Neglecting the losses and *assuming* that the *heat lost by steam is gained by water and calorimeter*, we have

$$(m_{cws} - m_{cw})\,[xh_{fg} + c_{pw}\,(t_s - t_{cws})] = (m_{cw} - m_c)c_{pw}\,(t_{cws} - t_{cw}) + m_c\,c_{pc}\,(t_{cws} - t_{cw})$$

$$\therefore \quad m_s[xh_{fg} + c_{pw}\,(t_s - t_{cws})] = (t_{cws} - t_{cw})\,[m_{cw} - m_c)(c_{pw} + m_c c_{pc}] \quad \text{...(3.19)}$$

or

$$m_s[xh_{fg} + c_{pw}\,(t_s - t_{cws})] = (t_{cws} - t_{cw})(m_w c_{pw} + m_c c_{pc})$$

The $m_c c_{pc}$ is known as *water equivalent of calorimeter*.

The value of dryness fraction 'x' can be found by solving the above equation.

The value of dryness fraction found by this method involves some *inaccuracy* since losses due to convection and radiation are *not* taken into account.

The calculated value of dryness fraction neglecting losses is *always less* than the actual value of the dryness.

Example *Steam at a pressure of 5 bar passes into a tank containing water where it gets condensed. The mass and temperature in the tank before the admission of steam are 50 kg and 20°C respectively. Calculate the dryness fraction of steam as it enters the tank if 3 kg of steam gets condensed and resulting temperature of the mixture becomes 40°C. Take water equivalent of tank as 1.5 kg.*

Solution. Pressure of steam, $p = 5$ bar
Mass of water in the tank = 50 kg
Initial temperature of water = 20°C
Amount of steam condensed, $m_s = 3$ kg
Final temperature after condensation of steam = 40°C
Water equivalent of tank = 1.5 kg

Dryness fraction of steam, x :

At 5 bar. From steam tables,

$$h_f = 640.1 \text{ kJ/kg} \; ; \; h_{fg} = 2107.4 \text{ kJ/kg}$$

Total mass of water, m_w = mass of water in the tank + water equivalent of tank

$$= 50 + 1.5 = 51.5 \text{ kg}$$

Also, heat lost by steam = heat gained by water

$$m_s\,[(h_f + xh_{fg}) - 1 \times 4.18\,(40 - 0)] = m_w[1 \times 4.18\,(40 - 20)]$$

or $$3[(640.1 + x \times 2107.4) - 4.18 \times 40] = 51.5 \times 4.18 \times 20$$

or $$3(472.9 + 2107.4x) = 4305.4$$

or $$472.9 + 2107.4x = 1435.13$$

$\therefore$ $$x = \frac{1435.13 - 472.9}{2107.4} = 0.456.$$

Hence **dryness fraction of steam = 0.456.** **(Ans.)**

Throttling Calorimeter

The steam to be sampled is taken from the pipe by means of suitable positioned and dimensioned sampling tube. It passes into an insulated container and is throttled through an orifice to atmospheric pressure. Here the temperature is taken and the steam ideally should have about 5.5 K of superheat.

The throttling process is shown on h-s diagram in Fig. by the line 1-2. If steam initially wet is throttled through a sufficiently large pressure drop, then the steam at state 2 will become superheated. State 2 can then be defined by the measured pressure and temperature.

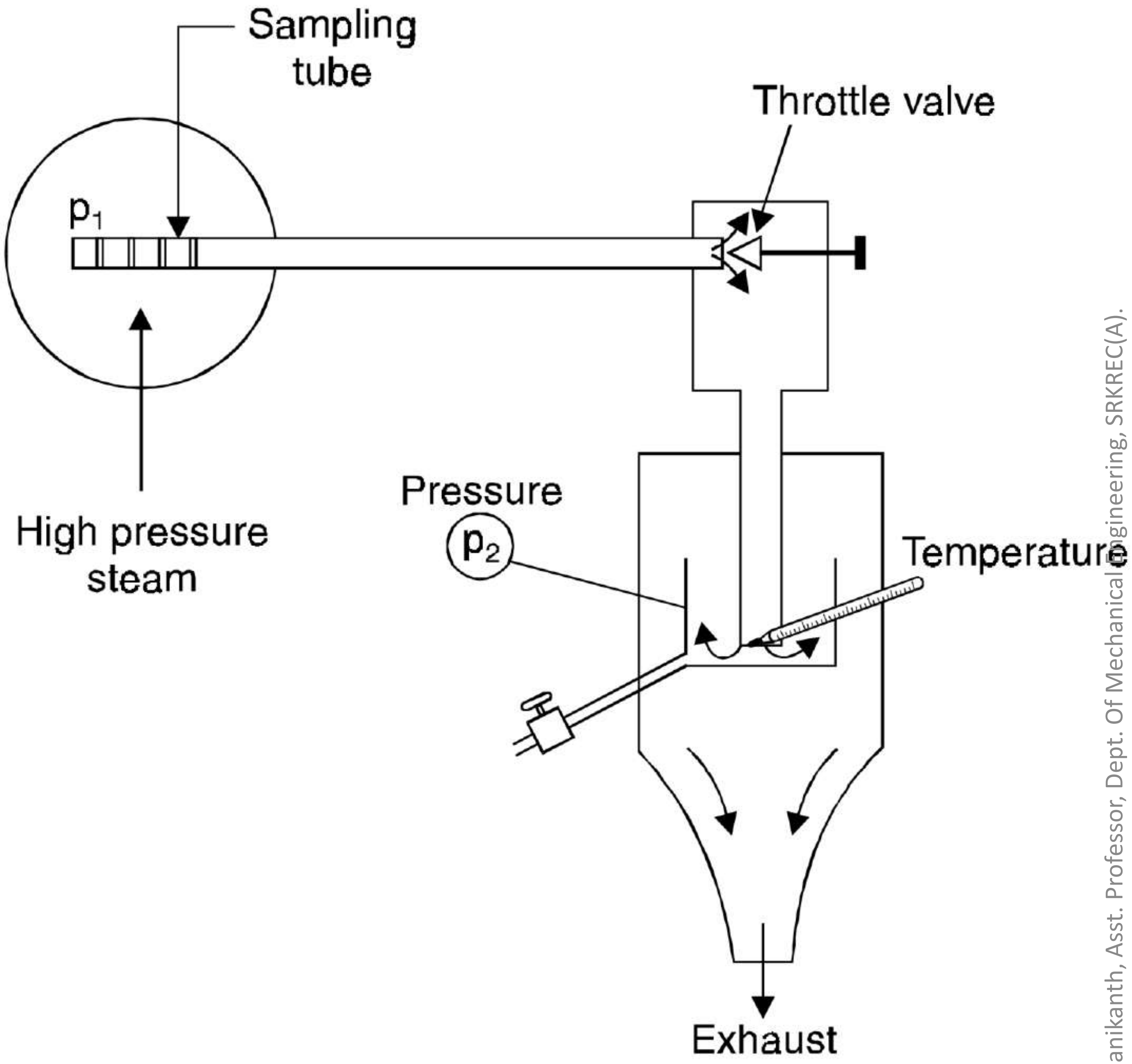


Fig. Throttling calorimeter.

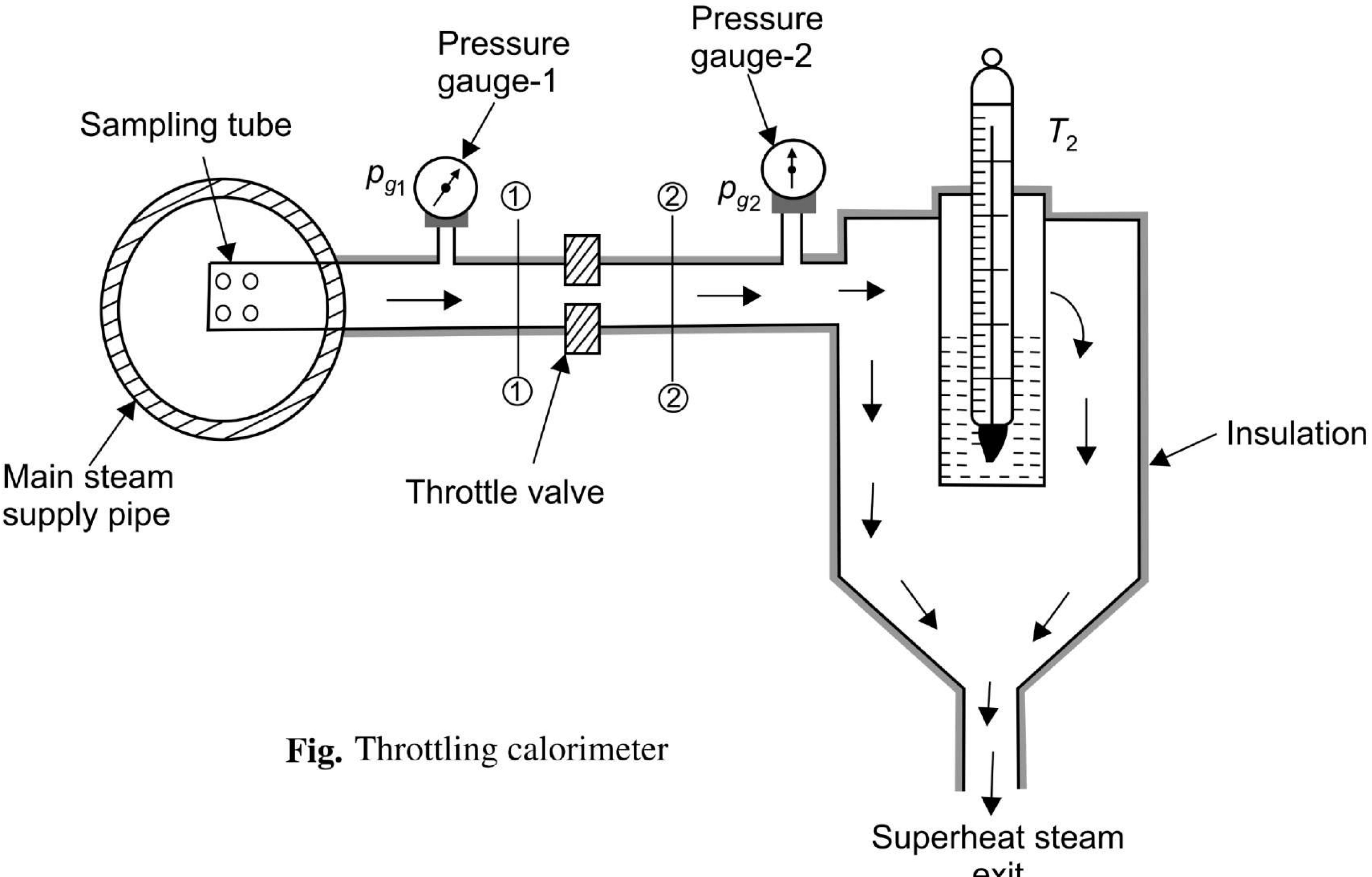


Fig. Throttling calorimeter

The enthalpy, h_2 can then be found and hence

$$h_2 = h_1 = (h_{f_1} + x_1 h_{fg_1}) \text{ at } p_1$$

$$[\text{where } h_2 = h_{f_2} + h_{fg_2} + c_{ps}(T_{sup_2} - T_{s_2})]$$

$$\therefore \quad \boxed{x_1 = \frac{h_2 - h_{f_1}}{h_{fg_1}}}$$

Hence the dryness fraction is determined and state 1 is defined.

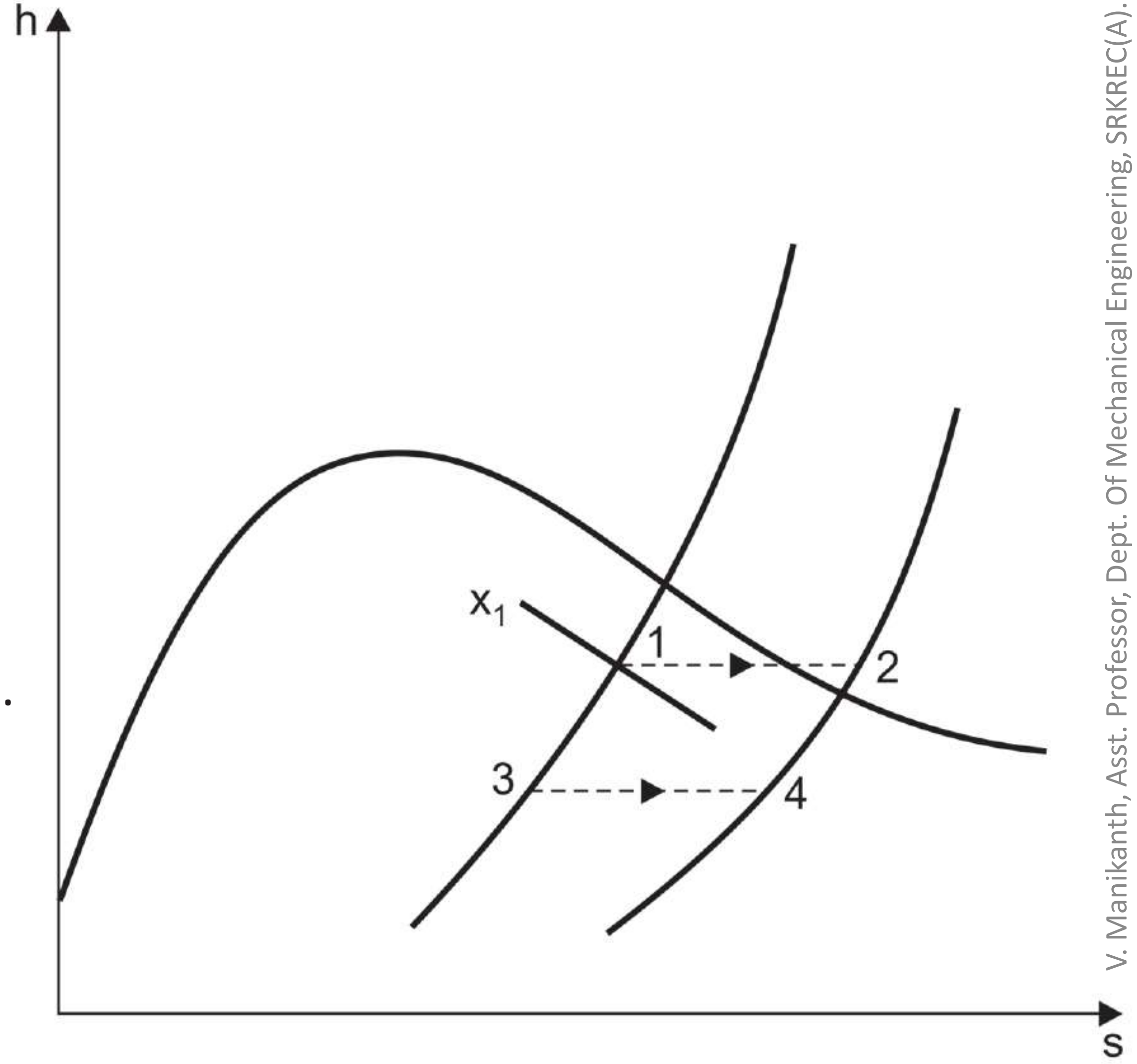


Fig. Throttling process.

Example *A throttling calorimeter is used to measure the dryness fraction of the steam in the steam main which has steam flowing at a pressure of 8 bar. The steam after passing through the calorimeter is at 1 bar pressure and 115°C.*

Calculate the dryness fraction of the steam in the main. Take $c_{ps} = 2.1$ *kJ/kg K.*

Solution. Condition of steam *before throttling* :

$$p_1 = 8 \text{ bar}, x_1 = ?$$

Condition of steam *after throttling* :

$$p_2 = 1 \text{ bar}, \; t_2 = t_{sup_2} = 115°\text{C}$$

As throttling is a *constant enthalpy* process

$$\therefore \quad h_1 = h_2$$

i.e.,

$$h_{f_1} + x_1 h_{gf_1} = h_{f_2} + h_{fg_2} + c_{p_s}(T_{\sup_2} - T_{s_2}) \quad [\because \; T_{sup_2} = 115 + 273 = 388 \text{ K}$$

$$T_{s_2} = 99.6 + 273 = 372.6 \text{ K (at 1 bar)}]$$

$$720.9 + x_1 \times 2046.5 = 417.5 + 2257.9 + 2.1(388 - 372.6)$$

$$720.9 + 2046.5\, x_1 = 2707.7$$

$$\therefore \quad x_1 = \frac{2707.7 - 720.9}{2046.5} = 0.97$$

Hence, **dryness fraction of steam in the main = 0.97. (Ans.)**

Separating and Throttling Calorimeter

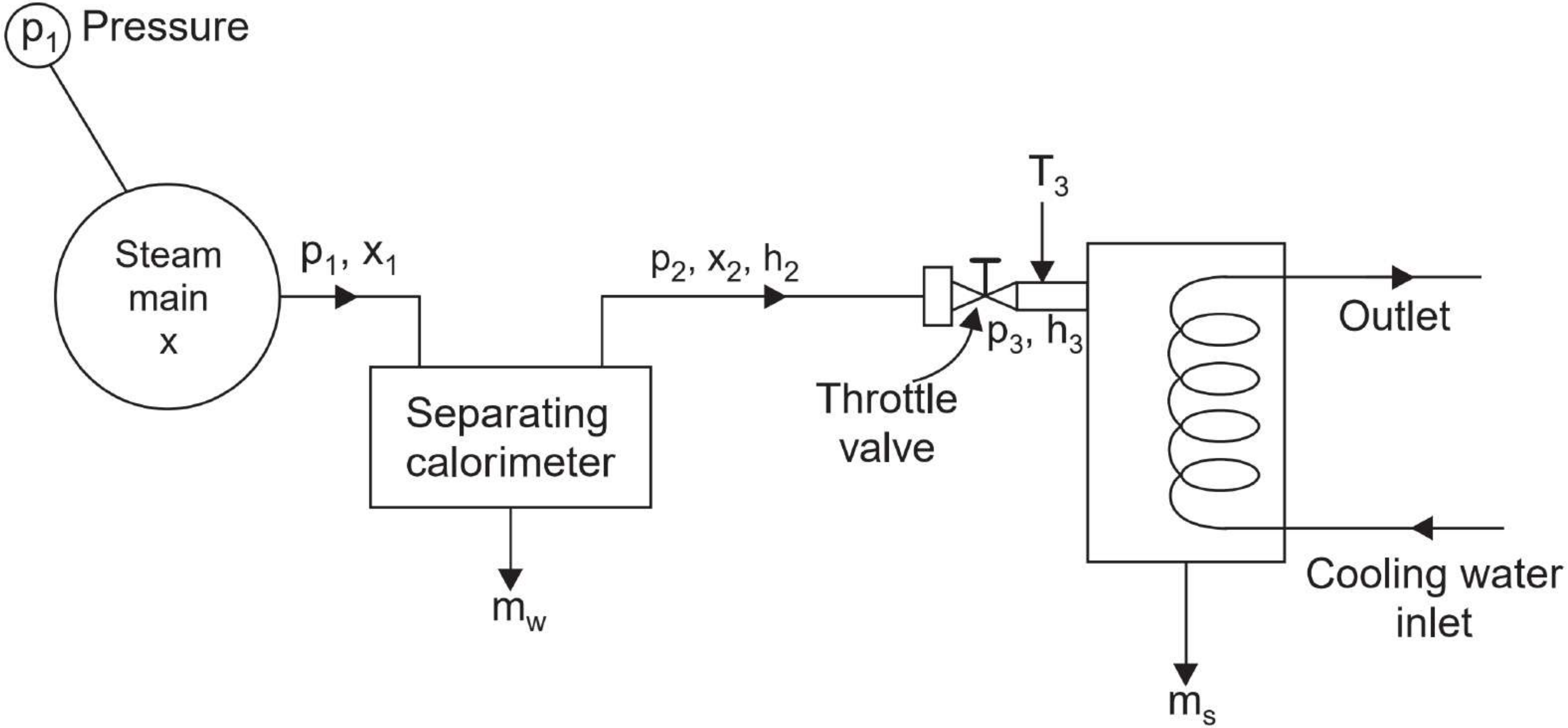


Fig. Separating and throttling calorimeter.

If the steam whose dryness fraction is to be determined is very wet then throttling to atmospheric pressure may not be sufficient to ensure superheated steam at exit. In this case it is necessary to dry the steam partially, before throttling. This is done by passing the steam sample from the main through a separating calorimeter as shown in Fig. The steam is made to change direction suddenly, and the water, being denser than the dry steam is separated out. The quantity of water which is separated out (m_w) is measured at the separator, the steam remaining, which *now has a higher dryness fraction*, is passed through the *throttling calorimeter*. With the combined separating and throttling calorimeter it is *necessary* to condense the steam after throttling and measure the amount of condensate (m_s). If a throttling calorimeter only is sufficient, there is no need to measure condensate, the pressure and temperature measurements at exit being sufficient.

Dryness fraction at 2 is x_2, therefore, the mass of dry steam leaving the separating calorimeter is equal to $x_2 m_s$ and this must be the mass of dry vapour in the sample drawn from the main at state 1.

Hence fraction in main, $x_1 = \frac{\text{Mass of dry vapour}}{\text{Total mass}} = \frac{x_2\, m_s}{m_w + m_s}$.

The dryness fraction, x_2, can be determined as follows :

$$^{*}h_3 = h_2 = h_{f_2} + x_2 h_{fg_2} \text{ at } p_2 \; [^{*}h_3 = h_{f_3} + h_{fg_3} + c_{ps}\,(T_{sup_3} - T_{s_3}) \text{ at pressure } p_3]$$

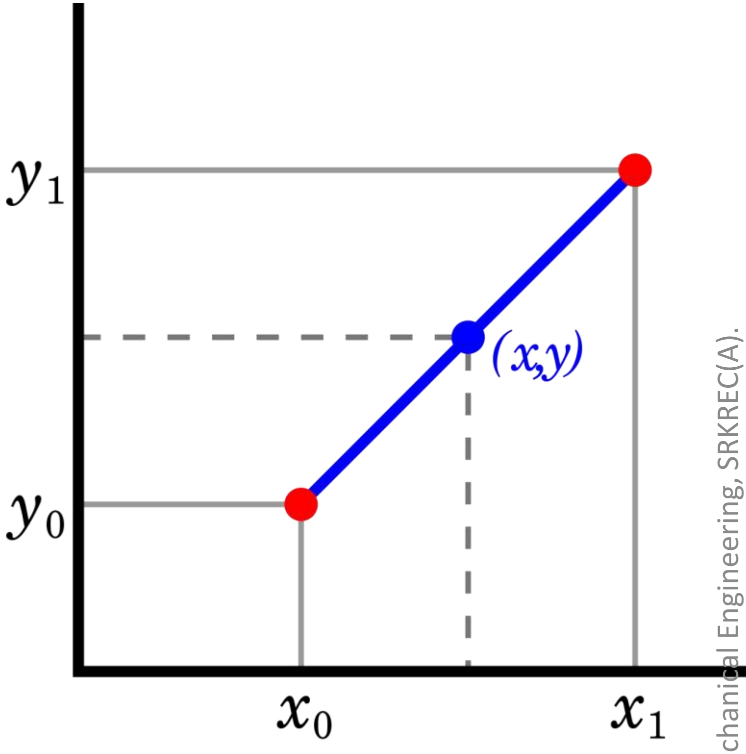
or
$$x_2 = \frac{h_3 - h_{f_2}}{h_{fg_2}}$$

The values of h_{f_2} and h_{fg_2} are read from steam tables at pressure p_2. The pressure in the separator is small so that p_1 is approximately equal to p_2.

Linear interpolation

If the two known points are given by the coordinates *(X0,Y0)* and *(X1,Y1)* , the linear interpolant is the straight line between these points. For a value *X* in the interval , the value *y* along the straight line is given from the equation of slopes.

$$\frac{y - y_0}{x - x_0} = \frac{y_1 - y_0}{x_1 - x_0}$$



Example *The following observations were taken with a separating and a throttling calorimeter arranged in series :*

Water separated = 2 kg, steam discharged from the throttling calorimeter = 20.5 kg, temperature of steam after throttling = 110°C, initial pressure = 12 bar abs., barometer = 760 mm of Hg, final pressure = 5 mm of Hg.

Estimate the quality of steam supplied.

Solution. Quantity of water separated out, $m_w = 2$ kg

Steam (condensate) discharged from the throttling calorimeter, $m_s = 20.5$ kg

Temperature of steam after throttling, $t_{sup} = 110°\text{C}$

Initial pressure of steam, $p_1 = 12$ bar abs.

Final pressure of steam, $p_3 = 760 + 5 = 765$ mm

$$= \frac{765}{1000} \times 1.3366 \qquad (\because \quad 1 \text{ m Hg} = 1.3366 \text{ bar})$$

$$\simeq 1 \text{ bar}$$

From steam tables :

At $p_1 = p_2 =$ **12 bar :** $h_f = 798.4$ kJ/kg, $h_{fg} = 1984.3$ kJ/kg

At $p_3 =$ **1 bar :** $t_s = 99.6$°C, $h_f = 417.5$ kJ/kg, $h_{fg} = 2257.9$ kJ/kg

$t_{sup} = 110$°C (given)

Also
$$h_3 = h_2$$

$$(h_{f_3} + h_{fg_3}) + c_{ps}(T_{sup_3} - T_{s_3}) = hf_2 + x_2 h_{fg_2}$$

Taking $c_{ps} = 2$ kJ/kg K, we get

$$417.5 + 2257.9 + 2[(110 + 273) - (99.6 + 273)] = 798.4 + x_2 \times 1984.3$$

$$2696.2 = 798.4 + 1984.3\, x_2$$

$$\therefore \quad x_2 = \frac{2696.2 - 798.4}{1984.3} = 0.956$$

Now, **quality of steam supplied,**

$$x_1 = \frac{x_2 m_s}{m_w + m_s} = \frac{0.956 \times 20.5}{2 + 20.5}$$

$$= \mathbf{0.87.} \quad \textbf{(Ans.)}$$

☞ **Example** *A vessel having a volume of 0.6 m^3 contains 3.0 kg of liquid water and water vapour mixture in equilibrium at a pressure of 0.5 MPa. Calculate :*

(*i*) *Mass and volume of liquid ;*

(*ii*) *Mass and volume of vapour.*

Solution. Volume of the vessel, $V = 0.6 \text{ m}^3$

Mass of liquid water and water vapour, $m = 3.0$ kg

Pressure, $p = 0.5 \text{ MPa} = 5 \text{ bar}$

Thus, specific volume, $v = \dfrac{V}{m} = \dfrac{0.6}{3.0} = 0.2 \text{ m}^3/\text{kg}$

At 5 bar : From steam tables,

$$v_{fg} = v_g - v_f = 0.375 - 0.00109 = 0.3739 \text{ m}^3/\text{kg}$$

We know that, $v = v_g - (1 - x)\, v_{fg}$, where x = quality of the vapour.

$$0.2 = 0.375 - (1 - x) \times 0.3739$$

$$\therefore \quad (1 - x) = \frac{(0.375 - 0.2)}{0.3739} = 0.468$$

$$x = 0.532$$

(*i*) **Mass and volume of liquid, $\mathbf{m_{liq.}}$ = ? $\mathbf{V_{liq.}}$ = ?**

$$\mathbf{m_{liq.}} = m(1 - x) = 3.0 \times 0.468 = \mathbf{1.404\ kg.} \quad \textbf{(Ans.)}$$

$$\mathbf{V_{liq.}} = m_{liq.}\, v_f = 1.404 \times 0.00109 = \mathbf{0.0015\ m^3.} \quad \textbf{(Ans.)}$$

(*ii*) **Mass and volume of vapour, $\mathbf{m_{vap.}}$ = ? $\mathbf{V_{vap.}}$ = ?**

$$\mathbf{m_{vap.}} = m.x = 3.0 \times 0.532 = \mathbf{1.596\ kg.} \quad \textbf{(Ans.)}$$

$$\mathbf{V_{vap.}} = m_{vap.}\, v_g = 1.596 \times 0.375 = \mathbf{0.5985\ m^3.} \quad \textbf{(Ans.)}$$

Example *A vessel having a capacity of 0.05 m^3 contains a mixture of saturated water and saturated steam at a temperature of 245°C. The mass of the liquid present is 10 kg. Find the following :*

(i) The pressure,	*(ii) The mass,*
(iii) The specific volume,	*(iv) The specific enthalpy,*
(v) The specific entropy, and	*(vi) The specific internal energy.*

Solution. From steam tables, corresponding to 245°C :

$$p_{sat} = 36.5 \text{ bar}, \ v_f = 0.001239 \text{ m}^3/\text{kg}, \ v_g = 0.0546 \text{ m}^3/\text{kg}$$
$$h_f = 1061.4 \text{ kJ/kg}, \ h_{fg} = 1740.2 \text{ kJ/kg}, \ s_f = 2.7474 \text{ kJ/kg K}$$
$$s_{fg} = 3.3585 \text{ kJ/kg K.}$$

(*i*) **The pressure = 36.5 bar** (or 3.65 MPa). **(Ans.)**

(*ii*) **The mass, m :**

Volume of liquid, $V_f = m_f v_f$

$= 10 \times 0.001239 = 0.01239 \text{ m}^3$

Volume of vapour, $V_g = 0.05 - 0.01239 = 0.03761 \text{ m}^3$

∴ Mass of vapour, $m_g = \dfrac{V_g}{v_g} = \dfrac{0.03761}{0.0546} = 0.688 \text{ kg}$

∴ The total mass of mixture,

$$\mathbf{m} = m_f + m_g = 10 + 0.688 = \mathbf{10.688\ kg.} \quad \textbf{(Ans.)}$$

(*iii*) **The specific volume, v :**

Quality of the mixture,

$$x = \frac{m_g}{m_g + m_f} = \frac{0.688}{0.688 + 10} = 0.064$$

∴ $\mathbf{v} = v_f + x v_{fg}$

$= 0.001239 + 0.064 \times (0.0546 - 0.001239) \qquad (\because \quad v_{fg} = v_g - v_f)$

$= \mathbf{0.004654\ m^3/kg.} \quad \textbf{(Ans.)}$

(*iv*) **The specific enthalpy, h :**

$$\mathbf{h} = h_f + xh_{fg}$$

$$= 1061.4 + 0.064 \times 1740.2 = \mathbf{1172.77\ kJ/kg.} \quad \textbf{(Ans.)}$$

(*v*) **The specific entropy, s :**

$$\mathbf{s} = s_f + xs_{fg}$$

$$= 2.7474 + 0.064 \times 3.3585 = \mathbf{2.9623\ kJ/kg\ K.} \quad \textbf{(Ans.)}$$

(*vi*) **The specific internal energy, u :**

$$\mathbf{u} = h - pv$$

$$= 1172.77 - \frac{36.5 \times 10^5 \times 0.004654}{1000} = \mathbf{1155.78\ kJ/kg.} \quad \textbf{(Ans.)}$$

Example 11. *If a certain amount of steam is produced at a pressure of 8 bar and dryness fraction 0.8. Calculate :*

(*i*) *External work done during evaporation.*

(*ii*) *Internal latent heat of steam.*

Solution. Pressure of steam, $p = 8$ bar

Dryness fraction, $x = 0.8$

At 8 bar. From steam tables,

$$v_g = 0.240 \text{ m}^3/\text{kg}, \; h_{fg} = 2046.5 \text{ kJ/kg}$$

(*i*) **External work done during evaporation**

$$= pxv_g = 8 \times 10^5 \times 0.8 \times 0.24 \text{ Nm}$$

$$= \frac{8 \times 10^5 \times 0.8 \times 0.24}{10^3} = \mathbf{153.6 \text{ kJ.}} \quad \textbf{(Ans.)}$$

(*ii*) **Internal latent heat**

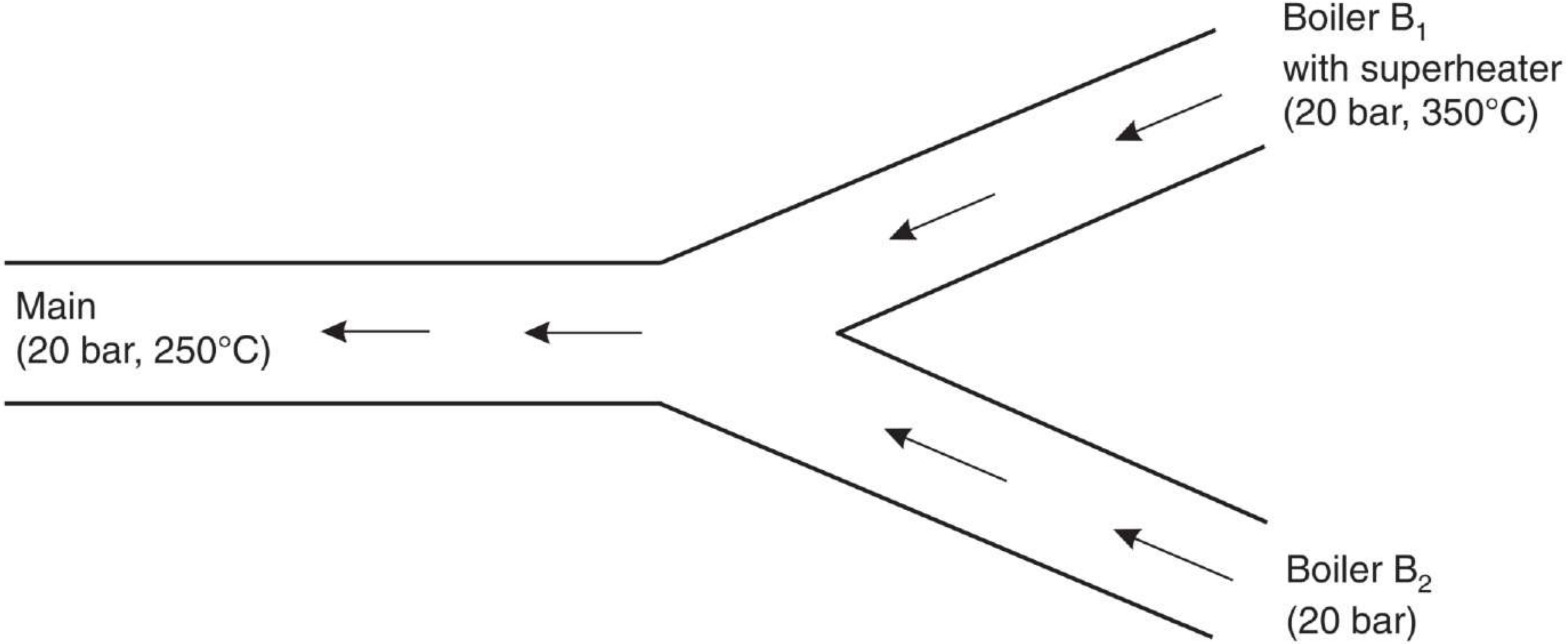
$$= xh_{fg} - \text{external work done}$$

$$= 0.8 \times 2046.5 - 153.6$$

$$= \mathbf{1483.6 \text{ kJ.}} \quad \textbf{(Ans.)}$$

Example . *Two boilers one with superheater and other without superheater are delivering equal quantities of steam into a common main. The pressure in the boilers and main is 20 bar. The temperature of steam from a boiler with a superheater is 350°C and temperature of the steam in the main is 250°C.*

Determine the quality of steam supplied by the other boiler. Take $c_{ps} = 2.25$ *kJ/kg.*



Solution. Boiler B_1. 20 bar, 350°C :

Enthalpy,
$$h_1 = h_{g_1} + c_{ps}(T_{sup} - T_s)$$
$$= 2797.2 + 2.25(350 - 212.4)$$
$$= 3106.8 \text{ kJ/kg} \qquad ...(i)$$

Boiler B_2. 20 bar (temperature not known) :

$$h_2 = h_{f_2} + x_2 h_{fg_2}$$
$$= (908.6 + x_2 \times 1888.6) \text{ kJ/kg} \qquad ...(ii)$$

Main. 20 bar, 250°C.

Total heat of 2 kg of steam in the steam main

$$= 2[h_g + c_{ps}(T_{sup} - T_s)]$$
$$= 2[2797.2 + 2.25(250 - 212.4)] = 5763.6 \text{ kJ} \qquad ...(iii)$$

Equating (*i*) and (*ii*) with (*iii*), we get

$$3106.8 + 908.6 + x_2 \times 1888.6 = 5763.6$$
$$4015.4 + 1888.6x_2 = 5763.6$$

$$\therefore \qquad x_2 = \frac{5763.6 - 4015.4}{1888.6} = 0.925$$

BOILERS

By

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Contents:

Steam Boilers: Working principle of various boilers their accessories and mountings (Simple vertical, Cochran, Babcock & Wilcox and Lancashire Boiler), Performance of boilers (simple problems)

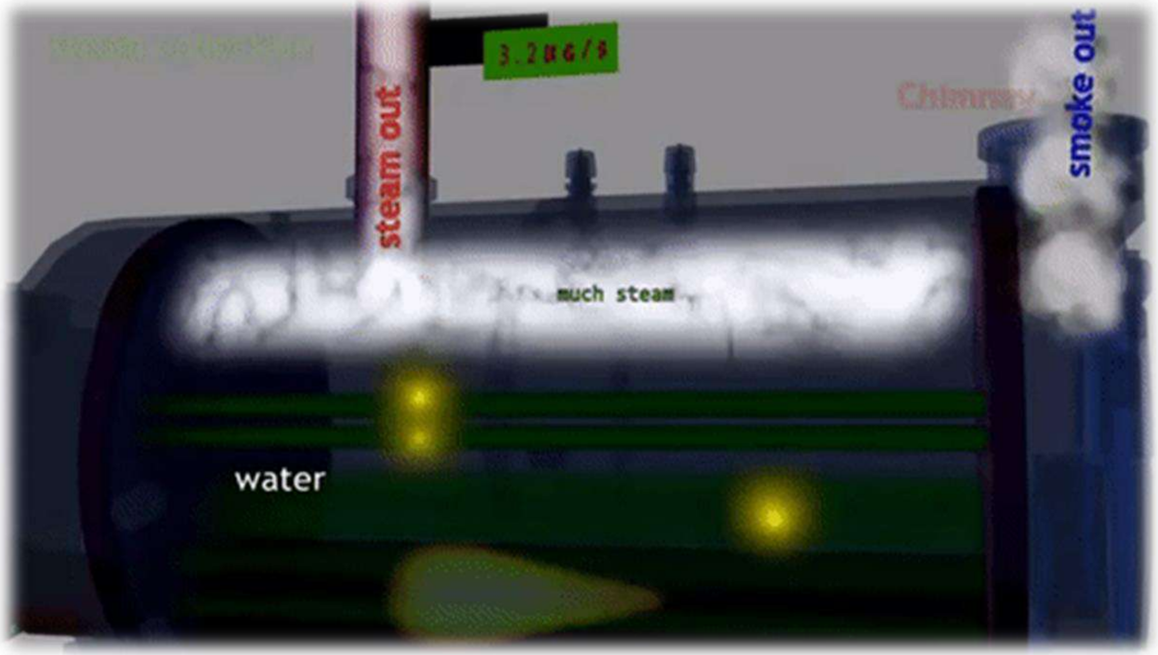
Reference Books:

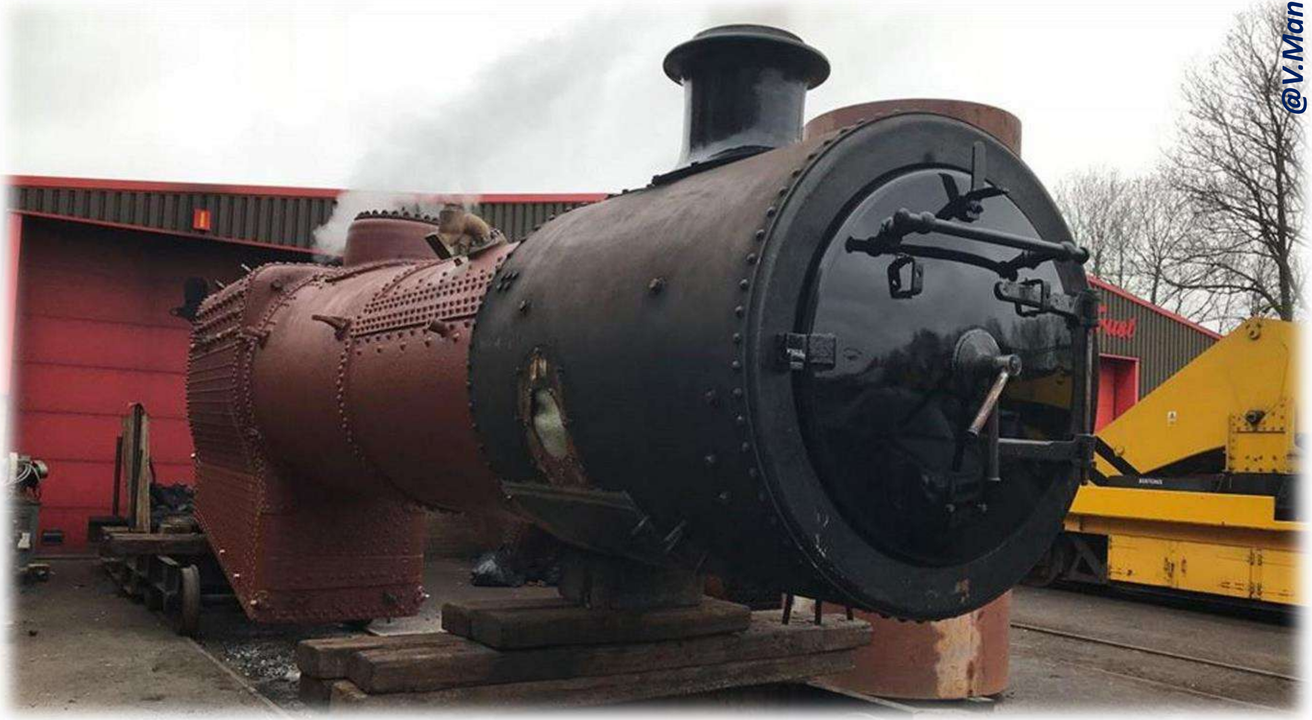
1.Thermal Engineering, by R.K.Rajput

2. Boilers-A Practical Reference,By Kumar Rayaprolu

The **boiler** is basically a large heat exchanger where the heat originating from combustion gases, nuclear reactors, or other sources is transferred to the water essentially at constant pressure. The boiler, together with the section where the steam is superheated (the super-heater), is often called the ***steam generator.***

Definition of Boiler according to ***IBR Act 1923*** (*Indian Boiler Regulation*), “A steam boiler means any closed vessel exceeding 22.75 liters in capacity and which is used expressively for generating steam under pressure.”





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A boiler transfers heat to the incoming water and generates the steam. The system is shown in Fig.

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

For this system, $\Delta Z = 0$ and $\Delta\left(\frac{{C_2}^2}{2}\right) = 0$

$W = 0$ since neither any work is developed nor absorbed.

Applying energy equation to the system

$$h_1 + Q = h_2$$

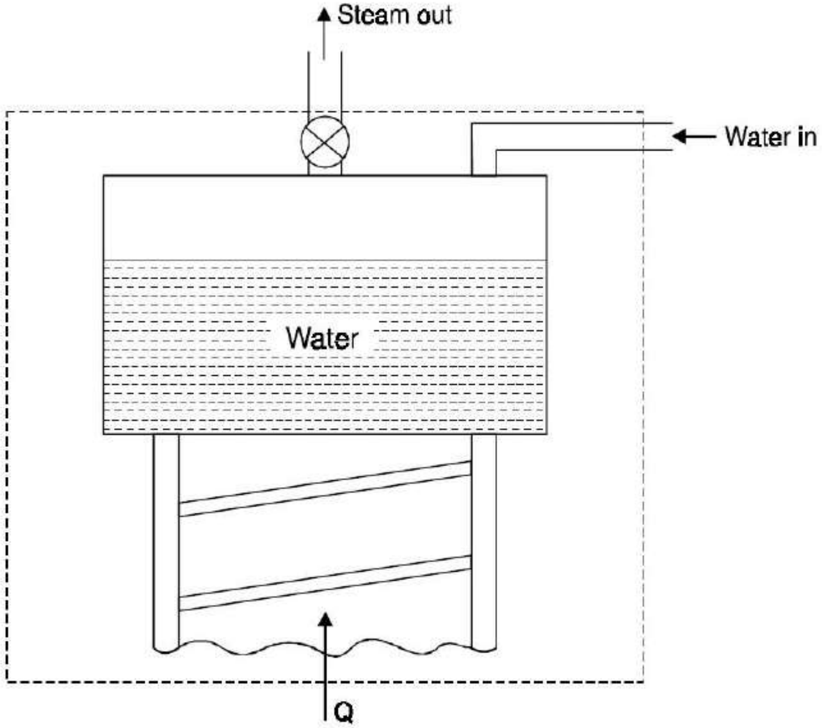


Fig. Boiler.

Applications of Boiler:

1. Operating steam engines.
2. Operating steam turbines.
3. Operating reciprocating pumps.
4. Industrial process work in chemical engineering.
5. For producing hot water required to be supplied to room in very cold areas.
6. In thermal power stations.
7. The heat content of the steam is large and thus it is suitable for process heating in many industries like sugar mills, textile mills, dairy industry and also in chemical industries.

CLASSIFICATION OF BOILERS

The boilers may be classified as follows :

1. Horizontal, Vertical or Inclined

If the axis of the boiler is horizontal, the boiler is called as *horizontal*, if the axis is vertical, it is called *vertical* boiler and if the axis is inclined it is known as *inclined boiler*. *The parts of a horizontal boiler can be inspected and repaired easily but it occupies more space. The vertical boiler occupies less floor area.*

2. Fire Tube and Water Tube

In the fire tube boilers, the hot gases are inside the tubes and the water surrounds the tubes. Examples : *Cochran, Lancashire* and *Locomotive boilers*.

In the water tube boilers, the water is inside the tubes and hot gases surround them. Examples : *Babcock* and *Wilcox, Stirling, Yarrow boiler etc.*

3. Externally Fired and Internally Fired

The boiler is known as externally fired if the fire is outside the shell. Examples : *Babcock and Wilcox boiler, Stirling boiler etc*.

In case of internally fired boilers, the furnace is located inside the boiler shell. Examples : *Cochran, Lancashire boiler etc*.

4. Forced Circulation and Natural Circulation

In *forced circulation* type of boilers, the circulation of water is done *by a forced pump*. Examples : *Velox, Lamont, Benson boiler etc*.

In *natural circulation* type of boilers, circulation of water in the boiler takes place due to *natural convention* currents produced by the application of heat. Examples : *Lancashire, Babcock* and *Wilcox boiler* etc.

5. High Pressure and Low Pressure Boilers

The boilers which produce steam at *pressures of 80 bar and above* are called *high pressure boilers*. Examples : *Babcock* and *Wilcox, Velox, Lamont, Benson boilers*.

The boilers which produce steam at pressure *below 80 bar* are called low *pressure boilers*. Examples : *Cochran, Cornish, Lancashire and Locomotive boilers*.

6. Stationary and Portable

Primarily, the boilers are classified as either stationary (*land*) or mobile (*marine* and *locomotive*).

— Stationary boilers are used for power plant-steam, for central station utility power plants, for plant process steam etc.

— Mobile boilers or portable boilers include locomotive type, and other small units for temporary use at sites (just as in small coal-field pits).

7. Single Tube and Multi-tube Boilers

The fire tube boilers are classified as single tube and multi-tube boilers, depending upon whether the fire tube is one or more than one. The examples of the former type are cornish, simple vertical boiler and rest of the boilers are multi-tube boilers.

SELECTION OF A BOILER

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

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

SIMPLE VERTICAL BOILER:

Simple vertical boiler shown in Fig. has a vertical boiler shell of cylindrical shape. It has fire box of cylindrical type inside the shell. Vertical passage of tubular type called *uptake* is provided over fire box for exhaust of flue gases. Cross tubes are provided for improving water circulation and increasing heating surface. At the bottom of fire box a fire grate is provided for burning fuel. Total heating surface area is about 7–10 times grate area. Man hole and hand holes are provided in the shell for access to inside of shell. Hot gases raising from fire grate go upwardly and heat the water contained in shell and tubes. Steam generated in shell can be tapped through a steam stop valve placed on the crown of shell. Such boilers have steam generation capacity up to 1000 kg per hour and maximum steam pressure up to 10 bar. Size of the boiler ranges from 0.6 m diameter to 2 m diameter and height from 1.2 m to 4 m high. Boiler efficiency is nearly 50%

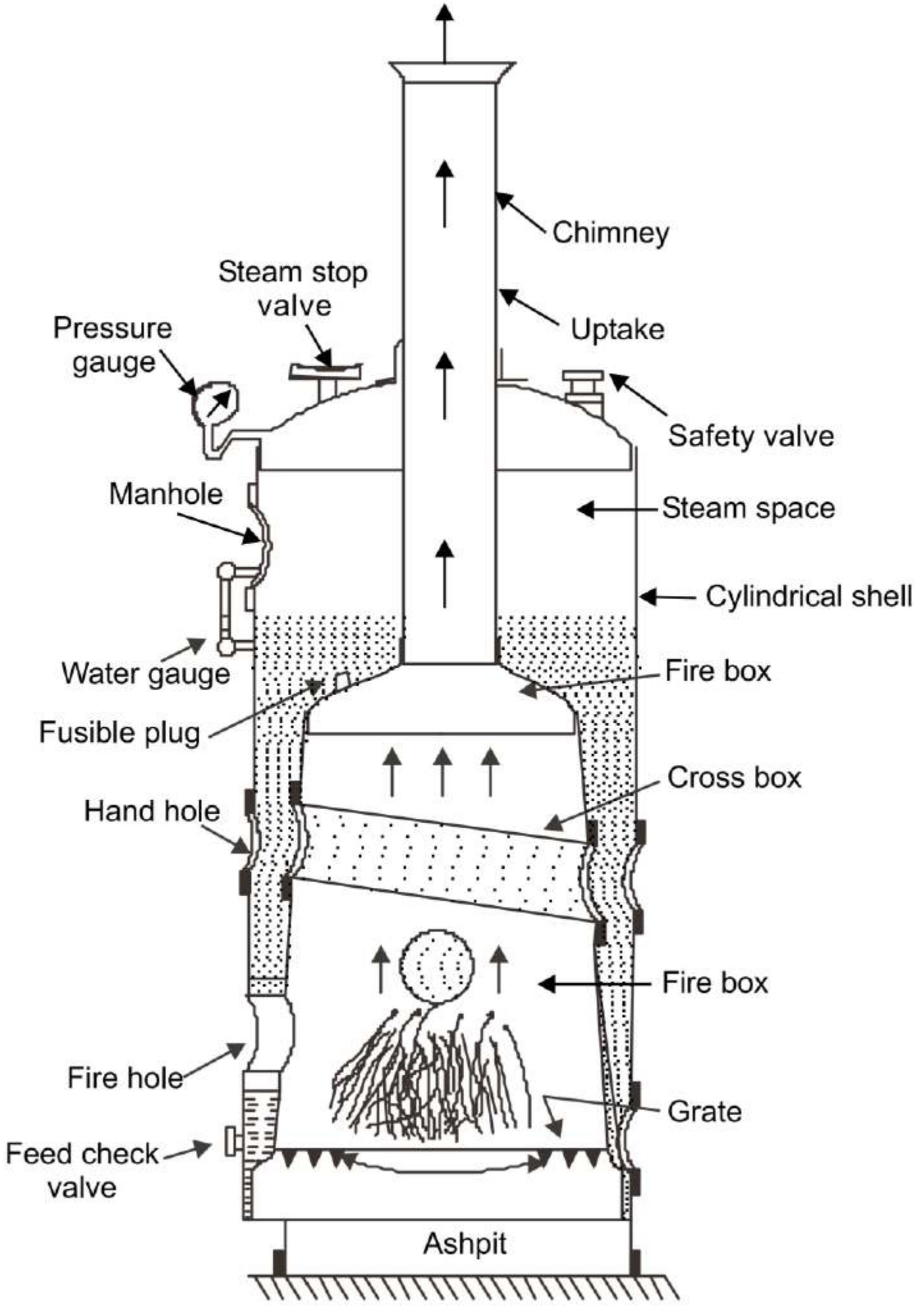


Fig. *Simple vertical boiler*

COCHRAN BOILER:

This is a fire tube boiler of vertical type and came up as a modification over the simple vertical boiler in order to maximize heating surface. Total heating surface area is 10–25 times the grate area. It has cylindrical shell with hemispherical crown. Hemispherical geometry offers maximum volume space for given mass of material and is also very good for strength and maximization of radiant heat absorption. Figure. shows the schematic of Cochran boiler with various mountings upon it. Fire box is also of hemispherical form. Flue gases flow from fire box to refractory material lined combustion chamber through a flue pipe. Incomplete combustion if any can get completed in combustion chamber and hot gases subsequently enter into tubes. After coming out of fire tubes hot gases enter into smoke box having chimney upon it. As the fire box is separately located so any type of fuel such as wood, paddy husk, oil fuel etc. can be easily burnt. These boilers are capable of generating steam up to pressure of 20 bar and steam generating capacity from 20 kg/hr to 3000 kg/hr. Boilers have dimensions ranging from 1 m diameter and 2 m height to 3 m diameter and 6 m height. Efficiency of such boilers ranges between 70 and 75%.

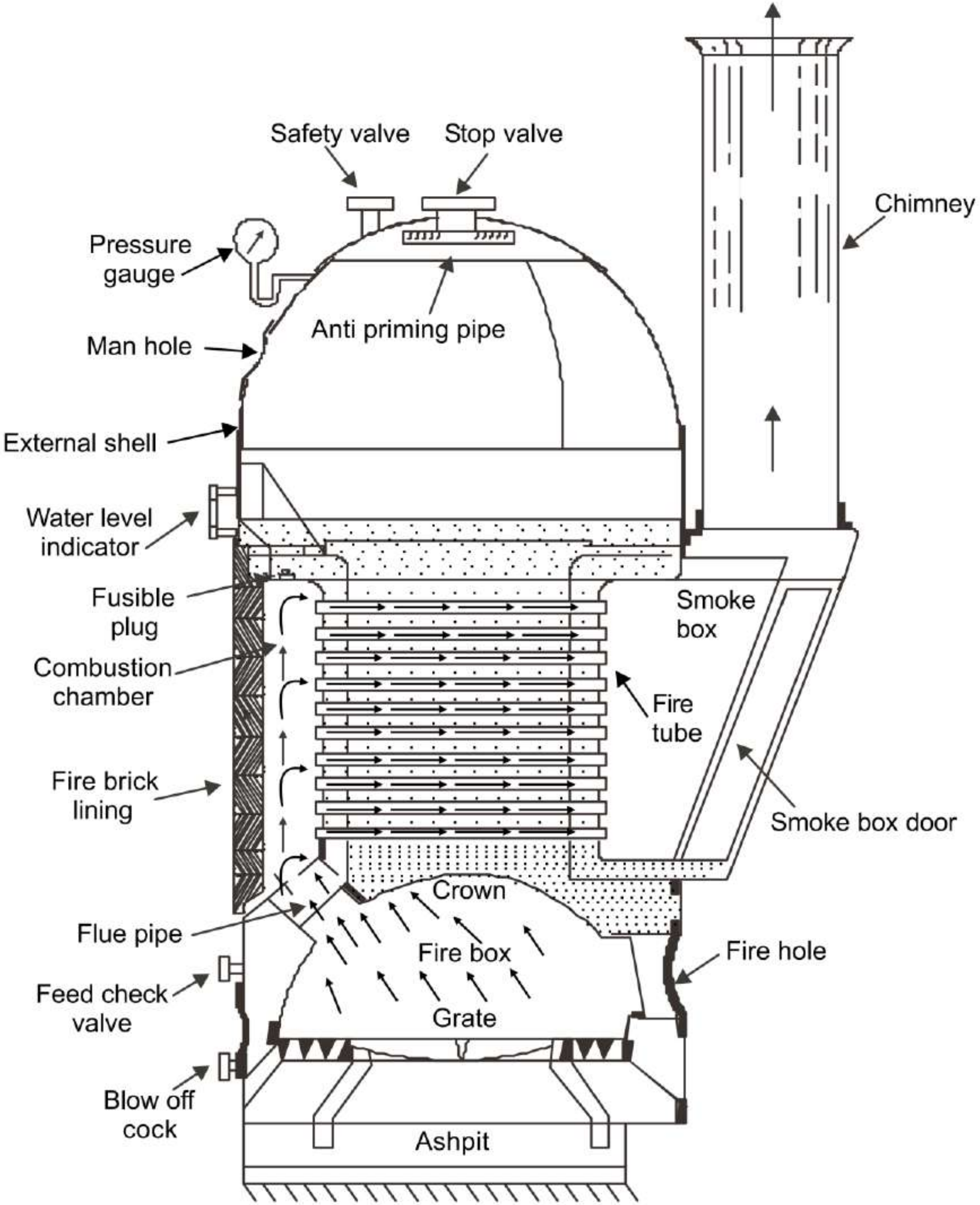


Fig. *Cochran boiler*

Lancashire boiler

It is a horizontal fire tube boiler. This boiler was invented in the year 1844, by William Fairbairn. Boiler is mounted on a brickwork setting with front end of shell sloping about 1 : 250 for emptying the shell. It has a circular shell connected to end plates supported by gusset plates. Two fire tubes run throughout the length of the boiler. Fire tubes are of diameter less than half the diameter of shell and diameter of fire tubes is reduced as shown to have access to lower side of boiler. Fire bridge is provided to prevent fuel from falling over the end of furnace. Fire bridge also helps in producing a better mixture of air and gases for perfect combustion by partly enveloping the combustion space. Hot gases start from grate area, enter fire tubes and come out at back of boiler from where these gases flow towards the front of boiler through bottom flue. Upon reaching the front these hot gases flow through the side flues and enter the main outlet. Outlet passage may also be used commonly by more than one boilers. About 85% of actual heat transferred is transferred through surface of fire tubes while 15% is transferred through bottom and side flues.

Plan, elevation and side views of Lancashire boiler shown in figure explain the furnace, different firetubes, bottom flues, side flues etc. Dampers are provided at the end of side flues for regulating the pressure difference (draught) for exit of burnt gases. Other mountings and accessories are shown in the elevation of Lancashire boiler.

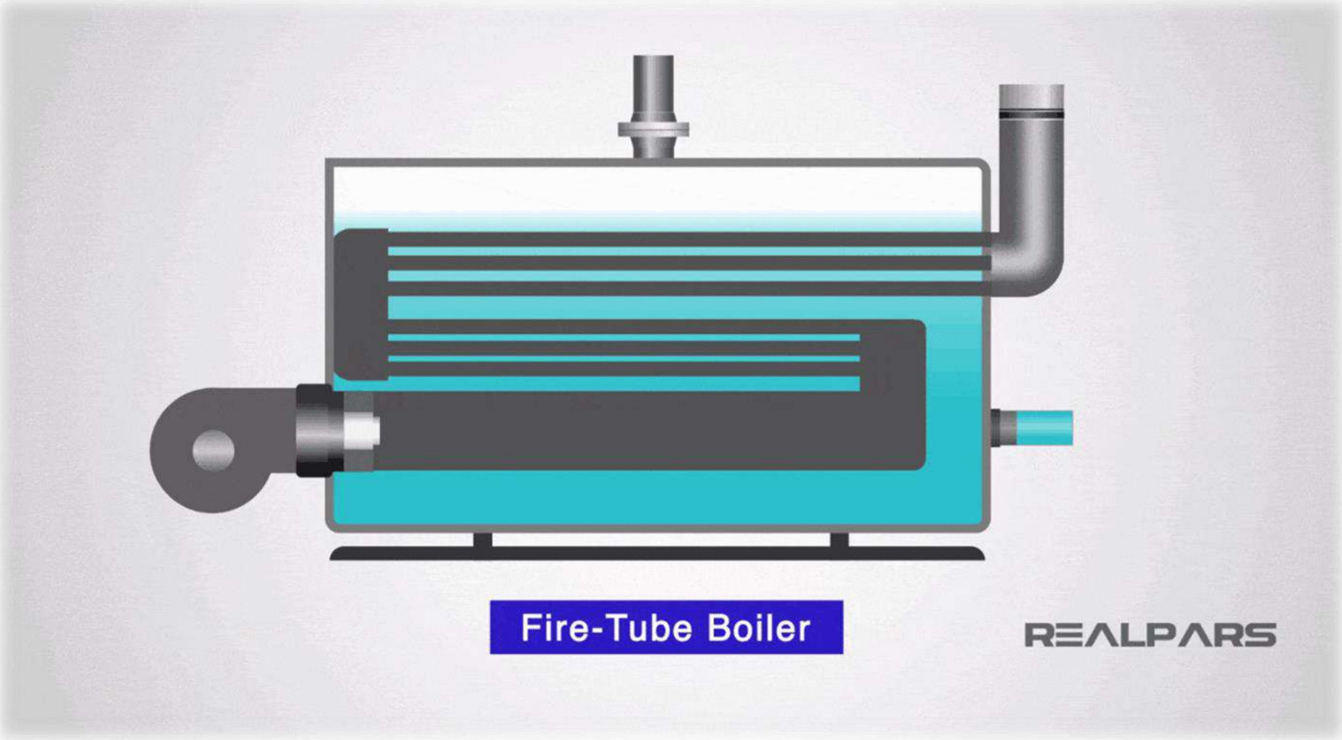


Fig: Lancashire boiler

Parts of a Lancashire Boiler:

A Lancashire Boiler consist of following parts:

- *Safety valve*
- *Pressure Gauge*
- *Feed check valve*
- *Water Level Indicator*
- *Blow off valve*
- *Steam stop valve*
- *Manhole*
- *Fire door*
- *Fusible Plug*
- *Ash pit*
- *Economizer*
- *Air preheater*
- *Superheater*

Safety valve:

The safety valve is used to blow off the steam when the pressure of the steam inside the boiler exceeds the working pressure.

Pressure gauge:

The function of the pressure gauge is to indicate the pressure of the steam inside the boiler.

Feed check valve:

It stops and allows the flow of water inside the boiler.

Water level indicator:

It indicates the level of water in the boiler. It is placed in front of the boiler. Two water level indicators are used in the boiler.

Blow off Valve:

Its function is to remove the sediments or mud periodically that is collected at the bottom of the boiler.

Steam stop valve:

The function of a steam stop valve is to stop and allows the flow of steam from the boiler to the steam

Manhole:

The hole is provided on the boiler so that a man can easily enter inside the boiler for the cleaning and repairing purpose.

Fire door:

This is used to ignite the present fuel inside or outside the boiler.

Fusible plug:

It is used to extinguish the fire inside the boiler when the water level inside the boiler falls to an unsafe level and prevent an explosion. It also prevents the damage that may happen due to the explosion.

Ash pit:

The Ash-pit used to collect the ash of the burnt fuel.

The other various accessories that are also used in Lancashire boiler are:

Economizer:

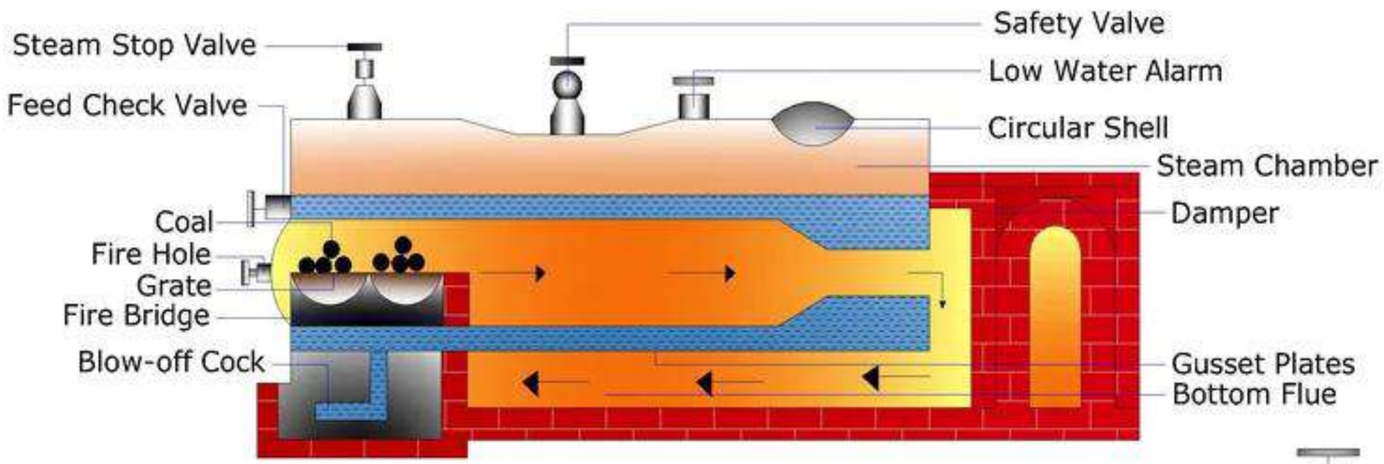
An economizer is a mechanical device that is used as a heat exchanger in the steam power plant. It is used to pre-heat the fluid or water by taking the residual heat from the combustion products(flue gases). It is installed to increase the efficiency of the boiler.

Air pre-heater:

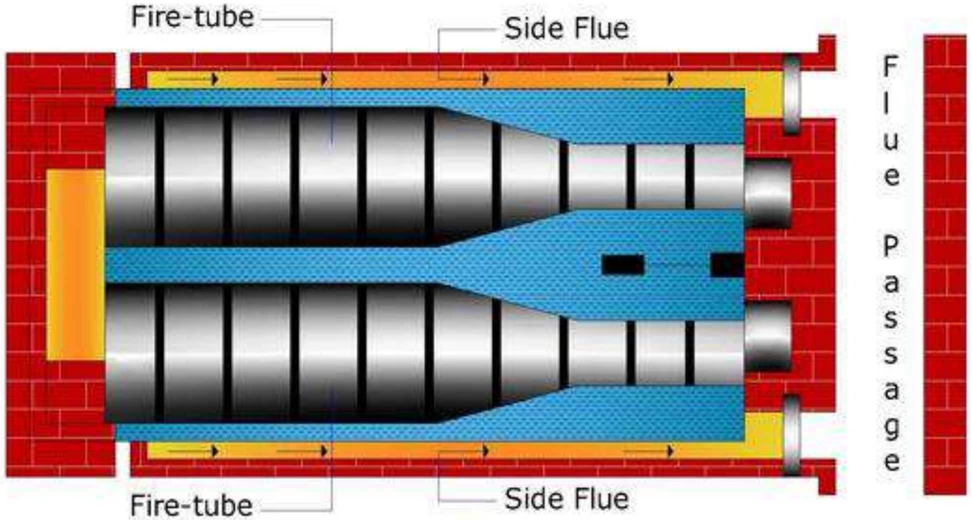
Air preheater is also a mechanical device that abstracts the heat from the flue gases and transfers it to the air(atmosphere).

Superheater:

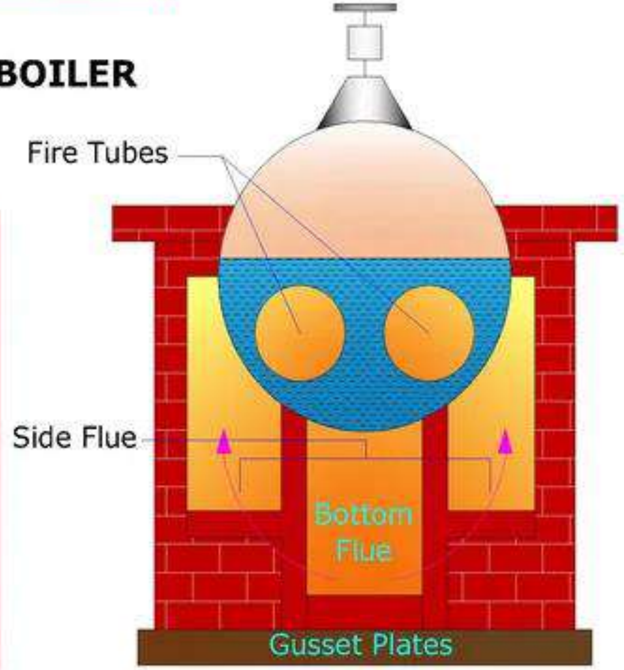
The main purpose of the superheater is to increase the temperature of the saturated steam without any change in the pressure.



FRONT VIEW OF LANCASHIRE BOILER



TOP VIEW OF LANCASHIRE BOILER



SIDE VIEW OF LANCASHIRE BOILER

LANCASHIRE BOILER

Working Principle of Lancashire Boiler:

- This Lancashire boiler works on the basic principle of the heat exchanger.
- It is basically a shell and tube type heat exchanger in which the flue gases flow through the tubes and the water flows through a shell.
- The heat is transfer from flue gases to the water through convection.
- It is a natural circulation boiler that uses the natural current to flow the water inside the boiler.

Applications of Lancashire Boiler:

Lancashire Boiler can be used in several filed like:

- *The Lancashire boiler is used to drive steam turbines, locomotives, marines, etc.*
- *It is used in industries like paper industries, textile industries, sugar industries, tire industries, etc.*

Advantages of Lancashire Boiler:

These are some advantages of Lancashire boiler:

- *This has high thermal efficiency. Thermal efficiency is about 80 to 90%.*
- *This is easy to operate.*
- *It can easily meet the load requirement.*
- *Easy to maintain.*
- *Low consumption of electricity due to natural circulation.*

Disadvantages of Lancashire boiler:

Although a Lancashire boiler has some disadvantages and those are:

- *This is a low-pressure type boiler, so high-pressure steam is not produced.*
- *It has a limited grate area due to the small diameter of the flue tubes.*
- *The steam production rate is low. It is about 9000 kg/hr.*
- *Corrosion occurs in the water legs.*
- *This boiler requires more floor space.*

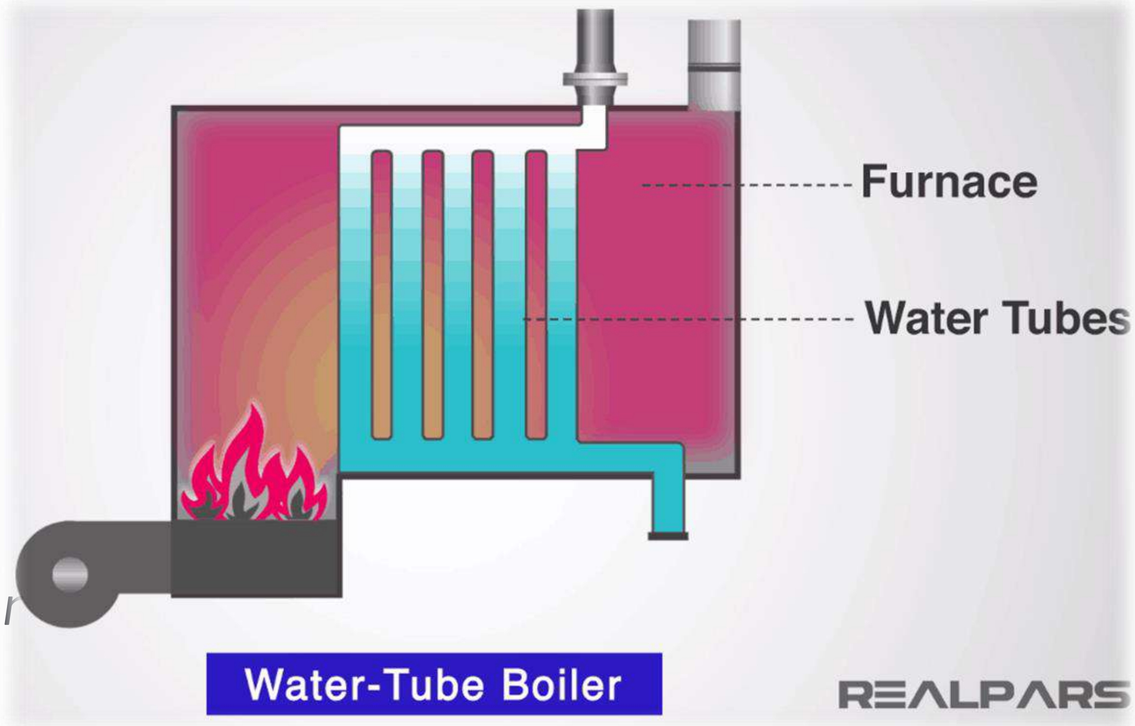
Babcock & Wilcox boiler

Babcock and Wilcox Boiler was discovered by George Herman Babcock and Stephen Wilcox in the year 1967. This is a *water tube boiler*, used in steam power plants. In this type of boiler, water is circulated inside the tubes and hot gases flow over the tubes. This is a Horizontal drum axis, natural draft, natural circulation, multitubular, stationary, high pressure, solid fuel fired, externally fired Water tube boiler.

Babcock and Wilcox Boiler Parts:

A Babcock and Willcox Boiler Parts or Construction consists of:

- *Drum*
- *Water Tubes*
- *Uptake and Downtake header*
- *Grate*
- *furnace*
- *Baffles*
- *Superheater*
- *Mud box*
- *Inspection Door*
- *Water Level Indicator*
- *Pressure Gauge*



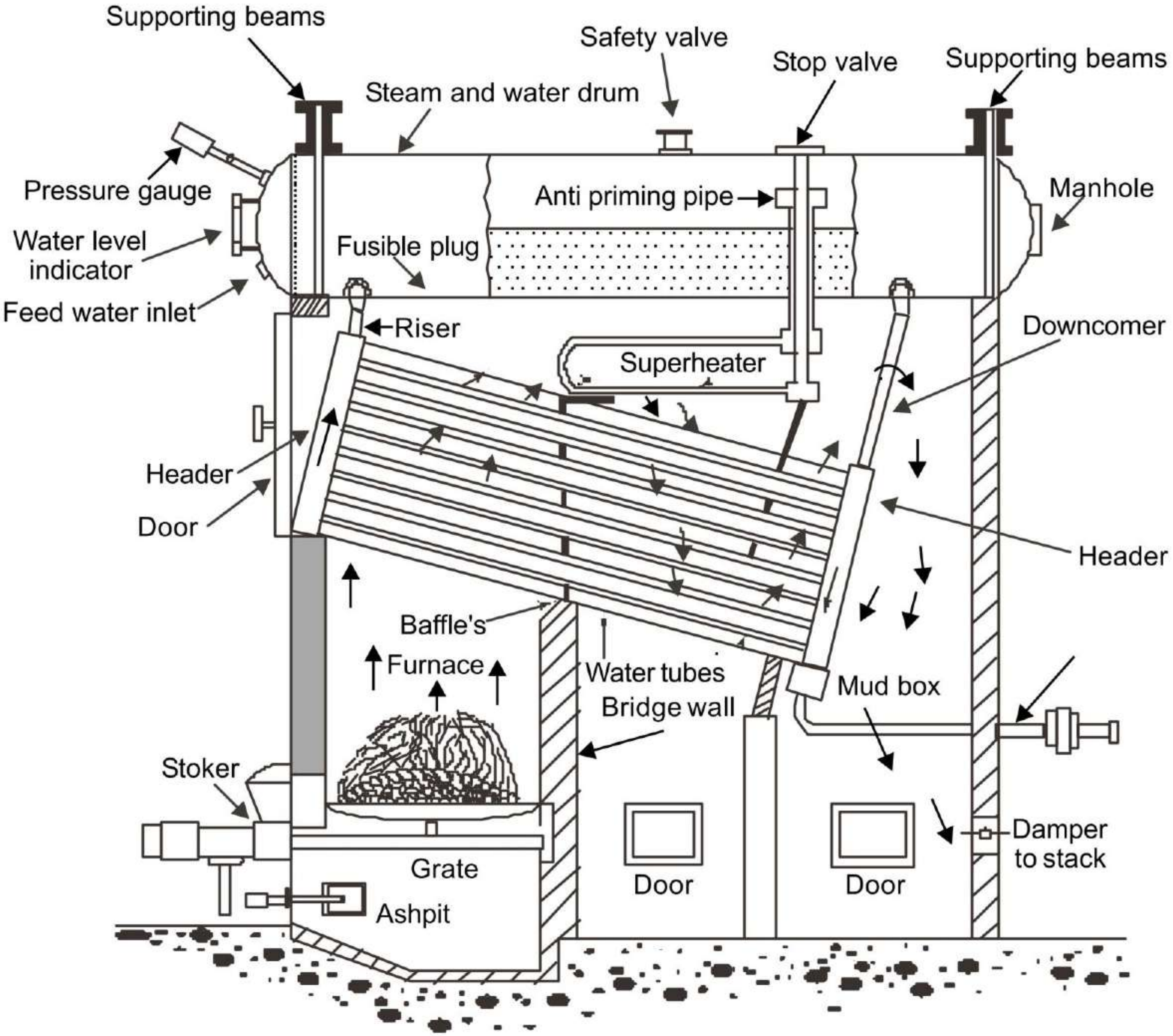


Fig. *Babcock and Wilcox boiler*

@V.Manikanth, Asst. Prof., Dept. of Mechanical, SRKREC(A).

Drum:

This is a horizontal axis drum which contains water and steam.

Water tubes:

Water tubes are placed between the drum and furnace in an inclined position (at an angle of 10 to 15 degrees) to promote water circulation.

Uptake and Downtake Header:

This is present at the front end of the boiler and connected to the front end of the drum. It transports the steam from the water tubes to the drum and this is present at the rear end of the boiler and connects the water tubes to the rear end of the drum. It receives water from the drum.

Grate:

Coal is fed to the grate through the fire door.

Furnace:

The furnace is kept below the uptake-header.

Baffles:

The fire-brick baffles, two in number, are provided to deflect the hot flue gases.

Superheater:

It increases the temperature of saturated steam to the required temperature before discharging it from the steam stop valve.

Mud Box:

This is used to collect the mud present in the water. Mud box is provided at the bottom end of the down-take header.

Inspection Door:

Inspection doors are provided for cleaning and inspection of the boiler.

Water Level Indicator:

The water level indicator shows the level of water within the drum.

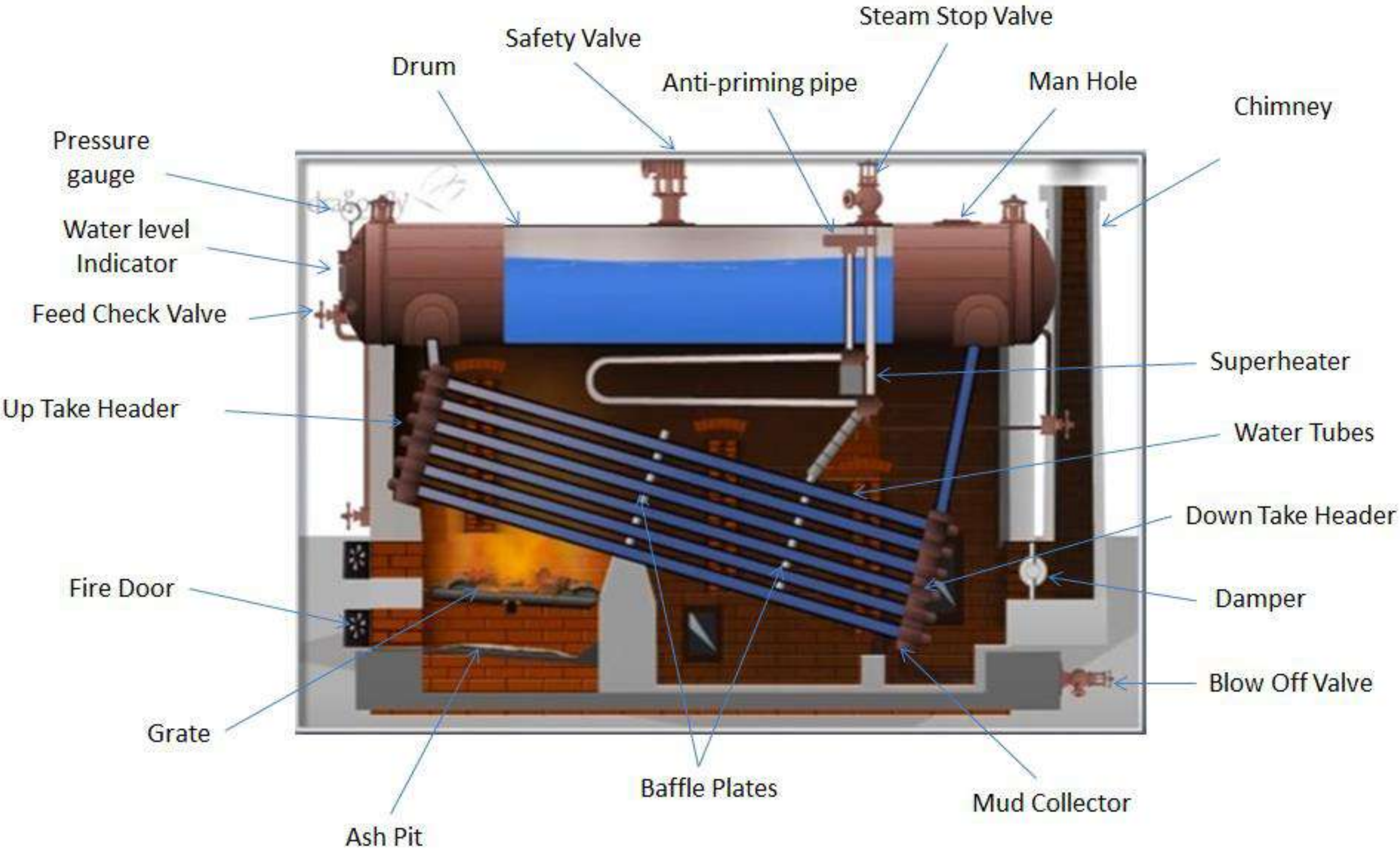


Fig: Babcock & Wilcox boiler

Working Principle of Babcock and Wilcox Boiler:

The working of Babcock and Wilcox boiler is first the water starts to come in the water tubes from the drum through down take header with the help of a boiler feed pump which continues to feed the water against the drum pressure. The water present in the inclined water tubes gets heated up by the hot flue gases produced by the burning of coal on the fire grate. These fuel gases are uniformly heated the water tube with the help of a baffle plate which works deflect the flues gas uniform throughout the tubes which absorbed the heating maximum from the flue gases.

Continuous circulation of water from the drum to the water tubes and water tubes to the drum is thus maintained. The circulation of water is maintained by convective current and it’s known as Natural Circulation. The Steam generated is moved upward, due to density difference and through the up-take header, it gets collected at the upper side in the boiler drum. Anti-priming pipe inside the drum which works separates the moisture from the steam and sends it’s to the superheater. The superheater receives the water-free steam from an anti-priming pipe.

It increases the temperature of the steam to the desired level and transfers it to the main steam stop valve of the boiler. The superheated steam stop valve is either collected in a steam drum or send it’s inside the steam turbine for electricity generation.

Applications Babcock and Wilcox Boiler:

The main application Babcock and Wilcox boiler to produce ***high-pressure steam*** in power generation industries.

Advantages of Babcock and Wilcox:

- The overall efficiency of this boiler is high.
- The steam generation rate is higher about 20 ton per hour at pressure 10 to 20 bars.
- The tubes can be replaced easily.
- The boiler can expand and contract freely.
- It is easy to repair maintenance and cleaning.

Disadvantages of Babcock and Wilcox Boiler:

1. It is less suitable for impure and sedimentary water, as a small deposit of scale may cause the overheating and bursting of tubes. Hence, water treatment is very essential for water tube boilers.
2. Failure in feed water supply even for a short period is liable to make the boiler overheated. Hence the water level must be watched very carefully during the operation of a water tube boiler.
3. The maintenance cost is high.

S.No.	Particulars	Fire-tube boilers	Water-tube boilers
1.	*Position of water and hot gases*	Hot gases inside the tubes and water outside the tubes.	Water inside the tubes and hot gases outside the tubes.
2.	*Mode of firing*	Generally internally fired.	Externally fired.
3.	*Operating pressure*	Operating pressure limited to 16 bar.	Can work under as high pressure as 100 bar.
4.	*Rate of steam production*	Lower	Higher.
5.	*Suitability*	Not suitable for large power plants.	Suitable for large power plants.
6.	*Risk on bursting*	Involves lesser risk on explosion due to lower pressure.	Involves more risk on bursting due to high pressure.
7.	*Floor area*	For a given power it occupies more floor area.	For a given power it occupies less floor-area.
8.	*Construction*	Difficult	Simple
9.	*Transportation*	Difficult	Simple

[Continue..]

S.No.	***Particulars***	***Fire-tube boilers***	***Water-tube boilers***
10.	*Shell diameter*	Large for same power	Small for same power
11.	*Chances of explosion*	Less	More
12.	Treatment of water	Not so necessary	More necessary
13.	*Accessibility of various parts*	Various parts not so easily accessible for cleaning, repair and inspection.	Various parts are more accessible.
14.	*Requirement of skill*	Require less skill for efficient and economic working.	Require more skill and careful attention.

BOILER MOUNTINGS AND ACCESSORIES:

Boiler mountings and accessories have been defined earlier and shown on the different boilers.

Different mountings are:

(i) Water level indicator

(ii) Safety valves

(iii) High steam and low water safety valves

(iv) Fusible plug

(v) Pressure gauge

(vi) Stop valve

(vii) Feed check valve

(viii) Blow off cock

(ix) Manhole and mud box

***Various boiler accessories are*:**

(i) Superheater

(ii) Economizer

(iii) Air preheater

(iv) Feed pump

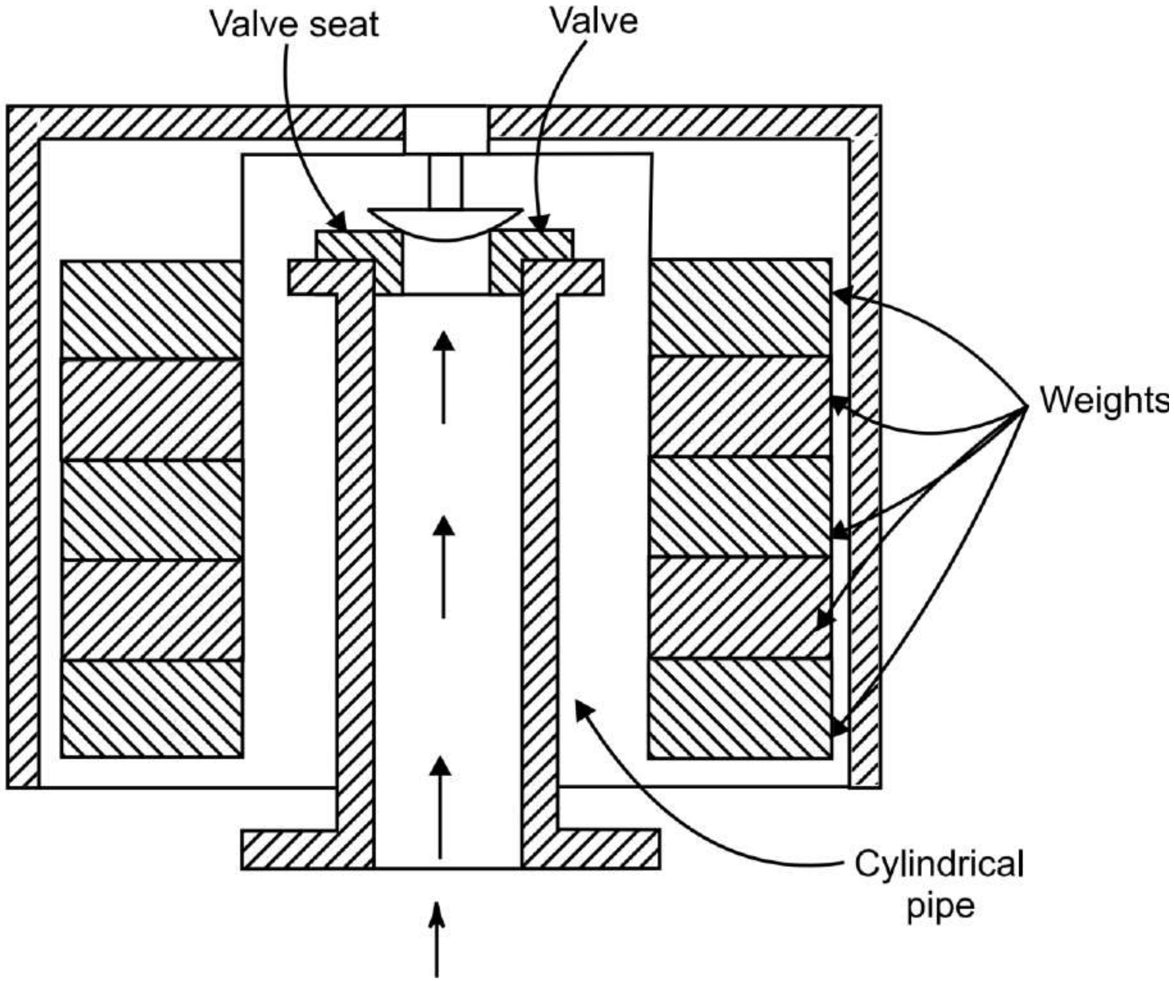


Fig. *Dead weight safety valve*

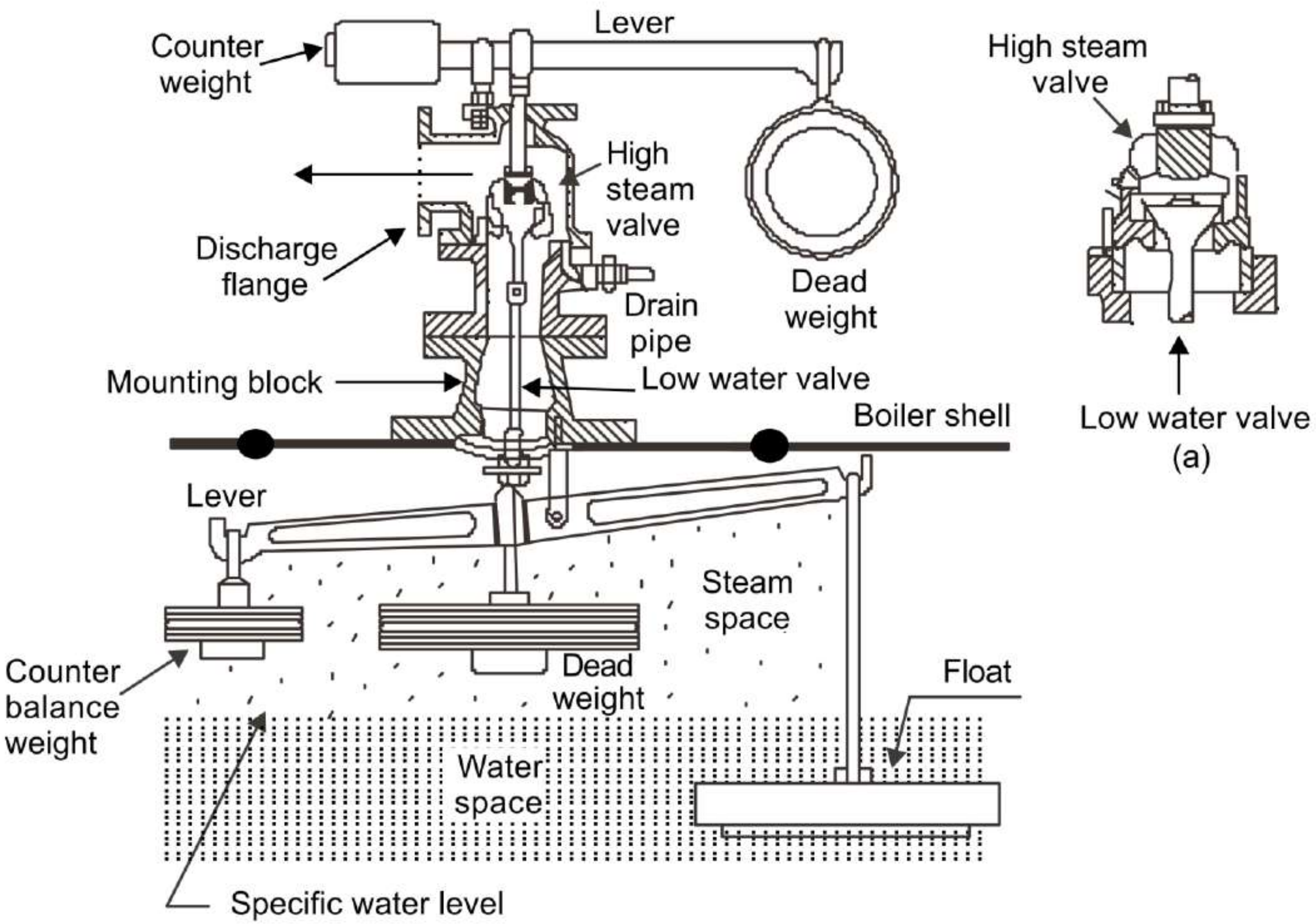


Fig. *High steam and low water safety valve*

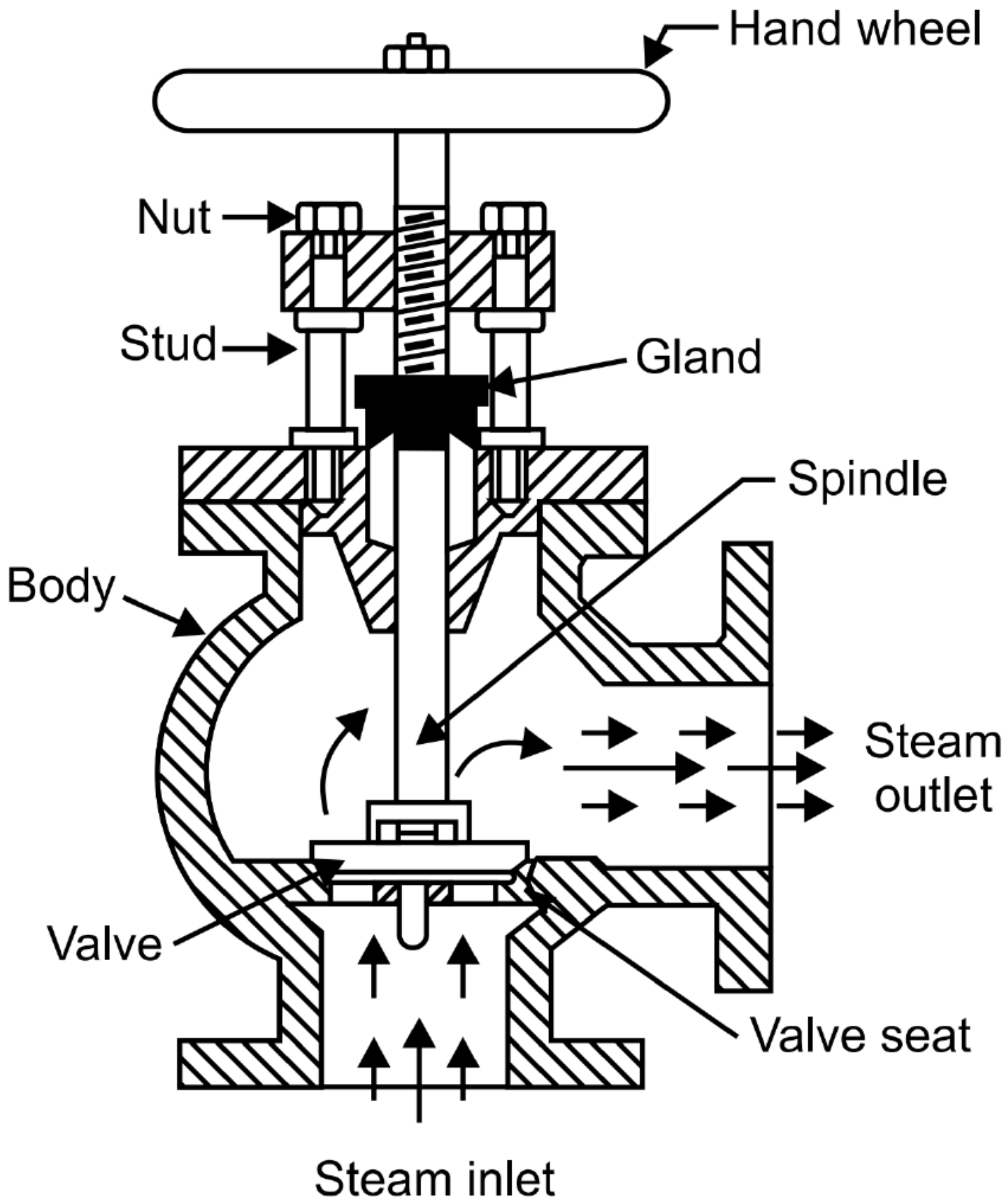


Fig. *Stop valve*

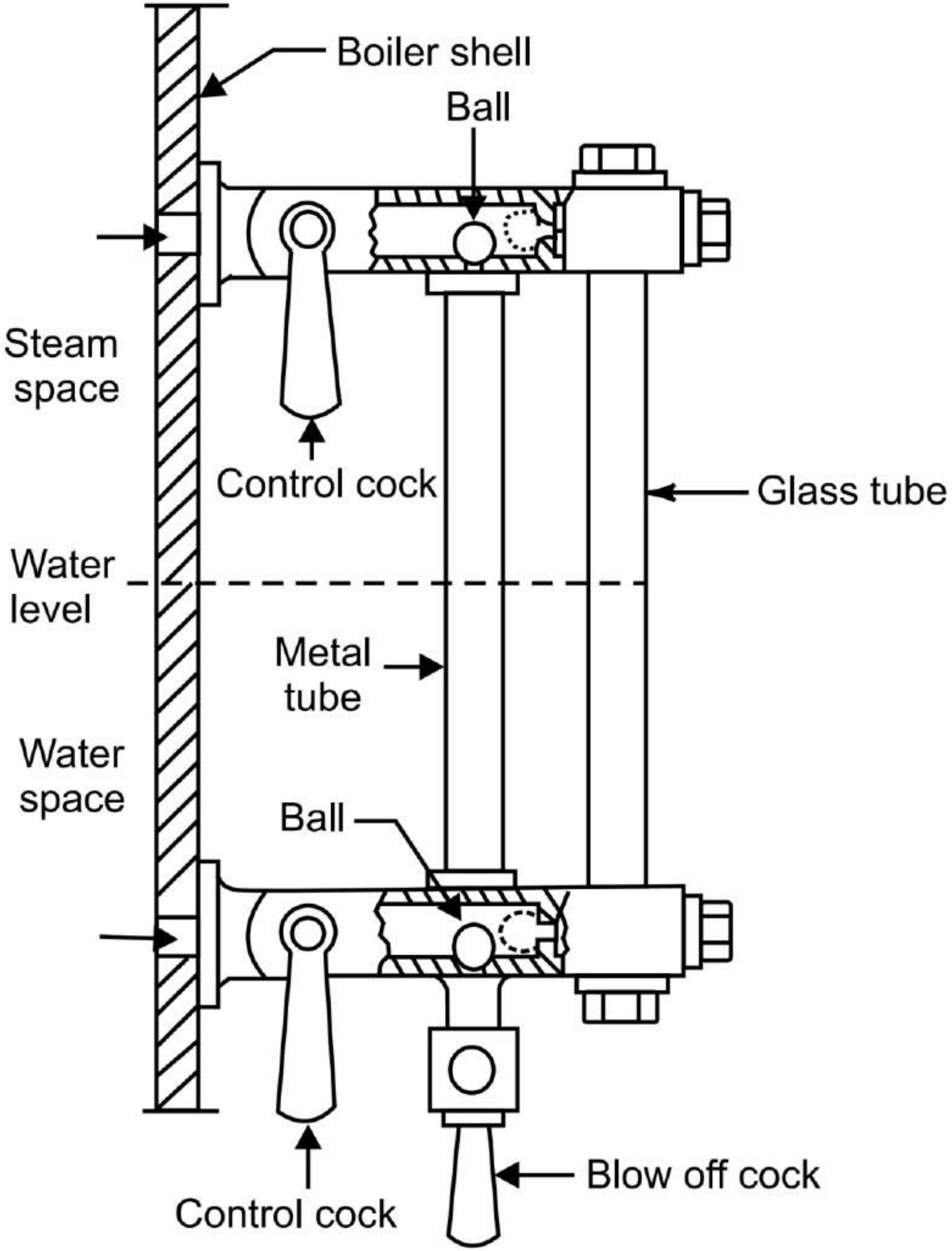


Fig. *Water level indicator*

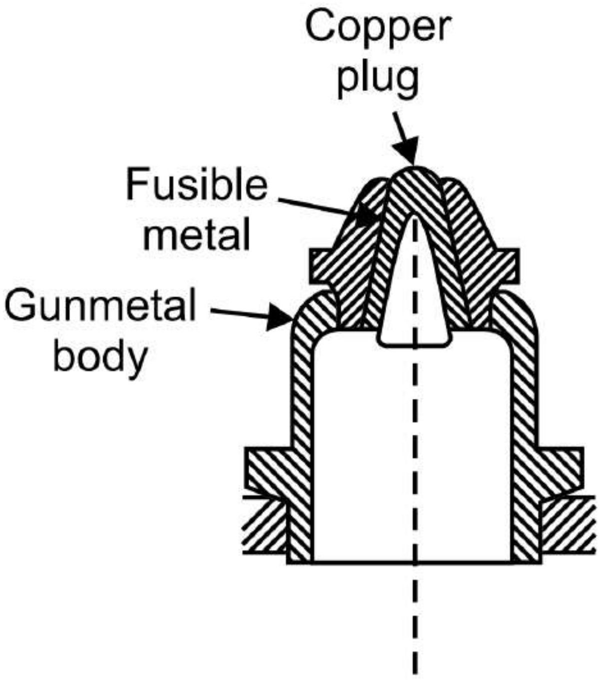


Fig. *Fusible plug*

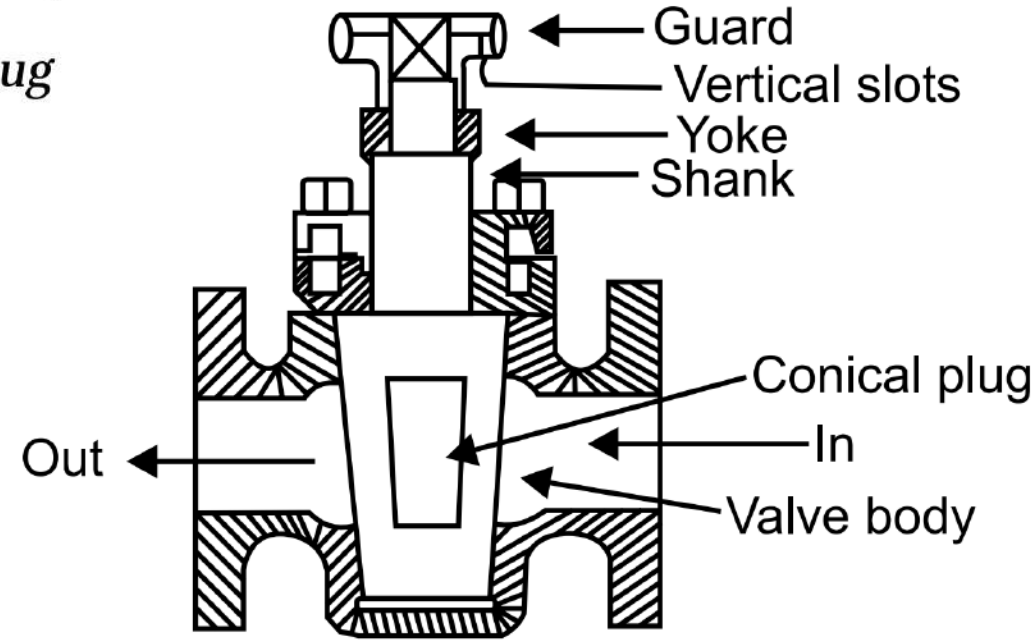


Fig. *Blow off cock*

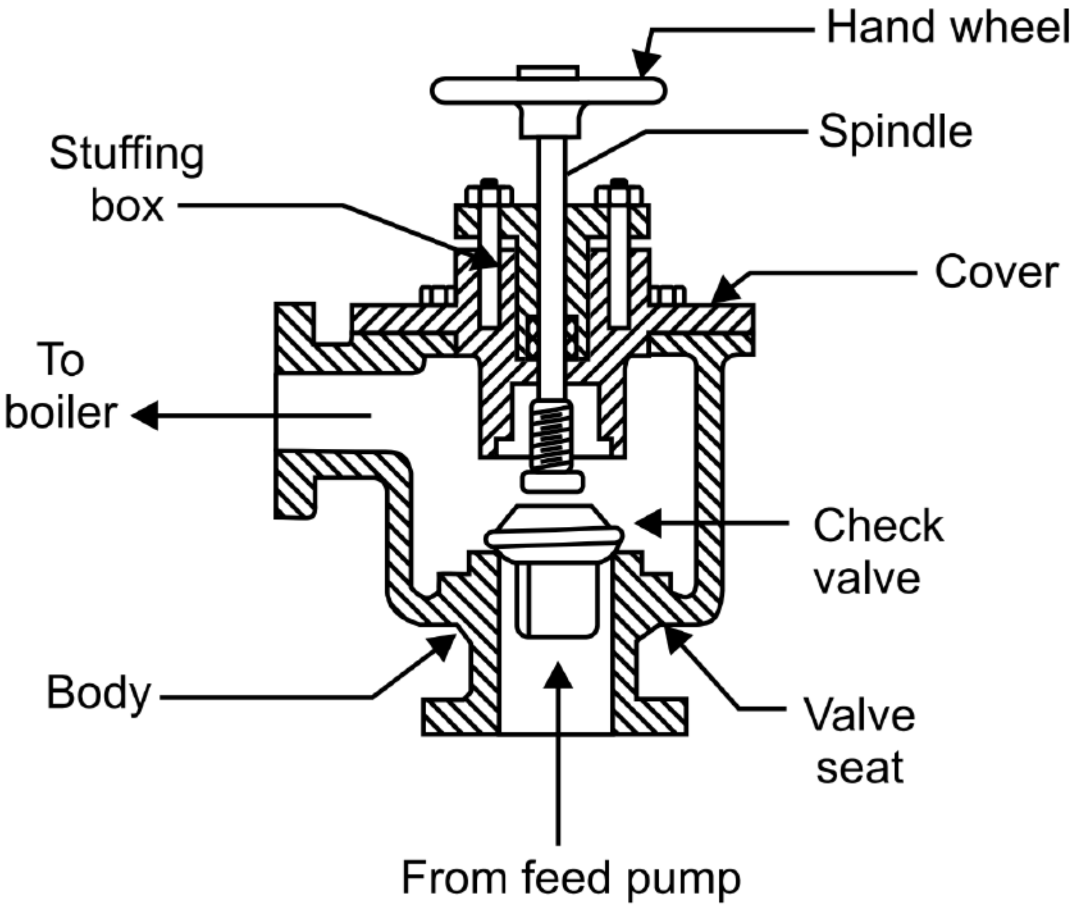
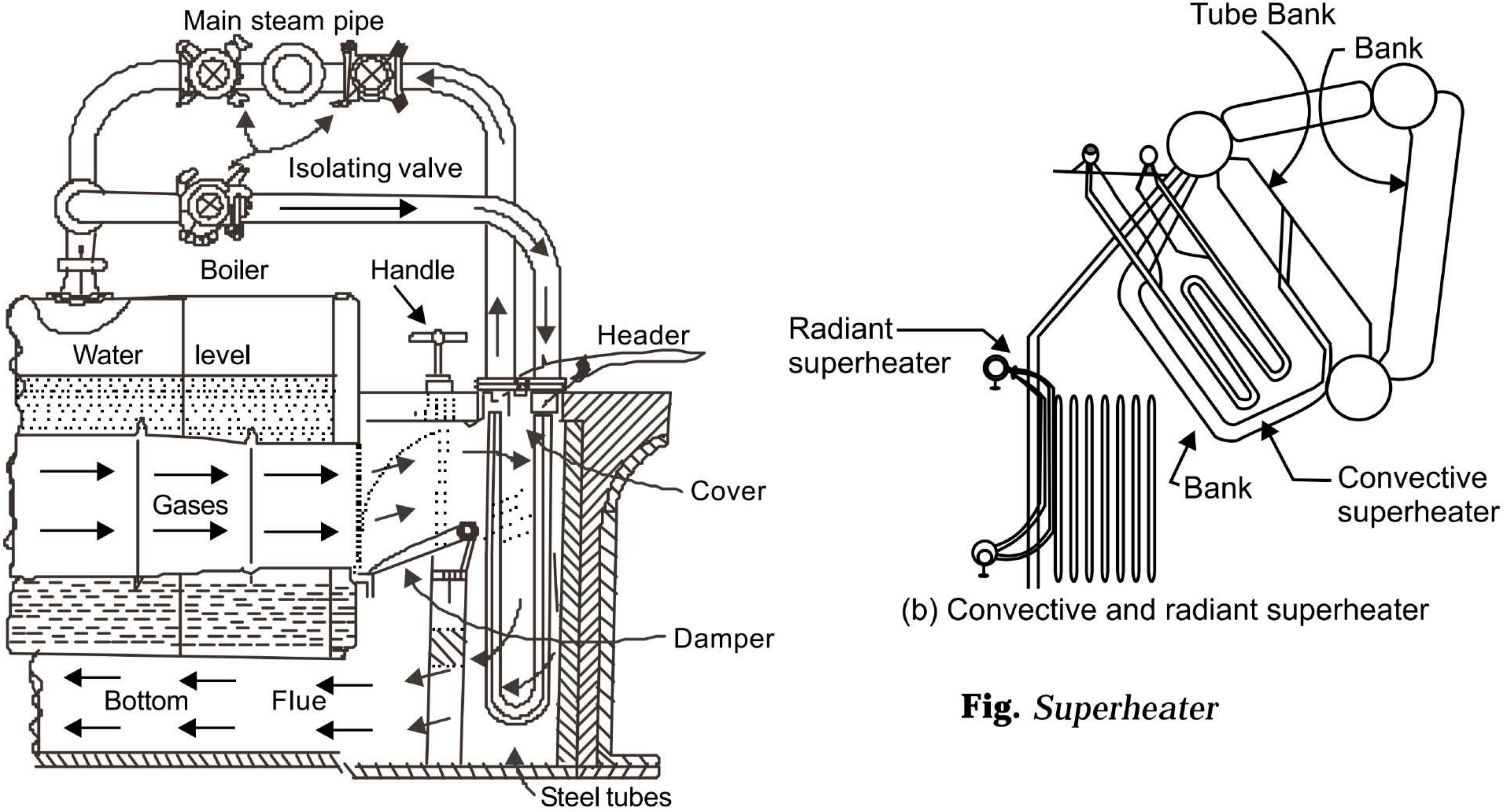


Fig. *Feed check valve*



(a) Smooth tube hairpin type superheater

(b) Convective and radiant superheater

Fig. *Superheater*

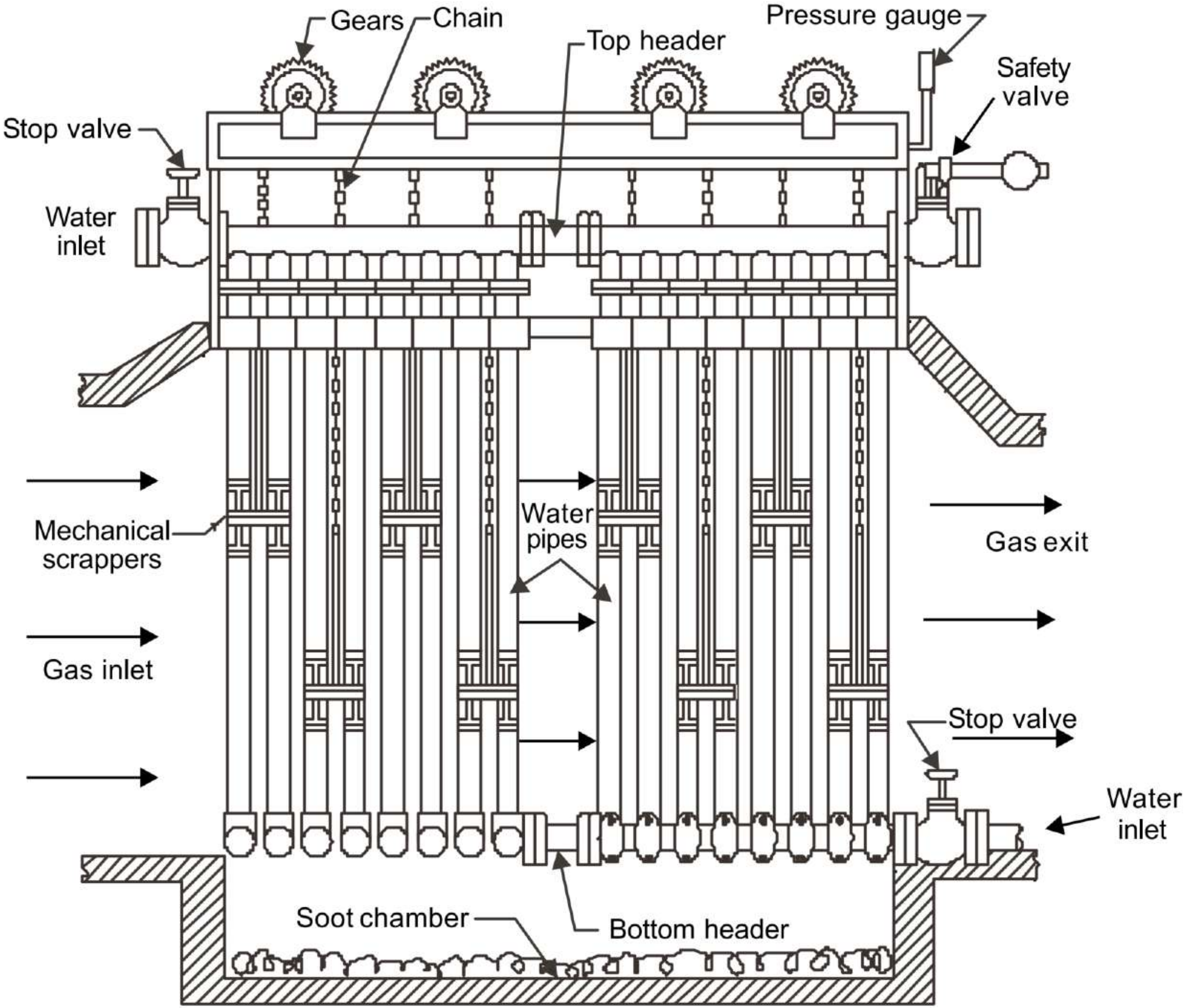


Fig. *Economizer*

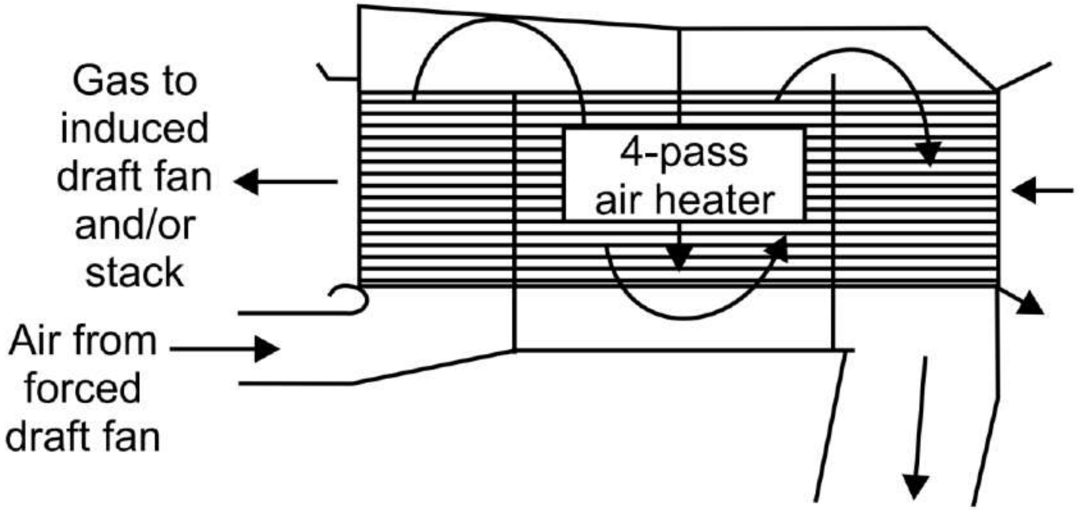


Fig. *Tubular air preheater*

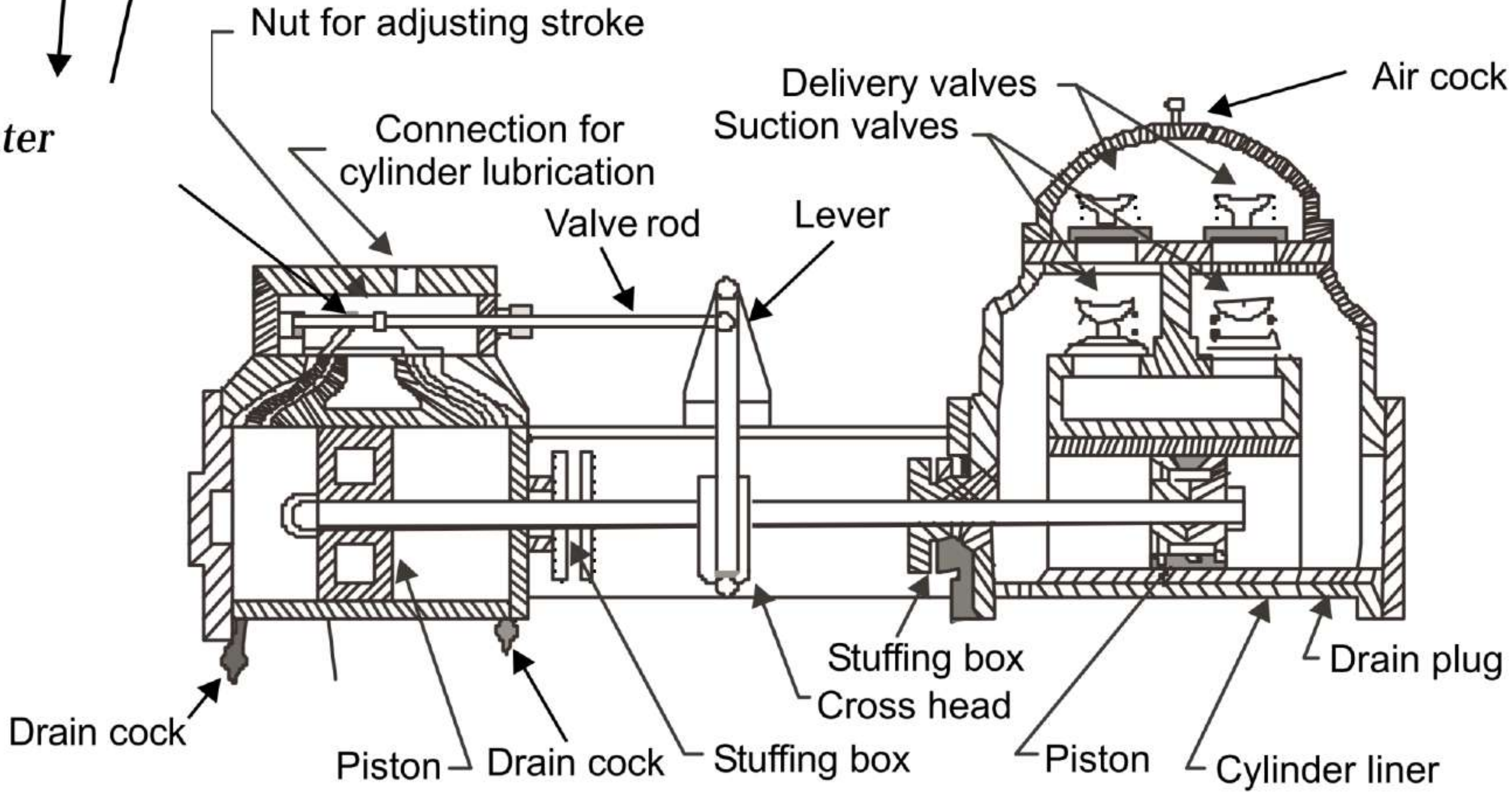


Fig. *Reciprocating type pump, Duplex feed pump*

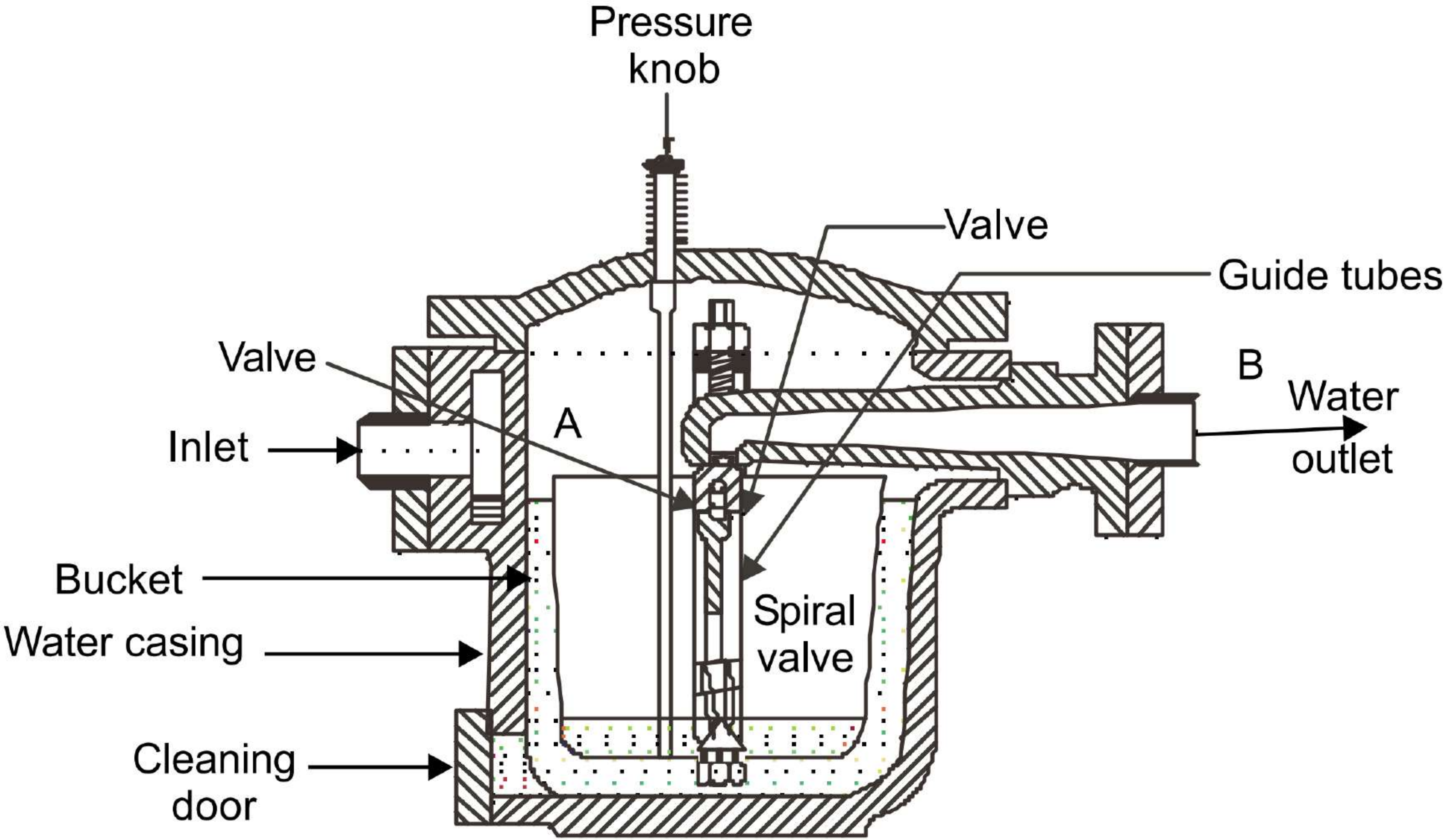


Fig. *Bucket type steam trap*

BOILER TRIAL:

Boiler trial refers to running the boiler under test conditions for its performance estimation. It gives the steam generation capacity of boiler, thermal efficiency of plant and heat balance sheet of the boiler. Under trial the boiler is run for quite long durations so as to attain steady state. Generally the boilers are run for 4 to 6 hours duration for the boilers of oil fired type and coal fired types. Duration of boiler run for attaining steady state changes from boiler to boiler. Observations are taken after the boiler attains steady state for a duration ranging from 10–15 minutes. Measurements are made for fuel supply, combustion analysis, steam generation rate and its quality/state, flue gas and their analysis, temperature and pressure at salient locations and all other measurements as required for heat balance sheet preparation.

BOILER EFFICIENCY

Boiler efficiency quantifies, how effectively the heat is being used in boiler. Thus it could be given by the ratio of heat actually used for steam generation and total heat available due to combustion of fuel in boiler.

$$\text{Boiler efficiency} = \frac{\text{Heat used in steam generation}}{\text{Total heat available due to fuel burning}}$$

$$= \frac{m\left(h - h_w\right)}{m_f \times \text{C.V.}}$$

here m_f is the mass of fuel burnt per hour, C.V. is calorific value of fuel used (kcal/kg), m is mass of steam generated per hour and enthalpies h and h_w are that of final steam and feed water, kcal/kg.

Generally high heating value of fuel is used as calorific value of fuel.

Steam Nozzles

By

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Estd. in 1980

SAGI RAMA KRISHNAM RAJU ENGINEERING COLLEGE

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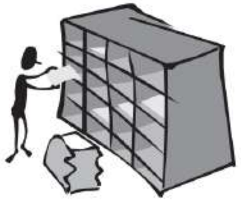
UNIT-III

Steam Nozzles: Type of nozzles-Flow through nozzles- Condition for maximum discharge- Nozzle efficiency- Super saturated flow in nozzles- - Steam injectors.

Textbooks:

1. Thermodynamics and Heat Engines/R.Yadav, Volume -II /Central Publishing House.
2. Heat Engineering by V.PVasandani and D.S Kumar/Metropolitan Book Company, New Delhi.
3. Thermal Engineering, by R. K. Rajput,Lakshmi Publications.
4. A Treatise on Heat Engineering by Vasandhani and Kumar.



Glossary

Critical pressure Pressure of steam at the throat

Critical velocity Velocity of steam at throat corresponds to maximum discharge condition

Supersaturation Existance of fluid in superheated state in wet region

Metastable State Supersaturation state of a fluid

Degree of supercooling Difference between saturation temperature and temperature of supersaturated vapour

Degree of supersaturation Ratio of actual pressure to the saturation pressure corresponding to temperature of steam condensate.

A ***steam nozzle*** may be defined as a passage of varying cross-section, through which heat energy of steam is converted to kinetic energy. Its major function is to produce steam jet with high velocity to drive steam turbines.

The cross-section of a nozzle at first tapers to a smaller section (to allow for changes which occur due to changes in velocity, specific volume and dryness fraction as the steam expands) ; the smallest section being known as *throat*, and then it diverges to a large diameter. The nozzle which converges to throat and diverges afterwards is known as ***convergent-divergent nozzle.***

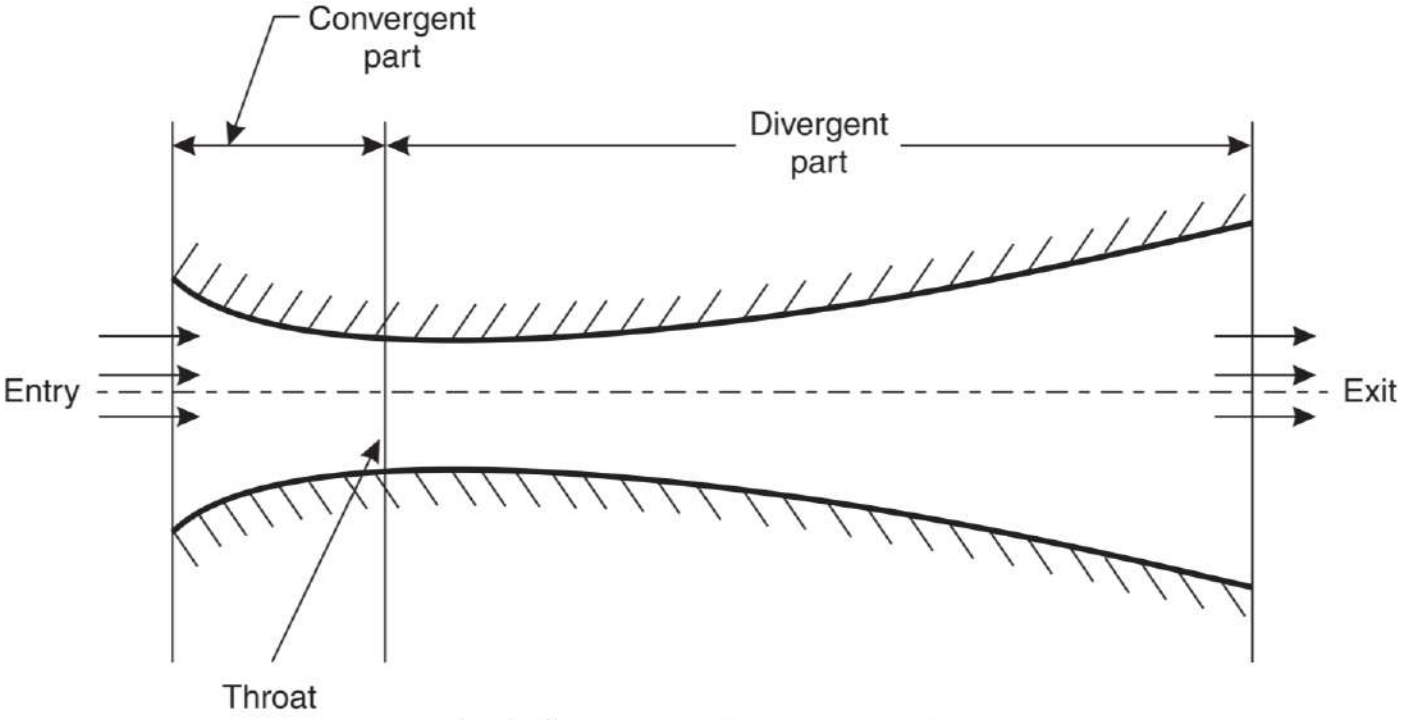


Fig. 1. Convergent-divergent nozzle.

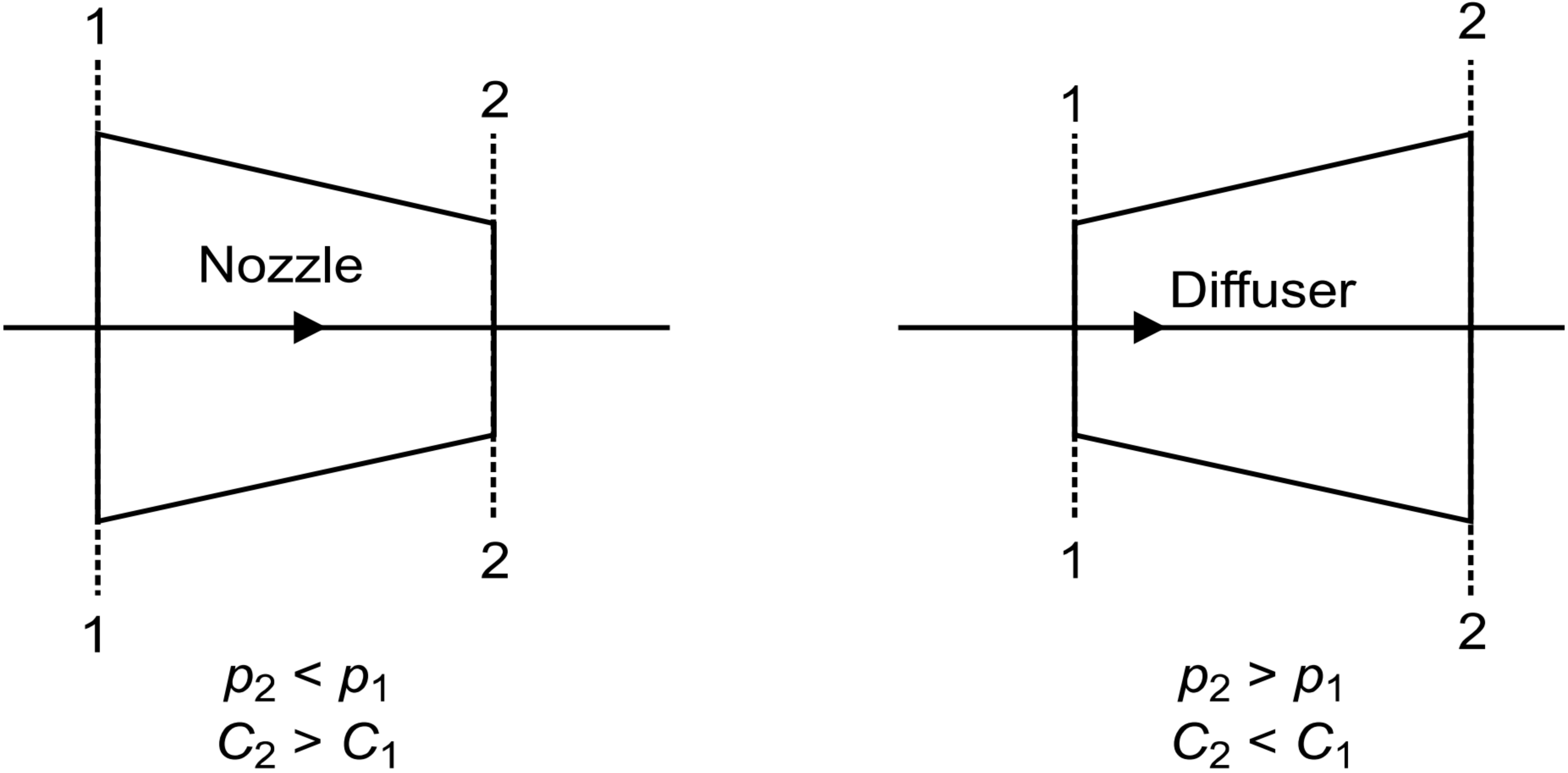


Fig. *General arrangement in nozzle and diffuser*

CONTINUITY EQUATION

Consider the flow of steam through a nozzle.

Let $\dot{m}$ = steady state mass flow rate of steam, kg/s

A = cross-sectional area of nozzle normal to the direction of steam flow at any section, m^2

v = specific volume of steam at the same section, m^3/kg

c = velocity of steam across the section, m/s

Then for steady flow of steam through the nozzle,

$$\boxed{\dot{m} = \frac{Ac}{v} = \frac{A_1 c_1}{v_1} = \frac{A_2 c_2}{v_2}}$$

Equation is the continuity equation for steam flow through the nozzle.

STEAM FLOW THROUGH NOZZLES

The steam flow through the nozzle may be assumed as ***adiabatic flow*** since during the expansion of steam in nozzle neither any heat is supplied nor rejected, work, however, is performed by increasing the kinetic energy of the steam. As the steam passes through the nozzle, it loses its pressure as well as the heat. *The work done is equal to the adiabatic heat drop which in turn is equal to Rankine area.*

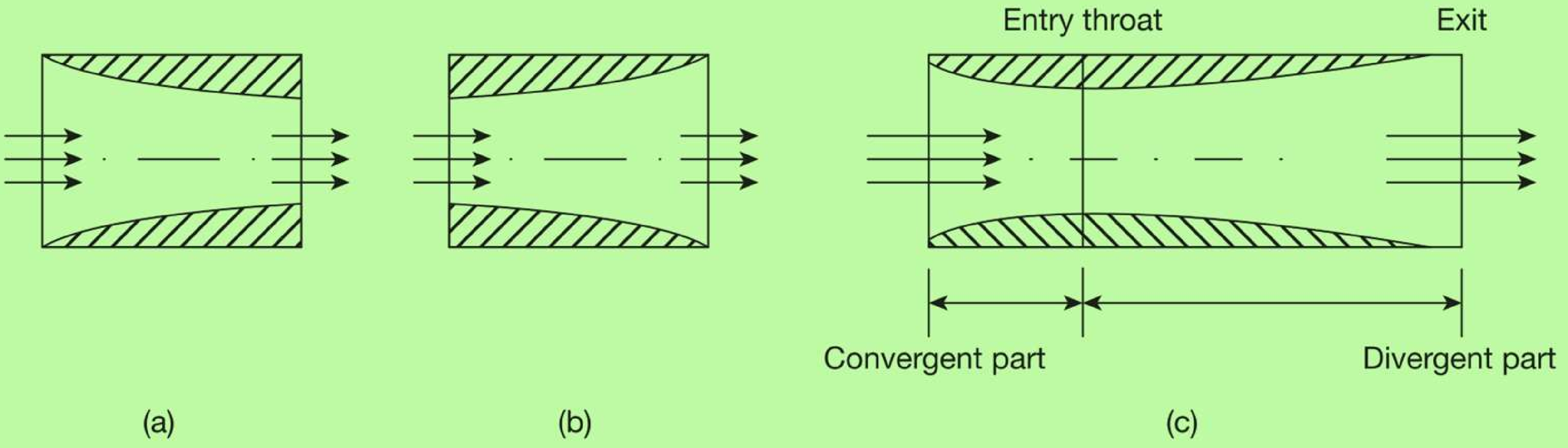


Figure *Types of nozzles: (a) Convergent nozzle (b) Divergent nozzle (c) Convergent-Divergent nozzle*

Types of nozzles :

1. ***Convergent nozzle.*** It is a smoothly varying cross-sectional area duct which is used for accelerating a steadily flowing fluid. The purpose of this nozzle is to *convert the internal energy of the fluid into the kinetic form.*

2. ***Convergent-divergent nozzle.*** This type of nozzle is a modification of the convergent type where there is a *divergent section which acts as an accelerator for supersonic flow.*

3. ***Steam nozzles.*** This is a special purpose convergent nozzle used in steam turbine for *accelerating the steam at the expense of its pressure.*

4. ***Flow nozzle.*** It is a device used for the measurement of discharge.

Velocity of Steam:

Steam enters the nozzle with high pressure and low initial velocity (*it is so small as compared to the final velocity that it is generally neglected*) and leaves it with high velocity and low pressure. This is due to the reason that heat energy of steam is converted into kinetic energy as it (steam) passes through the nozzle. The final or outlet velocity of steam can be found as follows :

Let C = Velocity of steam at the section considered (m/sec),
h_1 = Enthalpy of steam entering the nozzle,
h_2 = Enthalpy of steam at section considered, and
h_d = Heat drop during expansion of steam is the nozzle = $(h_1 - h_2)$.

Considering 1 kg of steam and flow to be frictionless adiabatic, we have :

Gain in kinetic energy = Adiabatic heat drop

$$\frac{C^2}{2} = h_d$$

$$\therefore \quad C = \sqrt{2 \times 1000\, h_d}\text{, where } h_d \text{ is in kJ}$$

$$= 44.72\sqrt{h_d} \qquad \ldots(1)$$

Discharge through the Nozzle and Conditions for its Maximum Value:

Let p_1 = Initial pressure of steam,
v_1 = Initial volume of 1 kg of steam at pressure p_1 (m^3),
p_2 = Steam pressure at the throat,
v_2 = Volume of 1 kg of steam at pressure p_2 (m^3),
A = Cross-sectional area of nozzle at throat (m^2), and
C = Velocity of steam (m/s).

The steam flowing through the nozzle follows approximately the equation given below :

$$pv^n = \text{Constant}$$

where, n = 1.135 for *saturated steam*, and
= 1.3 for *superheated steam*.

[For **wet steam,** the value of n can be calculated by Dr. *Zenner's equation,*
$n = 1.035 + 0.1x$, where x is the initial dryness fraction of steam]

Work done per kg of steam during the cycle (Rankine area)

$$= \frac{n}{n-1}(p_1v_1 - p_2v_2)$$

and, Gain in kinetic energy = Adiabatic heat drop

= Work done during Rakine cycle

or
$$\frac{C^2}{2} = \frac{n}{n-1}(p_1v_1 - p_2v_2)$$

$$= \frac{n}{n-1}\,p_1v_1\left(1 - \frac{p_2v_2}{p_1v_1}\right) \qquad \text{...(3)}$$

Also
$$p_1v_1^{\,n} = p_2v_2^{\,n}$$

or
$$\frac{v_2}{v_1} = \left(\frac{p_1}{p_2}\right)^{1/n} \qquad \text{...(4)}$$

or
$$v_2 = v_1\left(\frac{p_1}{p_2}\right)^{1/n} \qquad \text{...(5)}$$

Putting the value of v_2/v_1 from eqn. (4) in eqn. (3), we get

$$\frac{C^2}{2} = \frac{n}{n-1} \; p_1 v_1 \left[1 - \frac{p_2}{p_1} \left(\frac{p_1}{p_2} \right)^{1/n} \right] = \frac{n}{n-1} \; p_1 v_1 \left[1 - \left(\frac{p_2}{p_1} \right)^{1 - \frac{1}{n}} \right]$$

$$= \frac{n}{n-1} \; p_1 v_1 \left[1 - \left(\frac{p_2}{p_1} \right)^{\frac{n-1}{n}} \right]$$

$$C^2 = 2 \left(\frac{n}{n-1} \right) p_1 v_1 \left[1 - \left(\frac{p_2}{p_1} \right)^{\frac{n-1}{n}} \right]$$

$$C = \sqrt{2 \left(\frac{n}{n-1} \right) p_1 \, v_1 \left\{ 1 - \left(\frac{p_2}{p_1} \right)^{\frac{n-1}{n}} \right\}} \qquad \text{...(6)}$$

If m is the mass of steam discharged in kg/sec.,

Then
$$m = \frac{AC}{v_2} \qquad \text{...(7)}$$

Substituting the value of v_2 from eqn. (5) in eqn. (7),

$$m = \frac{AC}{v_1 \left(\dfrac{p_1}{p_2}\right)^{1/n}}$$

$$m = \frac{A}{v_1 \left(\dfrac{p_1}{p_2}\right)^{1/n}} \sqrt{2\left(\frac{n}{n-1}\right) p_1 v_1 \left[1 - \left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}}\right]}$$

$$= \frac{A}{v_1} \sqrt{\left[2\left(\frac{n}{n-1}\right) p_1 v_1 \left\{\left(\frac{p_2}{p_1}\right)^{2/n} - \left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} \left(\frac{p_2}{p_1}\right)^{2/n}\right\}\right]}$$

$$= \frac{A}{v_1} \sqrt{\left[2\left(\frac{n}{n-1}\right) p_1 v_1 \left\{\left(\frac{p_2}{p_1}\right)^{2/n} - \left(\frac{p_2}{p_1}\right)^{\frac{n+1}{n}}\right\}\right]} \qquad \text{...(8)}$$

It is obvious from above equation that there is only one value of the ratio (called *critical pressure ratio*) p_2/p_1 which will produce the *maximum discharge*. This can be obtained by differentiating 'm' with respect to (p_2/p_1) and equating it to zero.

As other quantities except the ratio p_2/p_1 are constant,

$$\therefore \qquad \frac{d}{d\left(\dfrac{p_2}{p_1}\right)}\left[\left(\frac{p_2}{p_1}\right)^{2/n}-\left(\frac{p_2}{p_1}\right)^{\frac{n+1}{n}}\right]=0$$

or
$$\frac{2}{n}\left(\frac{p_2}{p_1}\right)^{\frac{2}{n}-1}-\left(\frac{n+1}{n}\right)\left(\frac{p_2}{p_1}\right)^{\frac{n+1}{n}-1}=0$$

or
$$\left(\frac{p_2}{p_1}\right)^{\frac{2}{n}-1}=\frac{n+1}{n}\left(\frac{p_2}{p_1}\right)^{1/n}$$

or
$$\left(\frac{p_2}{p_1}\right)^{2-n}=\left(\frac{n+1}{2}\right)^{n}\left(\frac{p_2}{p_1}\right)$$

or
$$\left(\frac{p_2}{p_1}\right)^{2-n-1} = \left(\frac{n+1}{2}\right)^{n}$$

or
$$\frac{p_2}{p_1} = \left(\frac{2}{n+1}\right)^{\frac{n}{n-1}} \qquad \text{...(9)}$$

Hence the discharge through the nozzle will be the *maximum* when critical pressure ratio, *i.e.*,

$$\frac{\text{Throat pressure}}{\text{Inlet pressure}} = \frac{p_2}{p_1} = \left(\frac{2}{n+1}\right)^{\frac{n}{n-1}}$$

For saturated steam : $n = 1.135$

$$\frac{p_2}{p_1} = \left(\frac{2}{1.135+1}\right)^{\frac{1.135}{1.135-1}} = \left(\frac{2}{2.135}\right)^{\frac{1.135}{0.135}} = 0.58$$

For superheated steam : $n = 1.3$

$$\frac{p_2}{p_1} = \left(\frac{2}{1.3+1}\right)^{\frac{1.3}{1.3-1}} = \left(\frac{2}{2.3}\right)^{\frac{1.3}{0.3}} = 0.546$$

Substituting the value of $\frac{p_2}{p_1}$ from eqn. (9) into eqn. (8), we get the maximum discharge,

$$m_{max} = \frac{A}{v_1}\sqrt{2\left(\frac{n}{n-1}\right)p_1 v_1\left[\left\{\left(\frac{2}{n+1}\right)^{\frac{n}{n-1}}\right\}^{\frac{2}{n}}\left\{\left(\frac{2}{n+1}\right)^{\frac{n}{n-1}}\right\}^{\frac{n+1}{n}}\right]}$$

$$= \frac{A}{v_1}\sqrt{2\left(\frac{n}{n-1}\right)p_1 v_1\left[\left(\frac{2}{n+1}\right)^{\frac{2}{n-1}} - \left(\frac{2}{n+1}\right)^{\frac{n+1}{n-1}}\right]}$$

$$= A\sqrt{2\left(\frac{n}{n-1}\right)\frac{p_1}{v_1}\left[\left(\frac{2}{n+1}\right)^{\frac{2}{n-1}} - \left(\frac{2}{n+1}\right)^{\frac{n+1}{n-1}}\right]}$$

$$= A\sqrt{2\left(\frac{n}{n-1}\right)\frac{p_1}{v_1}\left(\frac{2}{n+1}\right)^{\frac{n+1}{n-1}}\left[\left(\frac{2}{n+1}\right)^{\frac{2}{n-1}-\frac{n+1}{n-1}} - 1\right]}$$

$$= A\sqrt{2\left(\frac{n}{n-1}\right)\left(\frac{p_1}{v_1}\right)\left(\frac{2}{n+1}\right)^{\frac{n+1}{n-1}}\left[\left(\frac{2}{n+1}\right)^{\frac{1-n}{n-1}} - 1\right]}$$

$$= A\sqrt{2\left(\frac{n}{n-1}\right)\left(\frac{p_1}{v_1}\right)\left(\frac{2}{n+1}\right)^{\frac{n+1}{n-1}}\left[\left(\frac{2}{n+1}\right)^{-1}-1\right]}$$

$$= A\sqrt{2\left(\frac{n}{n-1}\right)\left(\frac{p_1}{v_1}\right)\left(\frac{2}{n+1}\right)^{\frac{n+1}{n-1}}\left(\frac{n-1}{2}\right)}$$

i.e.,

$$\boxed{m_{max} = A\sqrt{n\left(\frac{p_1}{v_1}\right)\left(\frac{2}{n+1}\right)^{\frac{n+1}{n-1}}}} \qquad \text{...(10)}$$

From the above equation it is evident that *the maximum mass flow depends only on the initial condition of the steam* (p_1, v_1) *and the throat area* and is *independent of the final pressure of steam i.e.,* at the exit of the nozzle. *The addition of the divergent part of the nozzle after the throat does not affect the discharge of steam passing through the nozzle but it only accelerates the steam leaving the nozzle.*

It may be noted that the discharge through nozzle increases as the pressure at the throat of the nozzle (p_2) decreases, when the supply pressure p_1 is constant. But once the nozzle pressure p_2 reaches the critical value [given by equation (9)], the discharge reaches a maximum and after that the throat pressure and mass flow remains constant irrespective of the pressure at the exit.

The velocity of steam at the throat of the nozzle when the discharge is maximum is obtained by substituting the value of $\frac{p_2}{p_1}$ from eqn. (9) into eqn. (6).

$$C_{max} = \sqrt{2\left(\frac{n}{n-1}\right)p_1v_1\left[1-\left\{\left(\frac{2}{n+1}\right)^{\frac{n}{n-1}}\right\}^{\frac{n-1}{n}}\right]}$$

$$= \sqrt{2\left(\frac{n}{n-1}\right)p_1v_1\left(1-\frac{2}{n+1}\right)}$$

$$= \sqrt{2\left(\frac{n}{n-1}\right)p_1v_1\left(\frac{n-1}{n+1}\right)}$$

i.e.,

$$\boxed{C_{max} = \sqrt{2\left(\frac{n}{n+1}\right)p_1v_1}} \quad \text{...(11)}$$

The above equation indicates that the *velocity is also dependent on the initial conditions of the steam.*

NOZZLE EFFICIENCY

When the steam flows through a nozzle the final velocity of steam for a given pressure drop is reduced due to the following reasons :

(i) The friction between the nozzle surface and steam :

(ii) The internal friction of steam itself ; and

(iii) The shock losses.

Most of these frictional losses occur between the throat and exit in convergent-divergent nozzle. These frictional losses entail the following effects :

(i) The expansion is no more isentropic and enthalpy drop is reduced ;

(ii) The final dryness friction of steam is increased as the kinetic energy gets converted into heat due to friction and is absorbed by steam ;

(iii) The specific volume of steam is increased as the steam becomes more dry due to this frictional reheating.

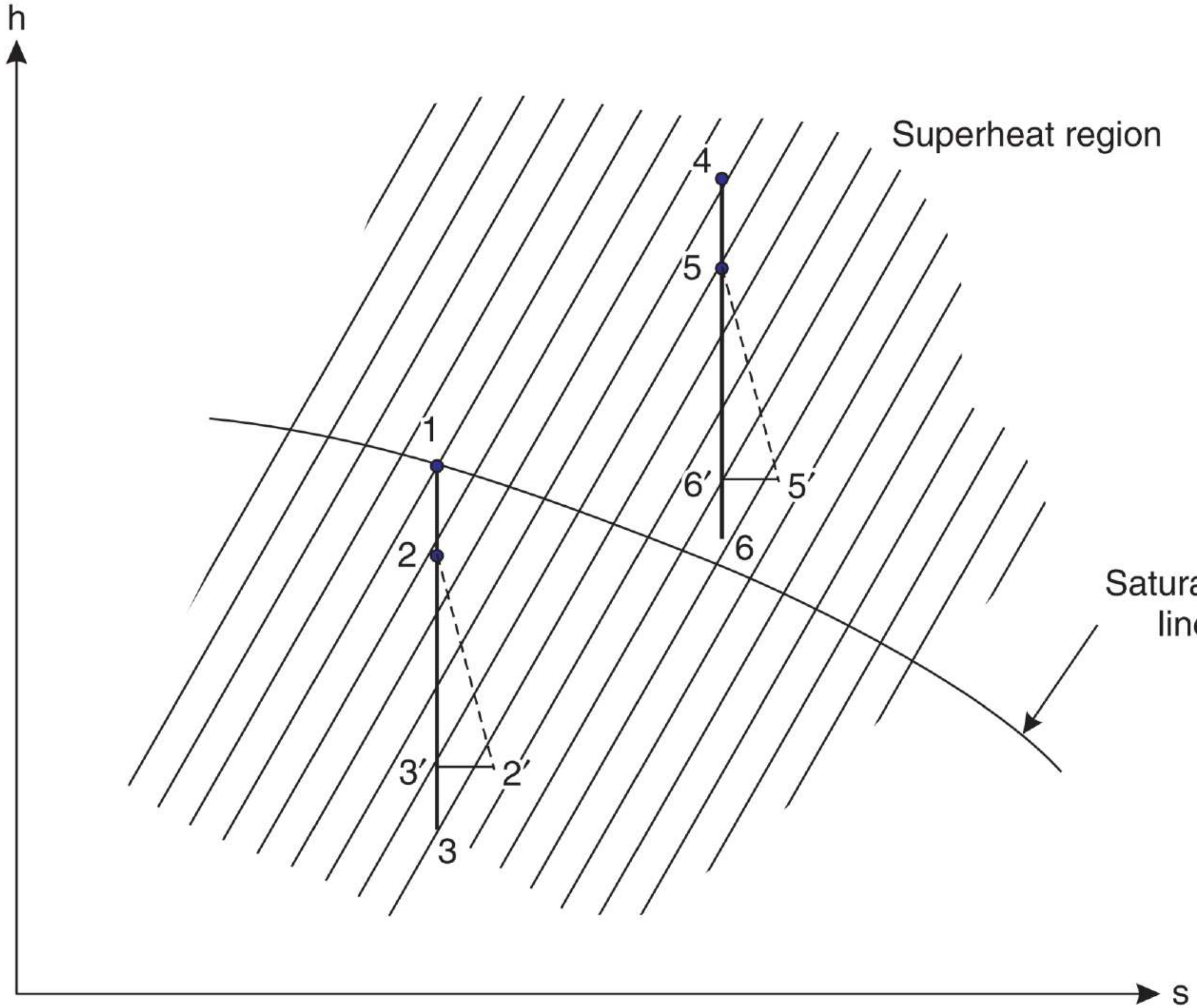


Fig. Effect of friction on steam flow through a nozzle.

Fig. represents on Mollier diagram the ***effect of friction*** *on steam flow through a nozzl*

The effect of friction is to improve the quality of steam. The value of co-efficient ‘k’ in the equation for the velocity of expanding steam is given by :

$$k = \frac{\text{Actual heat drop}}{\text{Isentropic heat drop}} = \frac{1-3'}{1-3} = \frac{h_1 - h_3{}'}{h_1 - h_3}$$

The actual expansion is represented by the curve 1-2-2′ since the ***friction occurs mainly between the throat and exit.***

On the other hand, if the steam at entry to nozzle were superheated corresponding to the point 4,

$$k = 4\text{-}6'/4\text{-}6 = (h_4 - h_6{}')/(h_4 - h_6).$$

It may be noted that the friction tends to superheat steam. Therefore, *it can be concluded that friction tends to decrease the velocity of steam and increase the final dryness fraction or superheat the steam.*

The nozzle efficiency is therefore defined as the ratio of *the actual enthalpy drop to the isentropy enthalpy drop between the same pressures,*

$$\text{i.e.,} \qquad \text{Nozzle efficiency} = \frac{h_1 - h_3'}{h_1 - h_3} \quad \text{or} \quad \frac{h_4 - h_6'}{h_4 - h_6}$$

If the actual velocity at exit from the nozzle is C_2' and the velocity at exit when the flow is isentropic is C_3, then using the steady flow energy equation, in each case we have

$$h_1 + \frac{C_1^2}{2} = h_3 + \frac{C_3^2}{2} \text{ or } h_1 - h_3 = \frac{C_3^2 - C_1^2}{2}$$

and

$$h_1 + \frac{C_1^2}{2} = h_2' + \frac{C_2'^2}{2} \text{ or } h_1 - h_2' = \frac{C_2'^2 - C_1^2}{2}$$

$$\therefore \quad \text{Nozzle efficiency} = \frac{C_2'^2 - C_1^2}{C_3^2 - C_1^2} \qquad \text{...(13)}$$

When the inlet velocity, C_1, is negligibly small then

Nozzle efficiency $= \dfrac{C_2'^2}{C_3^2}$ ...(14)

Sometimes a "*velocity co-efficient*" *is defined as the ratio of the actual exit velocity to the exit velocity when the flow is isentropic between the same pressures,*

i.e., *Velocity co-efficient* $= \dfrac{C_2'}{C_3}$ ...(15)

It can be seen from eqns. (14) and (15) that the "*velocity co-efficient*" *is the square root of the nozzle efficiency, when the inlet velocity is assumed to be negligible.*

The factors on which nozzle efficiency depends are :

1. **Material of the nozzle.**
2. **Workmanship of the manufacture of nozzle.**
3. **Size and shape of the nozzle.**
4. **Reynolds number of flow.**
5. **Angle of divergence of divergent portion.**
6. **Nature of fluid flowing and its state.**
7. **Turbulence in fluid and its state.**

SUPERSATURATED OR METASTABLE FLOWTHROUGH A NOZZLE

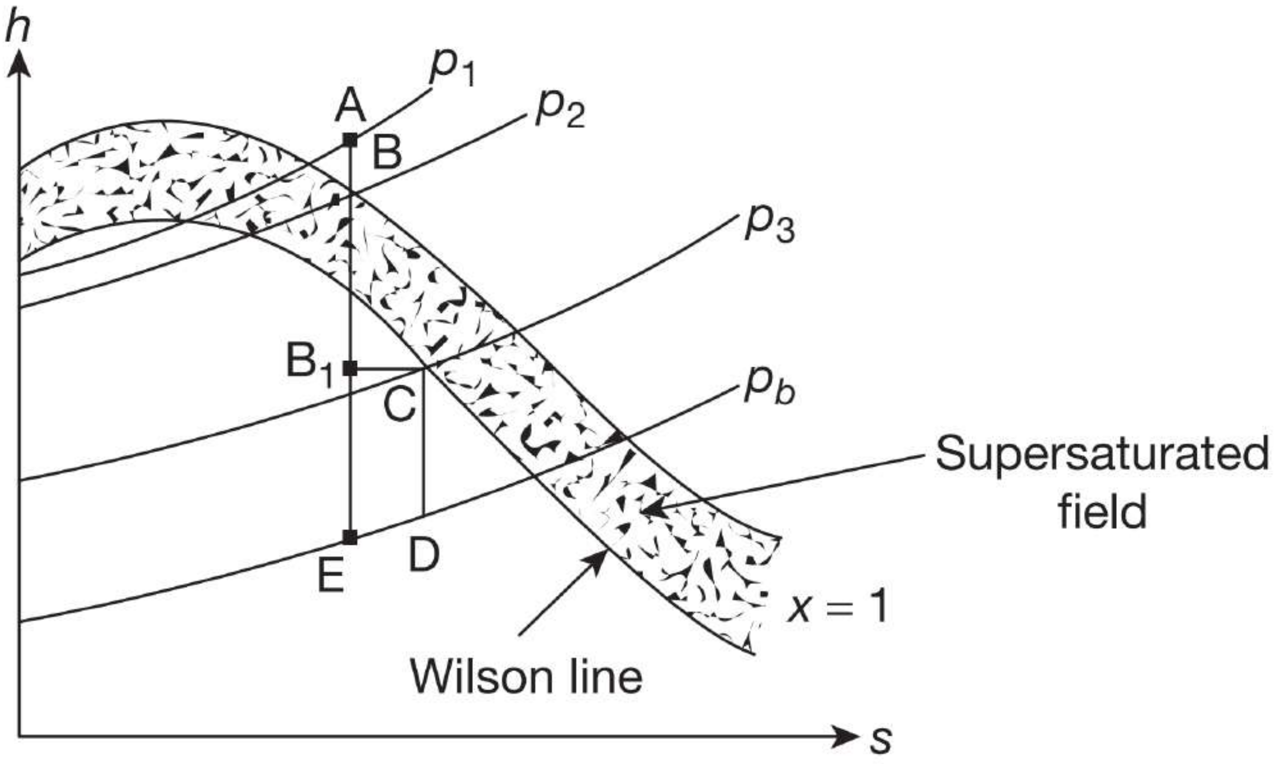
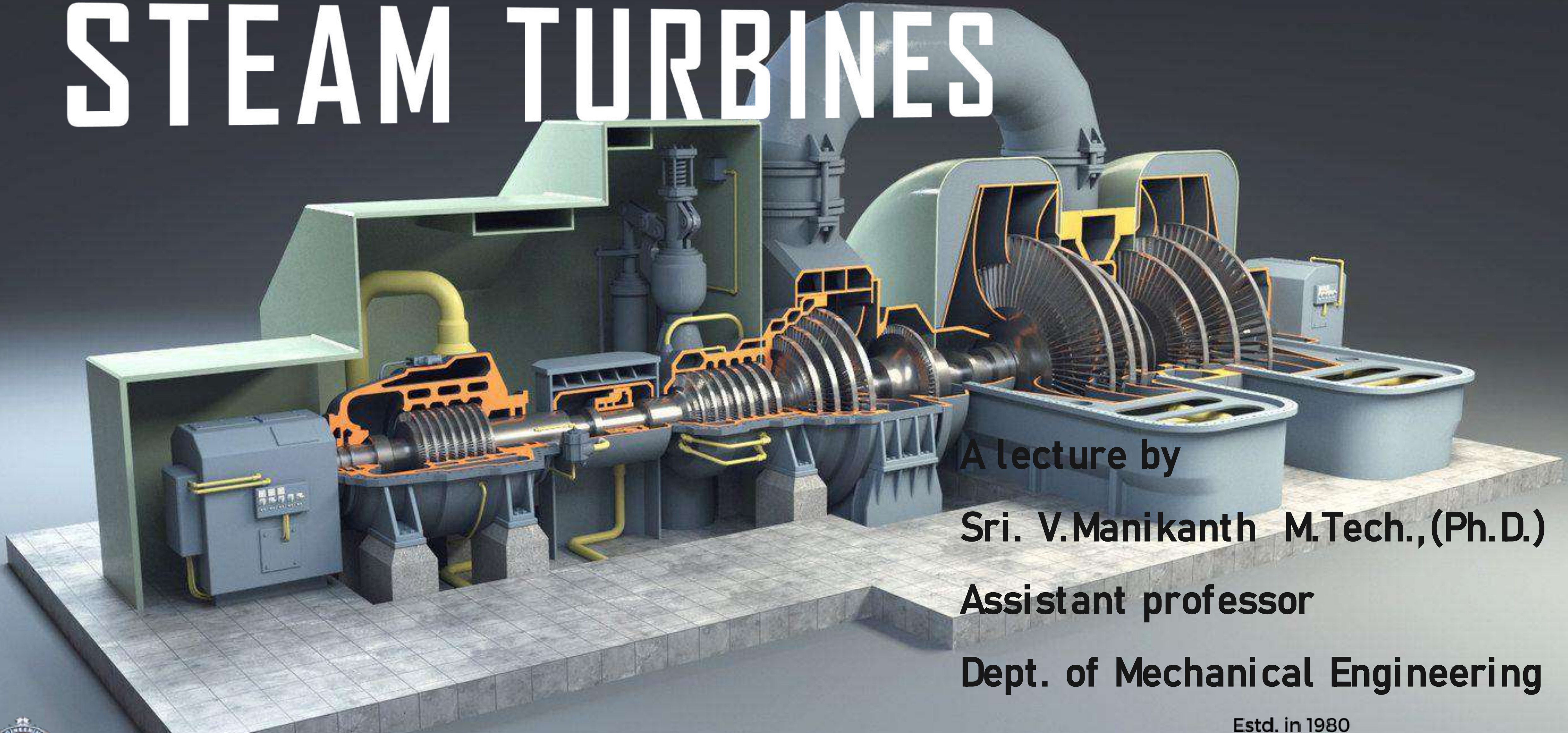


Figure *Super-saturated flow of steam in a nozzle*

STEAM TURBINES



A lecture by

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Assistant professor
Dept. of Mechanical Engineering

Estd. in 1980
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Estd. 1980

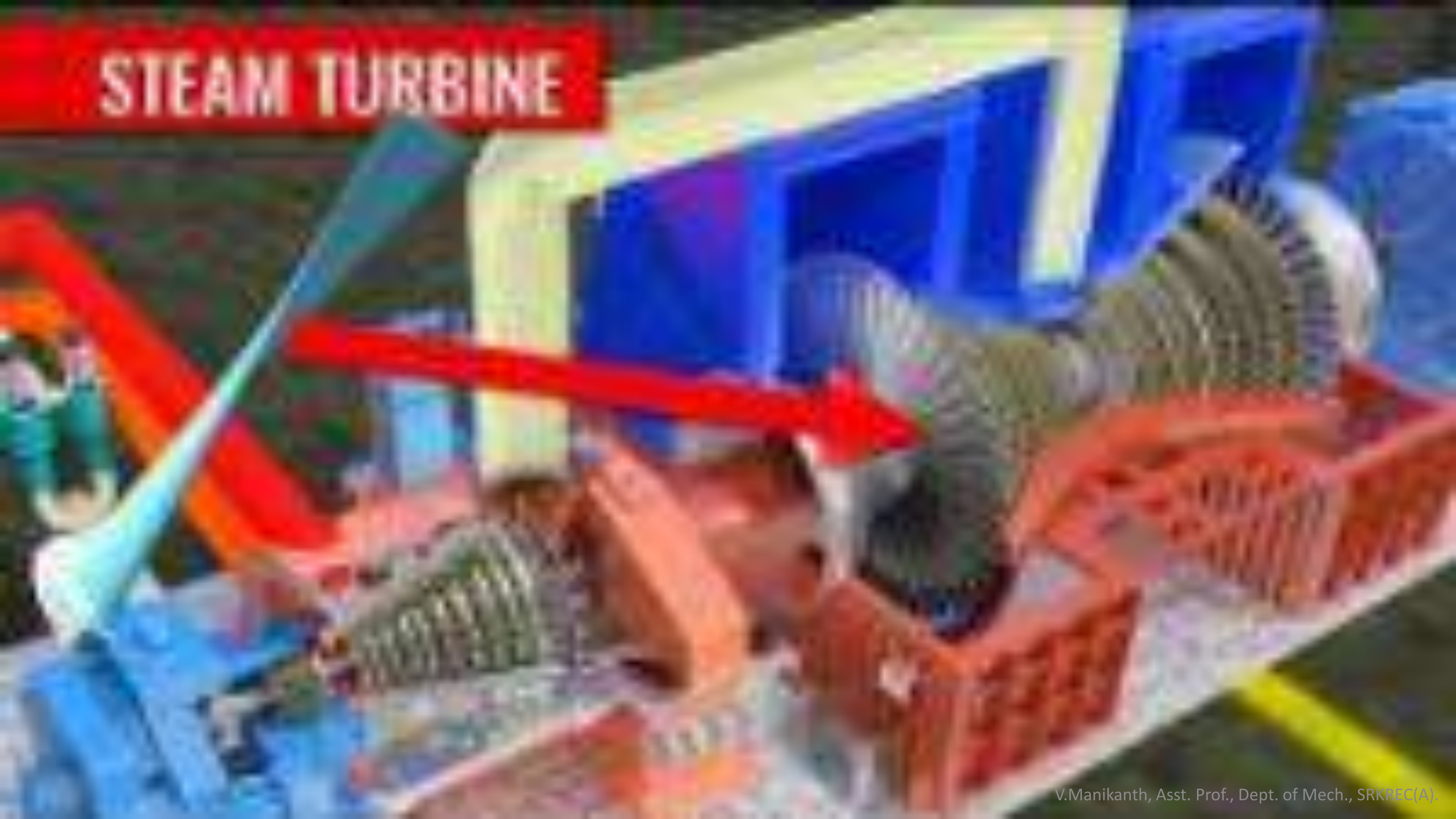
CONTENTS:

Steam turbines:

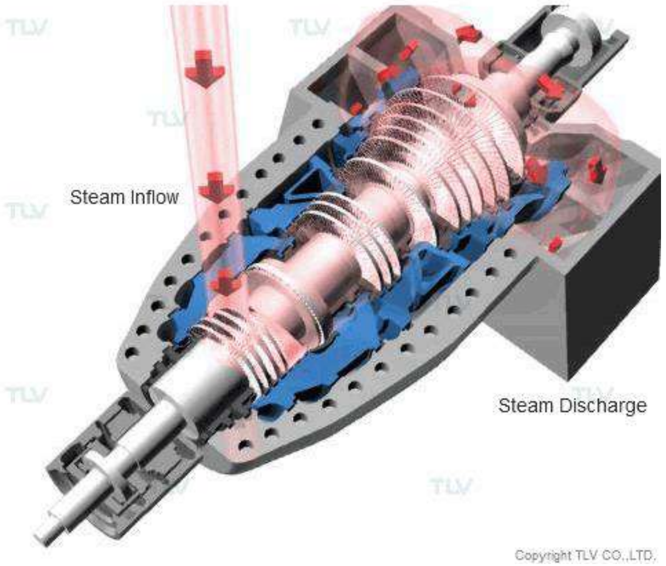
Classification of steam turbines- Impulse turbine and reaction turbine Compounding in turbines- Velocity diagrams in impulse and reaction turbines- Degree of reaction- Condition for maximum efficiency of reaction turbines

Textbooks:

1. Thermal Engineering by Rajput, Lakshmi publications
2. Thermal engineering by M.L. Mathur and F.S. Mehta, Jain Brothers.
3. Thermodynamics & Heat Engines by B.Yadav, Central Book Depot, Allahabad.

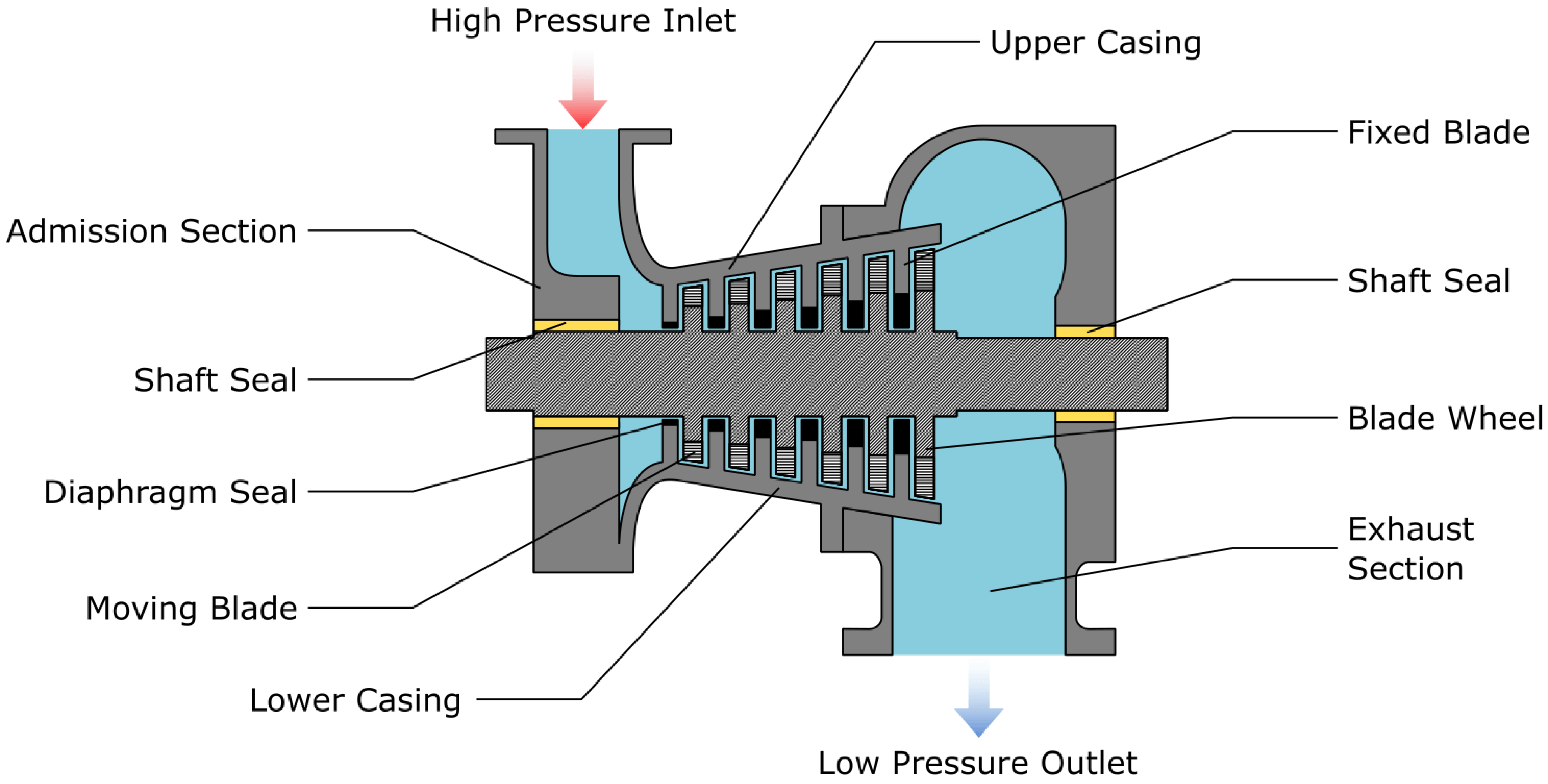

STEAM TURBINE

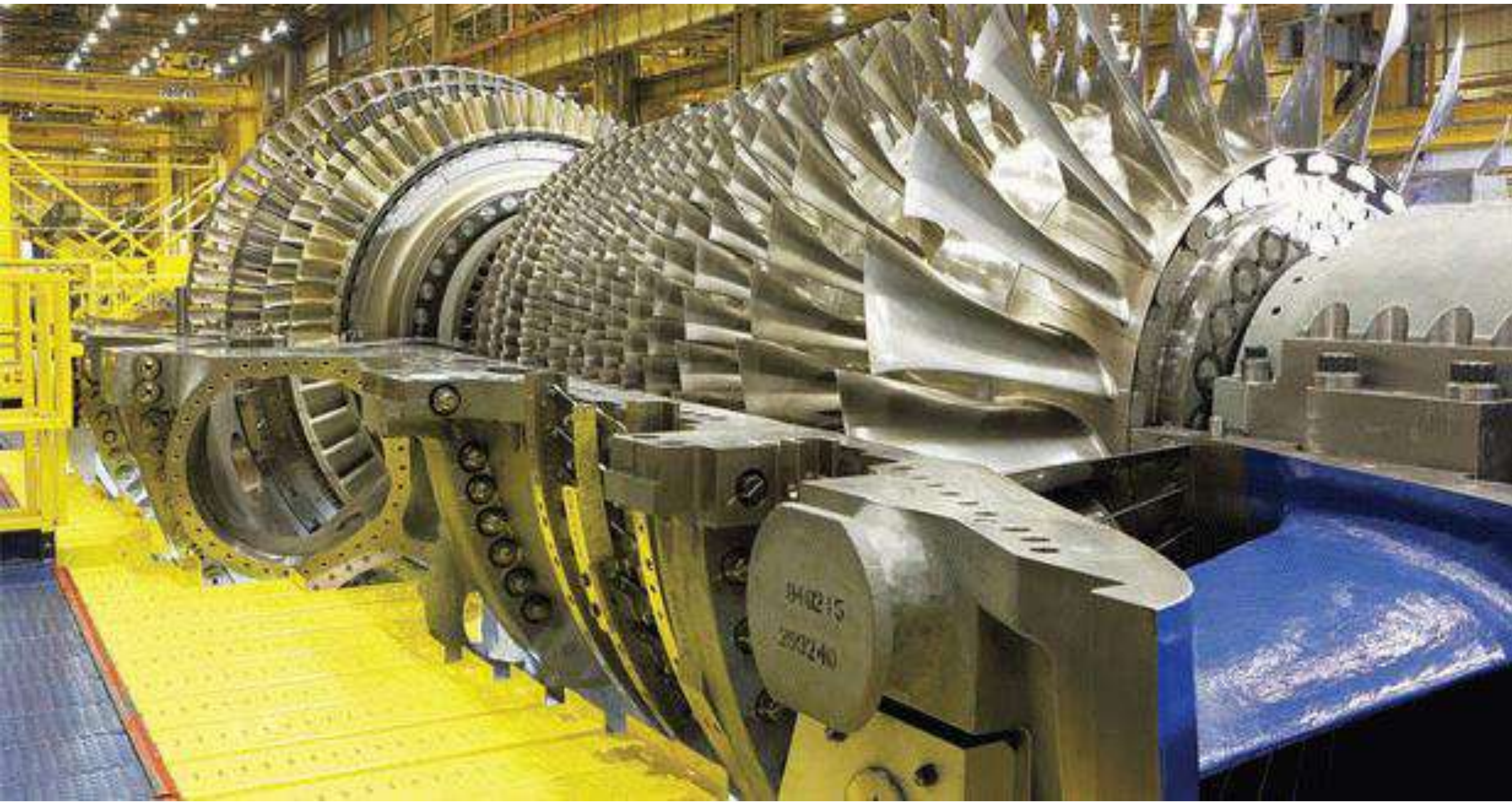
A ***steam turbine*** is a prime mover in which rotating shaft work is developed by a steadily flowing fluid. The work is produced by changing the momentum of steam as it passes through a rotor of the turbine. The change in angular momentum of steam causes the torque on the rotor, thus the rotor spins. Therefore, a steam turbine is a *rotodynamic machine*.



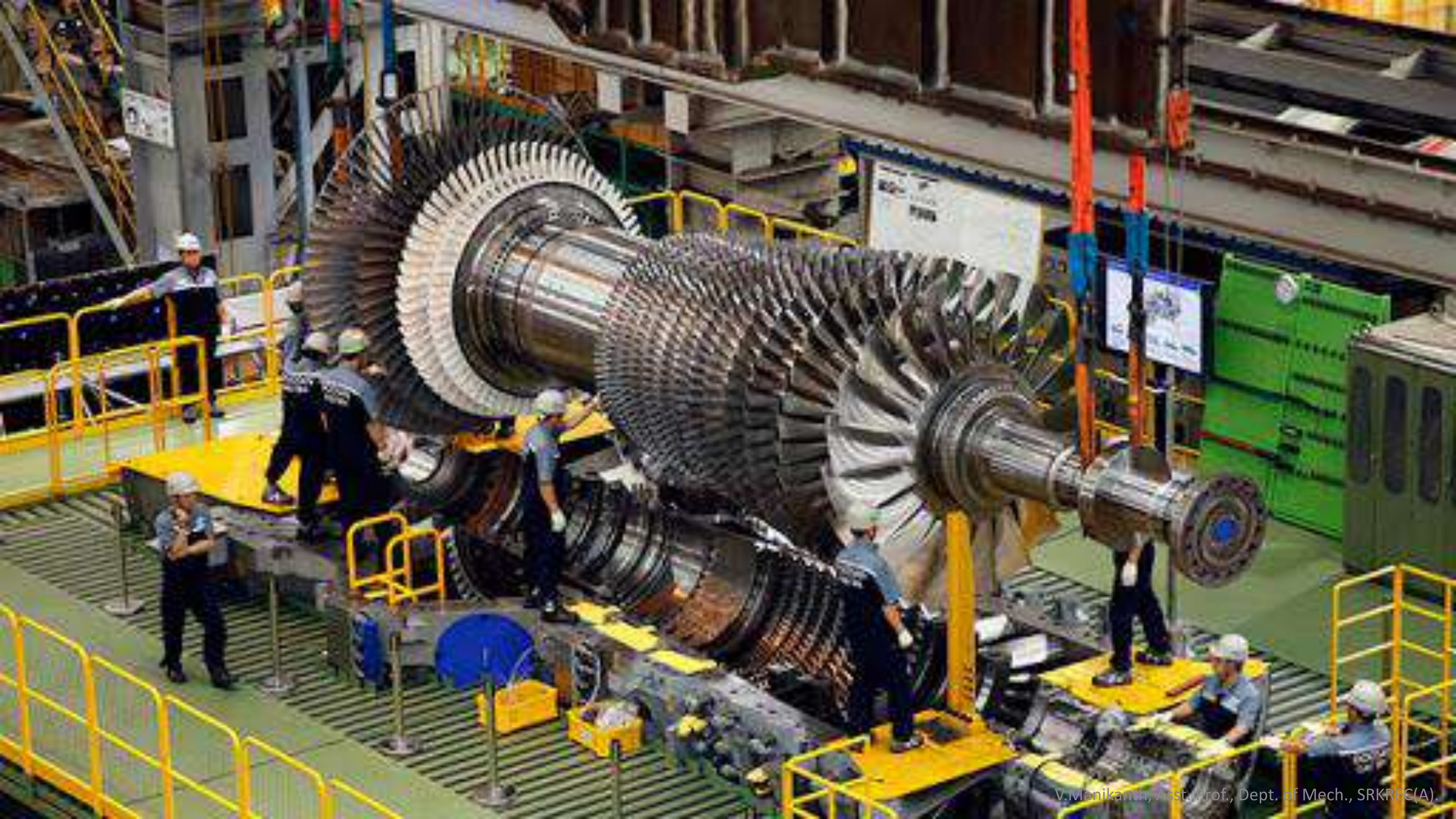
The steam turbine is a form of heat engine that derives much of its improvement in thermodynamic efficiency from the use of multiple stages in the expansion of the steam, which results in a closer approach to the ideal reversible expansion process.

The modern steam turbine was invented in 1884 by *Charles Parsons* who was an Anglo-Irish engineer, which revolutionized marine transport and naval warfare also made a cheap and plentiful electricity possible.

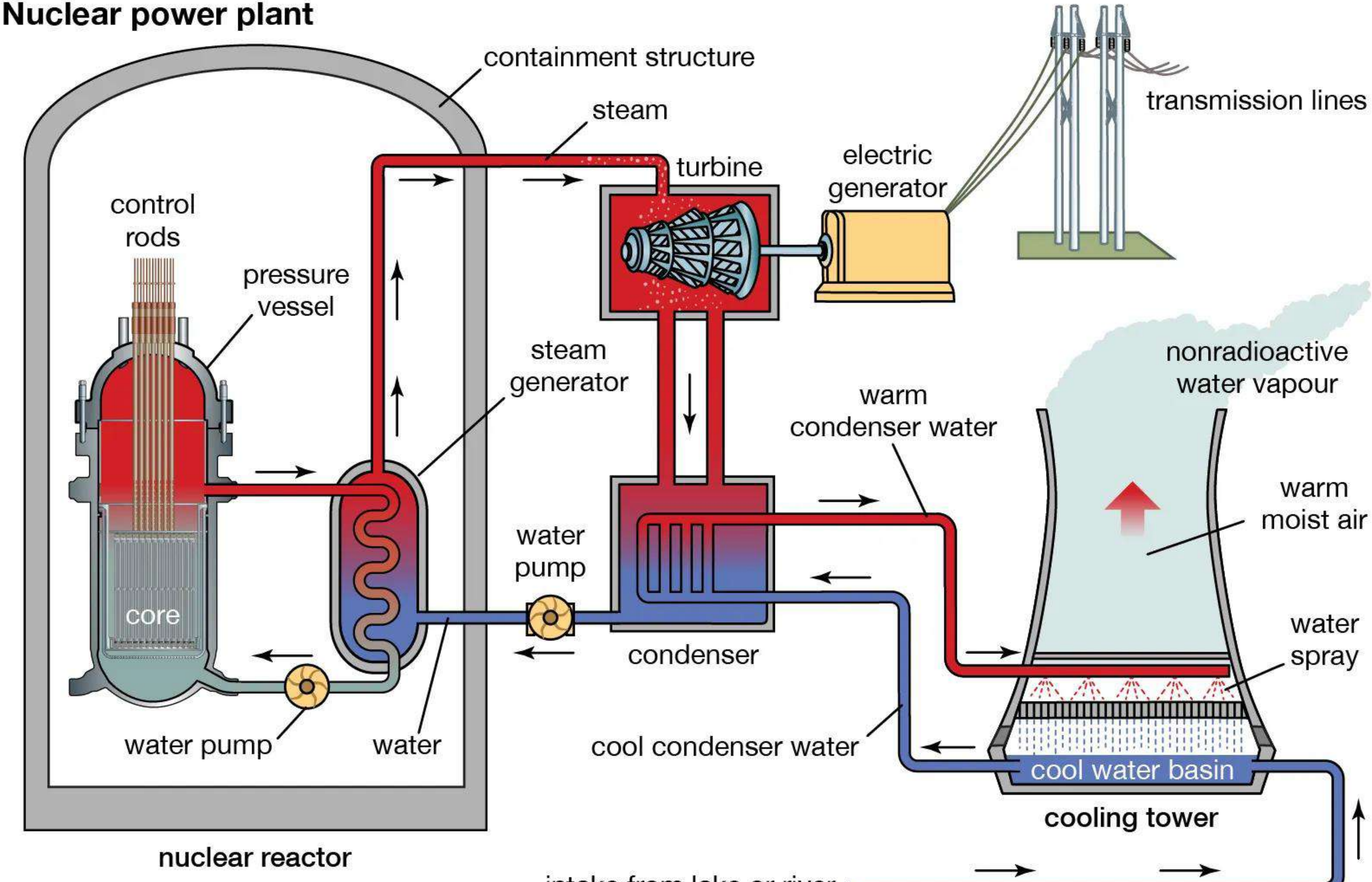

High Pressure Inlet
Upper Casing
Fixed Blade
Admission Section
Shaft Seal
Shaft Seal
Blade Wheel
Diaphragm Seal
Exhaust Section
Moving Blade
Lower Casing
Low Pressure Outlet

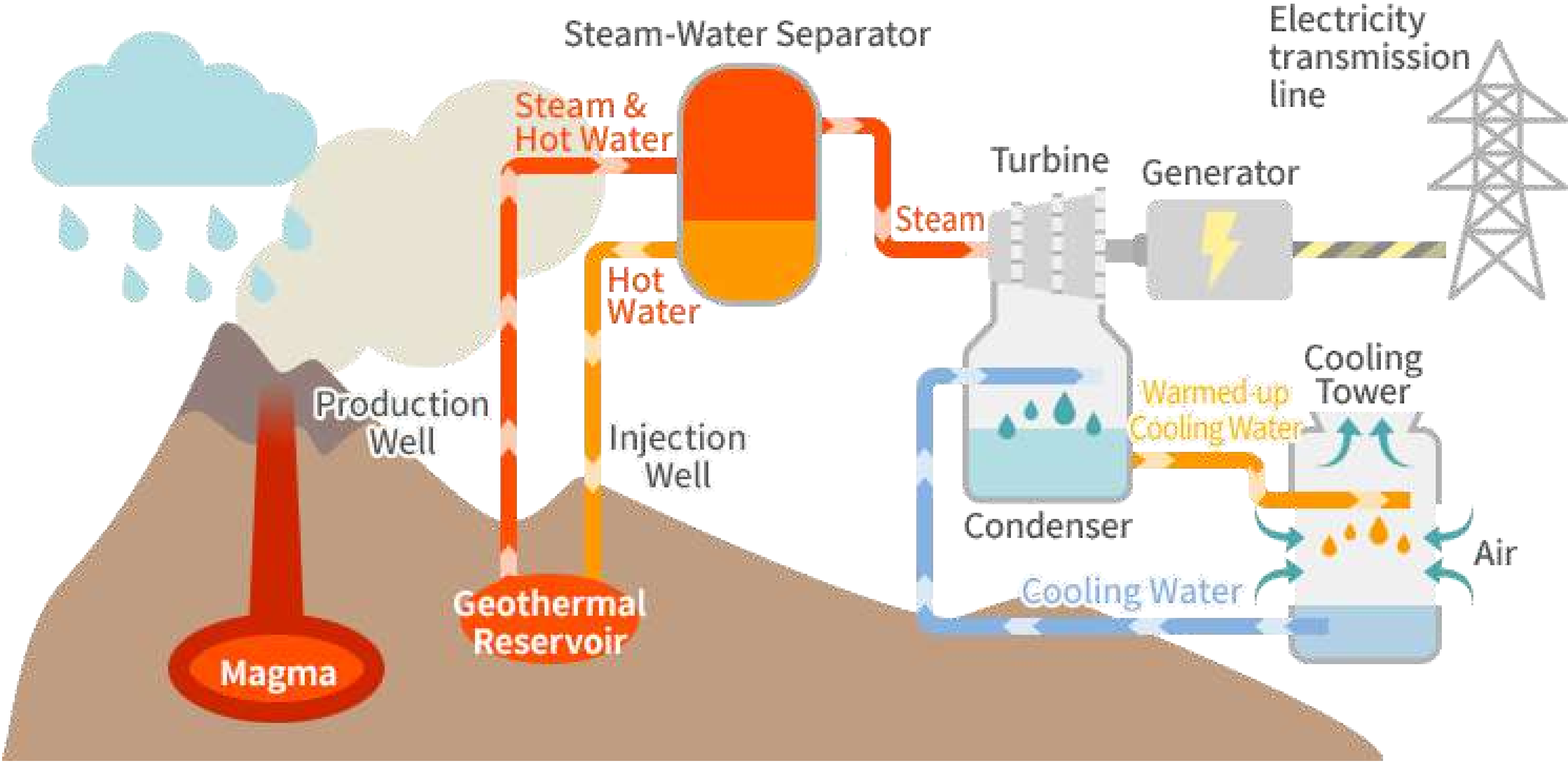


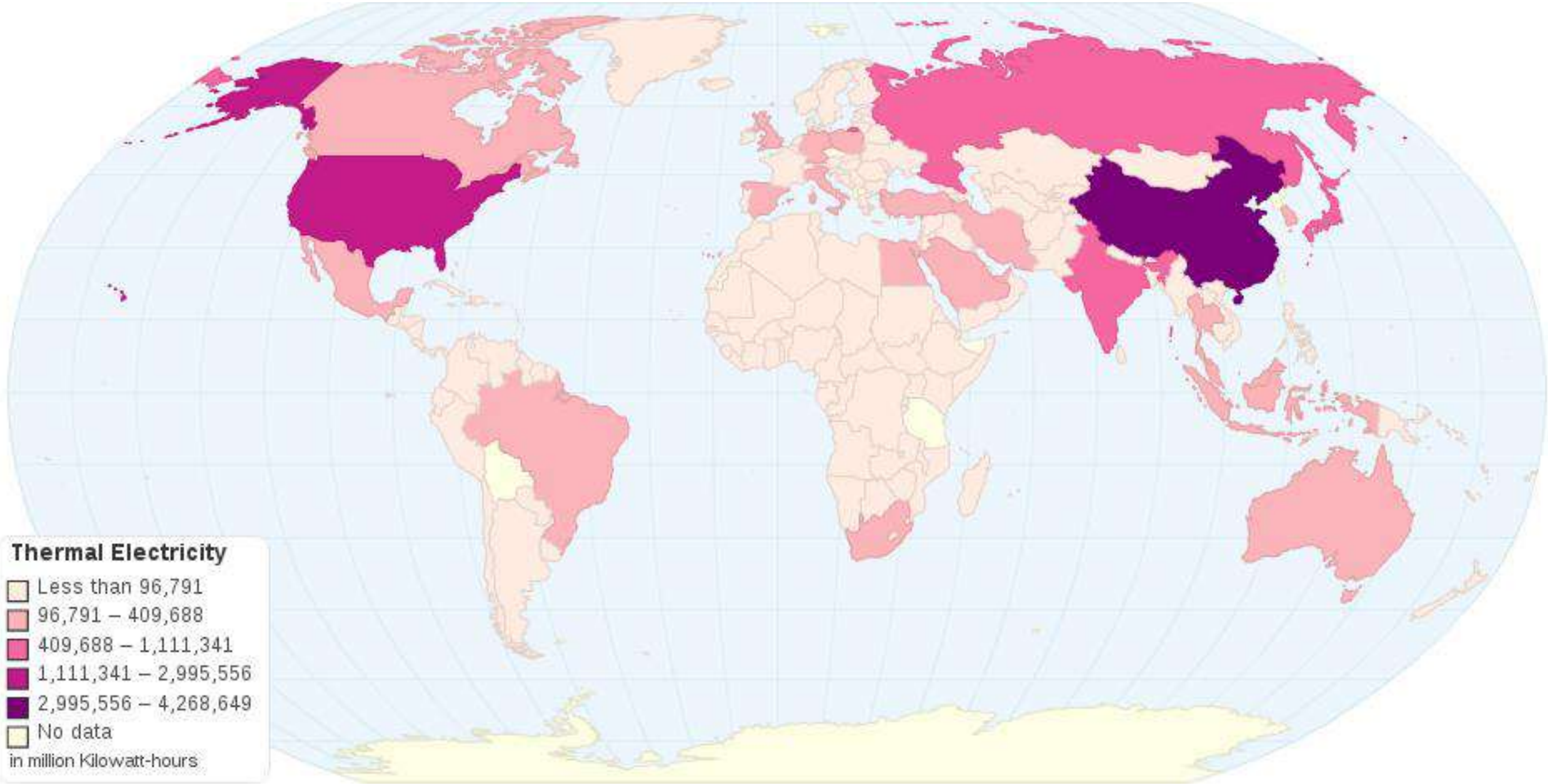

1 Blade
2 Inner housing
3 Outer housing
4 Shaft
5 Diaphragm
1
2
3
4
5



Nuclear power plant




Steam-Water Separator
Electricity transmission line
Steam & Hot Water
Turbine
Generator
Steam
Hot Water
Production Well
Injection Well
Cooling Tower
Warmed up Cooling Water
Condenser
Air
Cooling Water
Geothermal Reservoir
Magma


Thermal Electricity
Less than 96,791
96,791 – 409,688
409,688 – 1,111,341
1,111,341 – 2,995,556
2,995,556 – 4,268,649
No data
in million Kilowatt-hours

The below list shows the detailed classification of steam turbines

Based on stage design:

- Impulse
- Reaction
- Combination of both

Based on steam flow:

- Axial flow
- Radial flow
- Mixed flow

Based on stages:

- Single stage
- Multi stage

Based on exhaust condition:

- Condensing
- Non condensing
- Automatic extraction
- Mixed pressure
- Regenerative extraction
- Reheat

Based on type of governing:

1. Throttle governing
2. Nozzle governing
3. By-pass governing
4. Combination of 1&2 and 1&3

Based on casing or shaft design:

- Single casing
- Tandem casing (two or more)
- Cross compound

Based on exhaust stages in parallel:

- Double flow
- Triple flow

Based on type of drive:

- Generator
- Mechanical drive

Based on steam conditions at inlet to turbine :

(i) Low pressure turbines (1.2 to 2 ata)

(ii) Medium pressure turbines (upto 40 ata)

(iii) High pressure turbines (above 40 ata)

(iv) very high pressure turbines, (170 ata &550°C and higher)

(v) supercritical pressure turbines (225 ata and above)

Based on their usage in industry :

(i) Stationary turbines with constant speed of rotation primarily used for driving alternators.

(ii) Stationary steam turbines with variable speed meant for driving turbo-blowers, air circulators, pumps, etc.

(iii) Non-stationary turbines with variable speed ; turbines of this type are usually employed in steamers, ships, railway locomotives and rocket engines.

The following are the principal advantages of steam turbine over steam engines :

1. The thermal efficiency of a steam turbine is much higher than that of a steam engine.
2. The power generation in a steam turbine is at a uniform rate, therefore necessity to use a flywheel (as in the case of steam engine) is not felt.
3. Much higher speeds and greater range of speed is possible than in case of a steam engine.
4. In large thermal stations where we need higher outputs, the steam turbines prove very suitable as these can be made in big sizes.
5. With the absence of reciprocating parts (as in steam engine) the balancing problem is minimised.
6. No internal lubrication is required as there are no rubbing parts in the steam turbine.
7. In a steam turbine there is no loss due to initial condensation of steam.
8. It can utilise high vacuum very advantageously.
9. Considerable overloads can be carried at the expense of slight reduction in overall efficiency.

Steam turbines may be classified into different categories depending on their construction, working pressures, size and many other elements. But there are two basic types of steam turbines which are called **impulse** and **reaction** turbines. There are other types of steam turbines that are actually derivatives of these two main types. The main difference between these turbines lies in the way in which the steam is expanded while it moves through them.

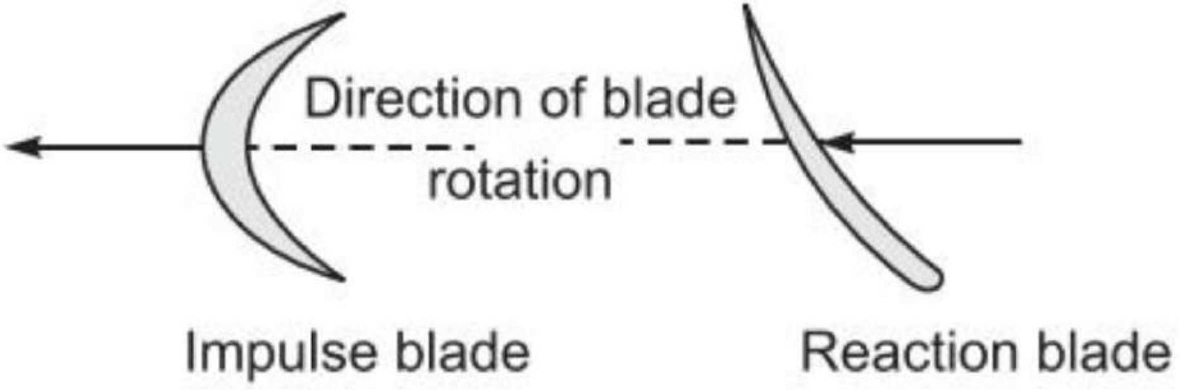
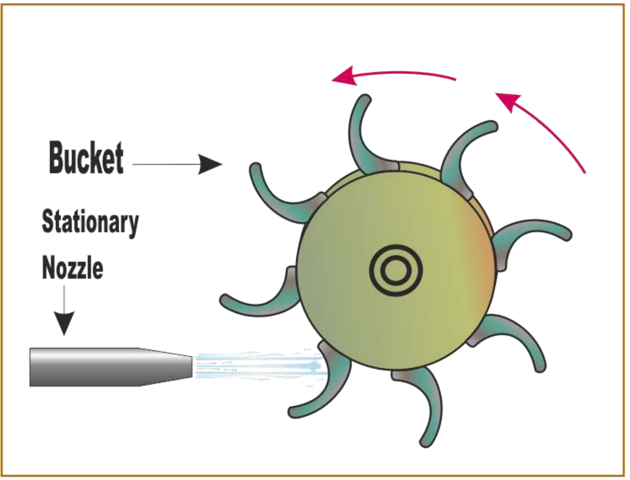
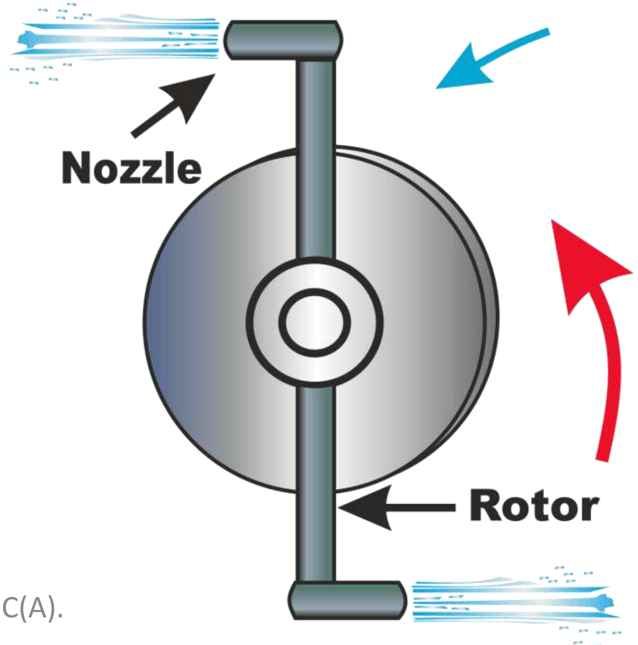
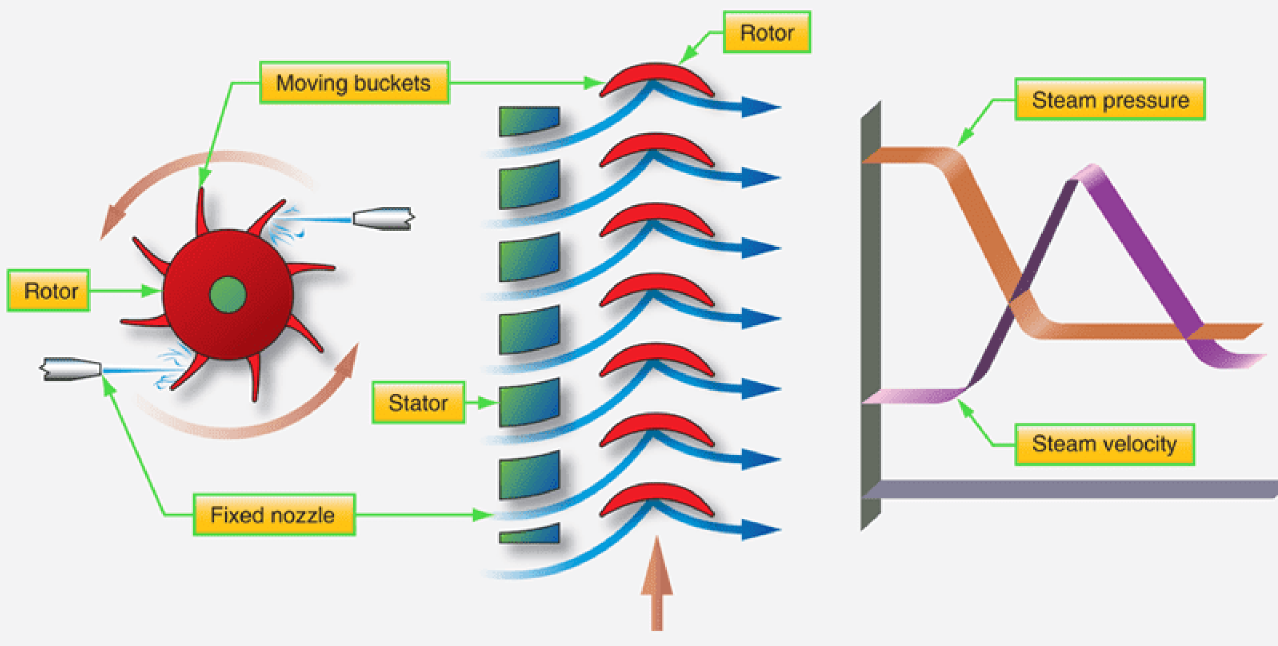


Fig. *Shape of blades in impulse and reaction turbine*

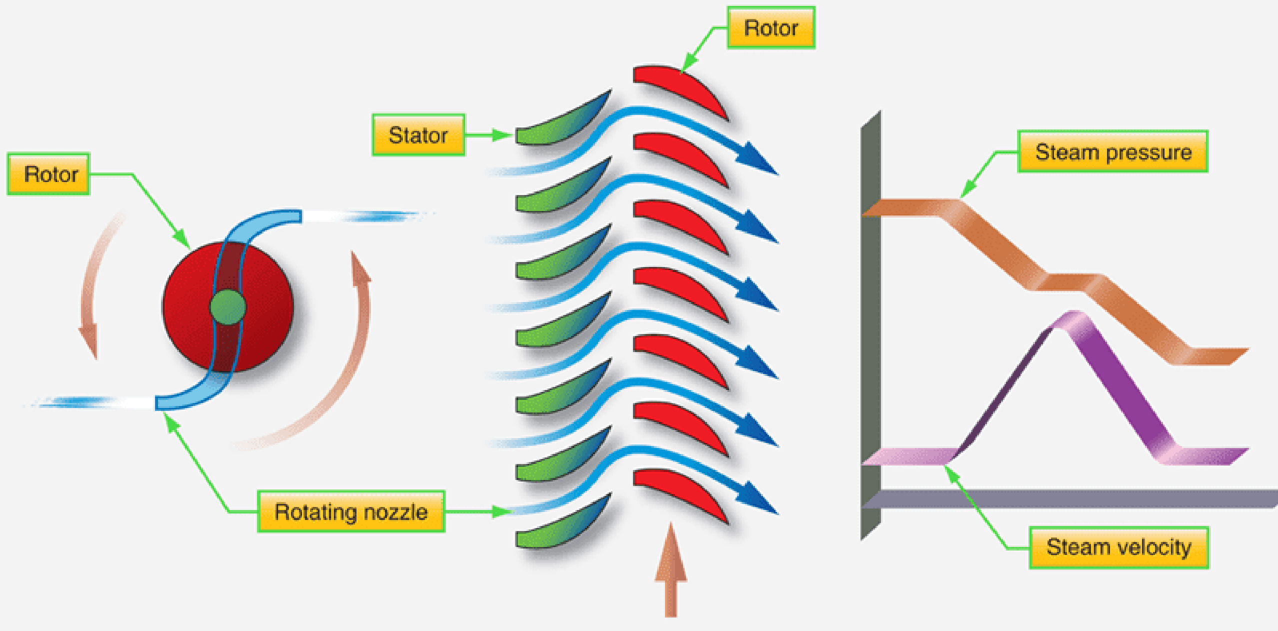
In a **Simple impulse turbine** steam expands in the nozzles and its pressure does not alter as it moves over the blades. while in the **Reaction** type the steam expands continuously as it passes over the blades and thus there is gradual fall in the pressure during expansion.







Impulse Turbine



Reaction Turbine

Simple impulse turbines

The top portion of the figure exhibits a longitudinal section through the upper half of the turbine, the middle portion shows one set of nozzles which is followed by a ring of moving blades, while lower part of the diagram indicates approximately changes in pressure and velocity during the flow of steam through the turbine. This turbine is called ‘simple’ impulse turbine since the expansion of the steam takes place in one set of the nozzles.

As the steam flows through the nozzle its pressure falls from steam chest pressure to condenser pressure (or atmospheric pressure if the turbine is non-condensing). Due to this relatively higher ratio of expansion of steam in the nozzles the steam leaves the nozzle with a very high velocity.

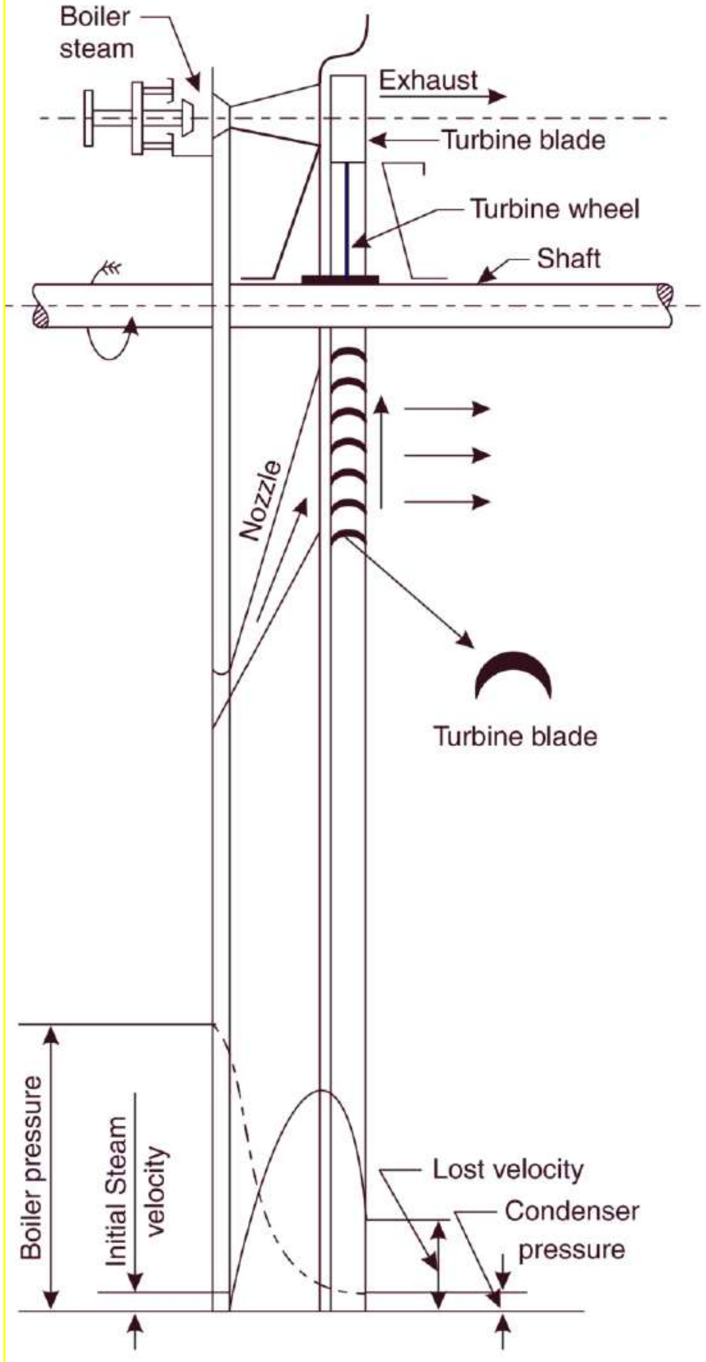
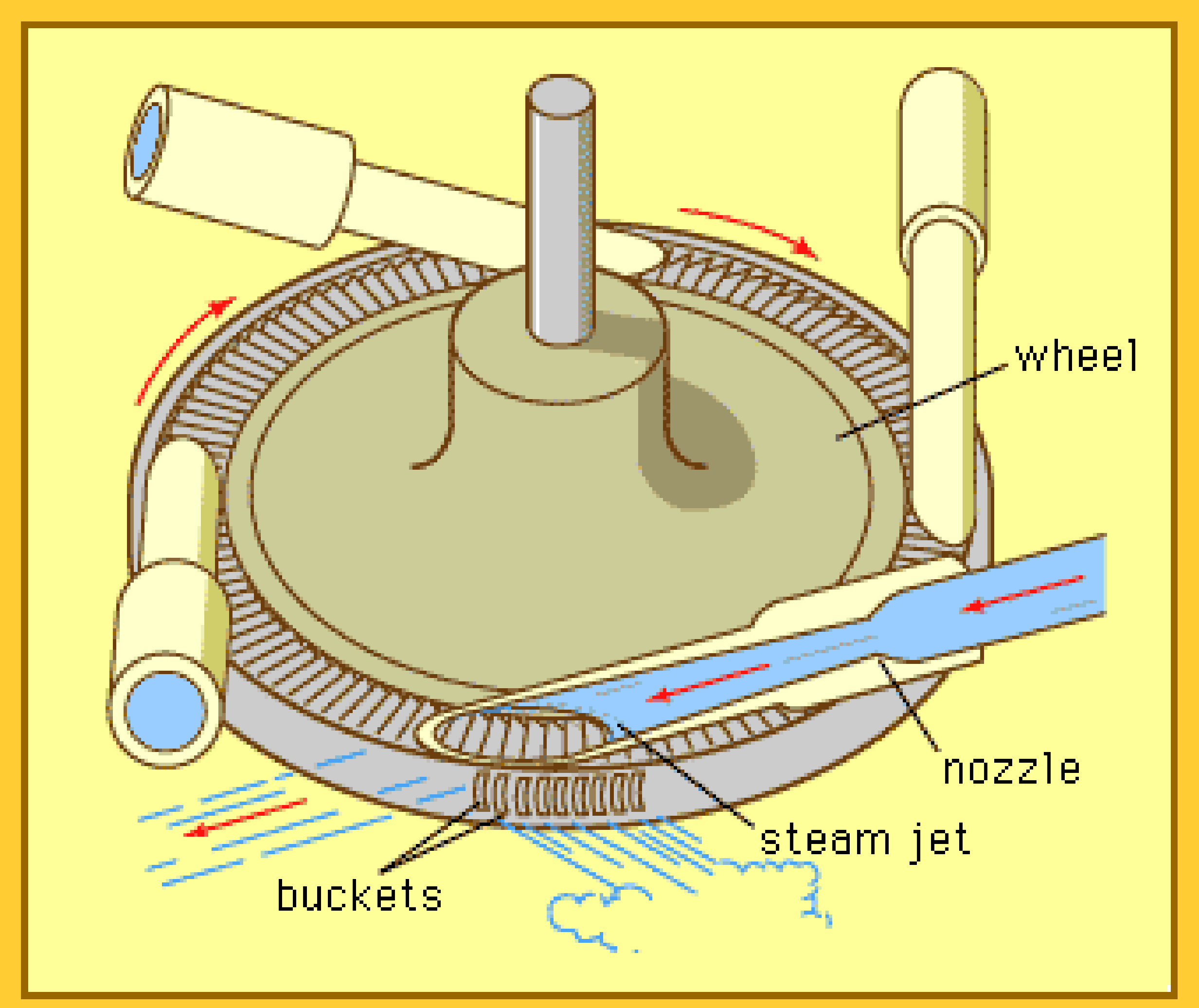


Fig. Simple impulse turbine.

Refer Fig. , it is evident that the velocity of the steam leaving the moving blades is a large portion of the maximum velocity of the steam when leaving the nozzle. The loss of energy due to this higher exit velocity is commonly called the **“carry over loss”** or **“leaving loss”.**

The principal example of this turbine is the well known **“De laval turbine”** and in this turbine the ‘exit velocity’ or ‘leaving velocity’ or ‘lost velocity’ may amount to 3.3 per cent of the nozzle outlet velocity. Also since all the kinetic energy is to be absorbed by one ring of the moving blades only, the velocity of wheel is too high (varying from 25000 to 30000 r.p.m.). This wheel or rotor speed however, can be reduced by different methods.



Reaction turbine

In this type of turbine, there is a gradual pressure drop and takes place continuously over the fixed and moving blades. The function of the fixed blades is (the same as the nozzle) that they alter the direction of the steam as well as allow it expand to a larger velocity. As the steam passes over the moving blades its kinetic energy (obtained due to fall in pressure) is absorbed by them. Fig. shows a three stage reaction turbine. The changes in pressure and velocity are also shown there in.

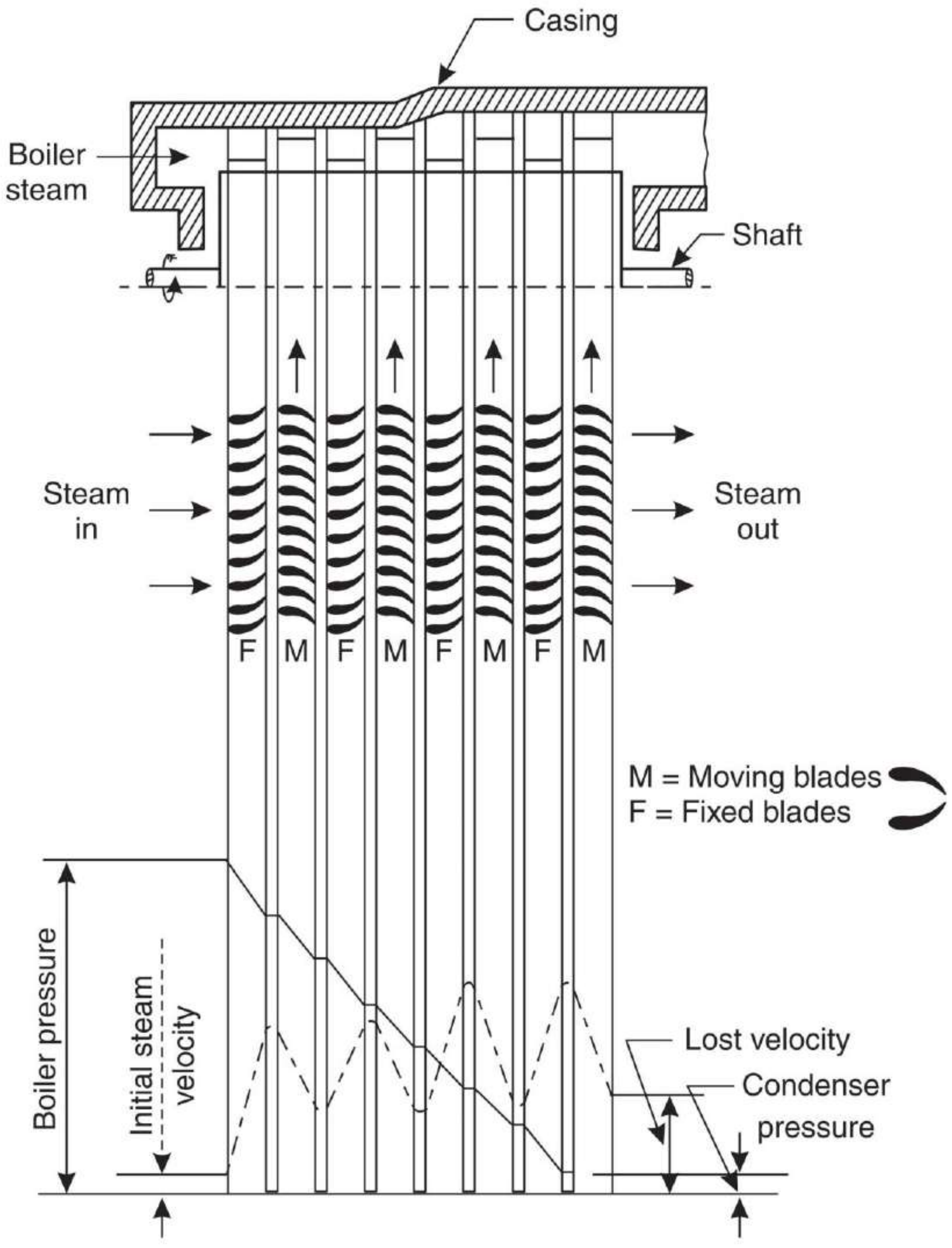
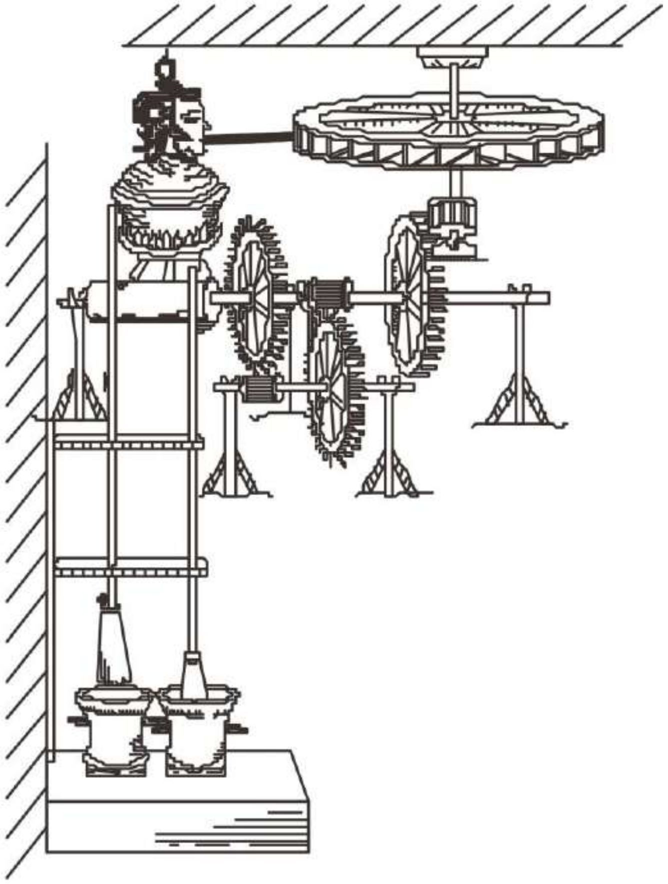


Fig. Reaction turbine (three stage).

As the volume of steam increases at lower pressures therefore, the diameter of the turbine must increase after each group of blade rings. It may be noted that in this turbine since the pressure drop per stage is small, therefore the number of stages required is much higher than an impulse turbine of the same capacity.

The reaction turbines which are used these days are really impulse-reaction turbine. Pure reaction turbines are not in general use. The expansion of steam and heat drop occur both in fixed and moving blades



Branca's impulse turbine

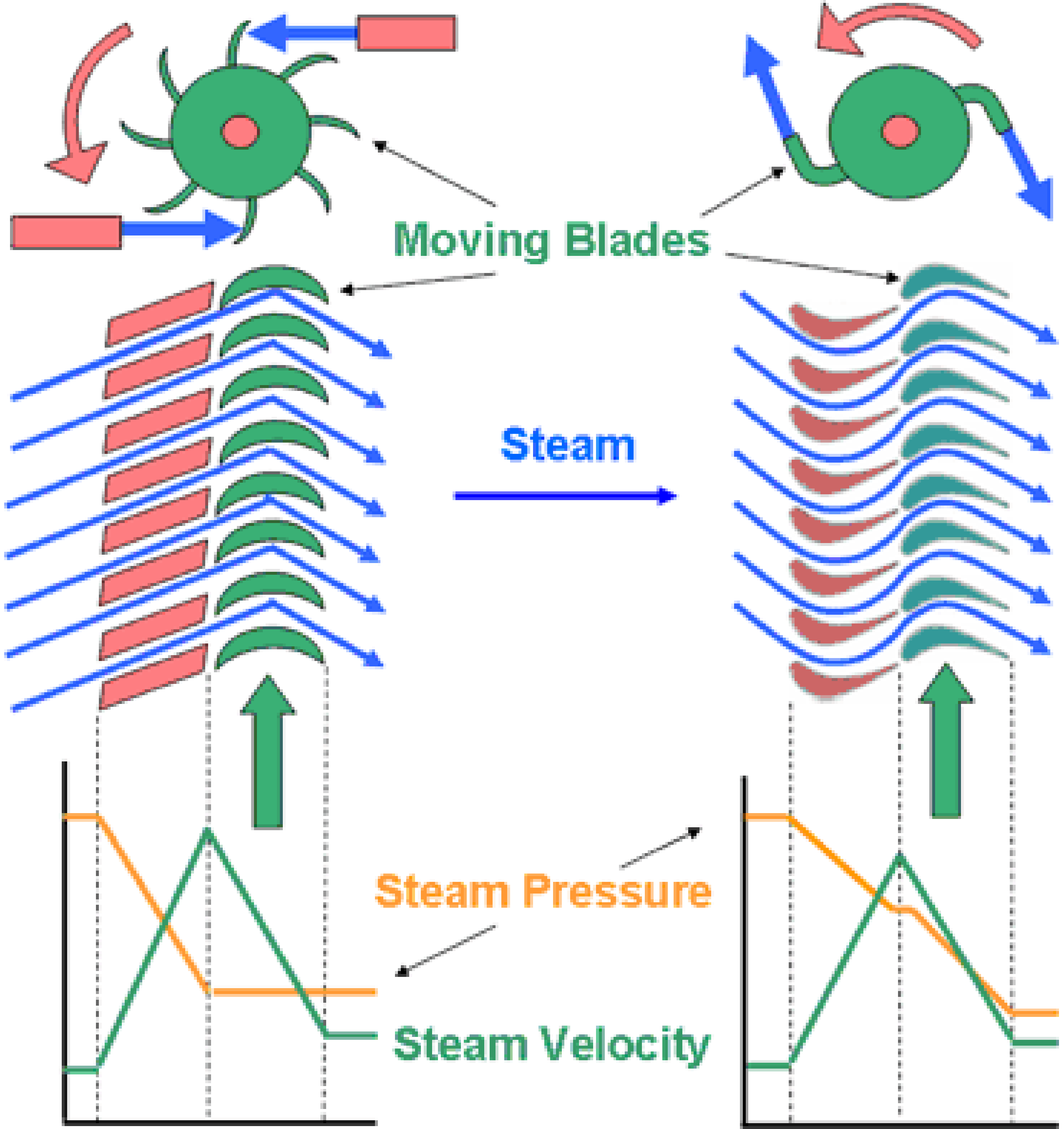
G.Branca developed the first impulse turbine in 1629 AD

Fig. *Hero's turbine*

The first steam turbine was made by Hero of Alexandria, 2000 years ago. This turbine worked on the pure reaction principle and no power was produced by it

Impulse Turbine

Reaction Turbine



Moving Blades

Steam

Steam Pressure

Steam Velocity

S. No.	Particulars	Impulse turbine	Reaction turbine
1.	*Pressure drop*	Only in nozzles and not in moving blades.	In fixed blades (nozzles) as well as in moving blades.
2.	*Area of blade channels*	Constant.	Varying (converging type).
3.	*Blades*	Profile type.	Aerofoil type.
4.	*Admission of steam*	Not all round or complete.	All round or complete.
5.	*Nozzles / fixed blades*	Diaphram contains the nozzle.	Fixed blades similar to moving blades attached to the casing serve as nozzles and guide the steam.
6.	*Power*	Not much power can be developed.	Much power can be developed.
7.	*Space*	Requires less space for same power.	Requires more space for same power.
8.	*Efficiency*	Low.	High.
9.	*Suitability*	Suitable for small power requirements.	Suitable for medium and higher power requirements.
10.	*Blade manufacture*	Not difficult.	Difficult.

METHODS OF REDUCING WHEEL OR ROTOR SPEED

As already discussed under the heading ‘simple impulse turbine’ that if the steam is expanded from the boiler pressure to condenser pressure in one stage the speed of the rotor becomes tremendously high which crops up practical complicacies. There are several methods of reducing this speed to lower value ; all these methods utilize a multiple system of rotor in series, keyed on a common shaft and the steam pressure or jet velocity is absorbed in stages as the steam flows over the blades. This is known as **‘compounding’**.

The different methods of compounding are :

1. Velocity compounding.
2. Pressure compounding.
3. Pressure velocity compounding.
4. Reaction turbine.

1.Velocity compounding

This velocity compounded impulse turbine (*Curtis Turbine*) was developed by an American engineer *CG Curtis* in year 1985. Steam is expanded through a stationary nozzle from the boiler or inlet pressure to condenser pressure. So the pressure in the nozzle drops, the kinetic energy of the steam increases due to increase in velocity. A portion of this available energy is absorbed by a row of moving blades. The steam (whose velocity has decreased while moving over the moving blades) then flows through the second row of blades which are fixed. The function of these fixed blades is to re-direct the steam flow without altering its velocity to the following next row moving blades where again work is done on them and steam leaves the turbine with a low velocity. Fig. shows a cut away section of such a stage and changes in pressure and velocity as the steam passes through the nozzle, fixed and moving blades.

Though this method has the advantage that the initial cost is low due to lesser number of stages yet its efficiency is low.

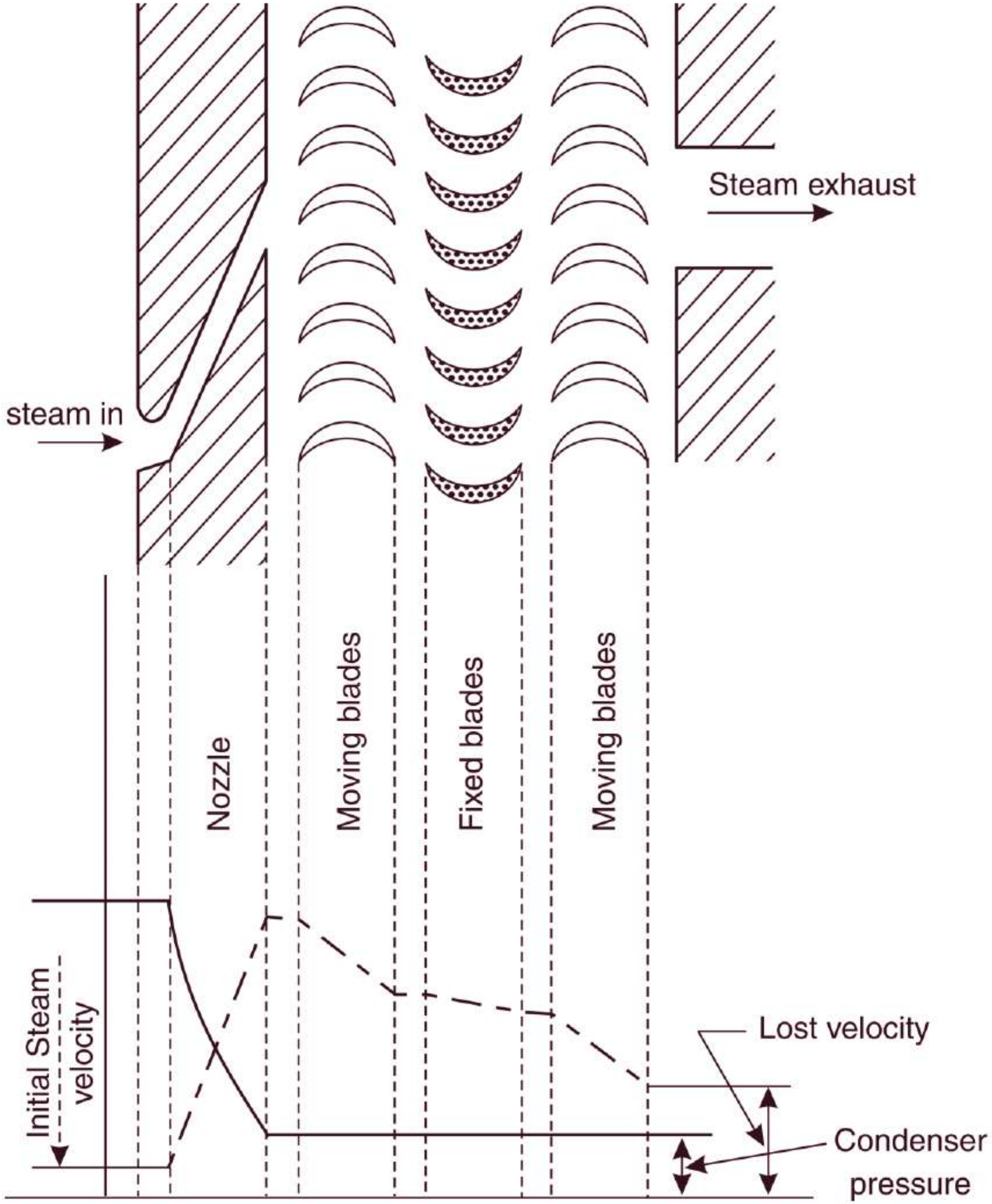
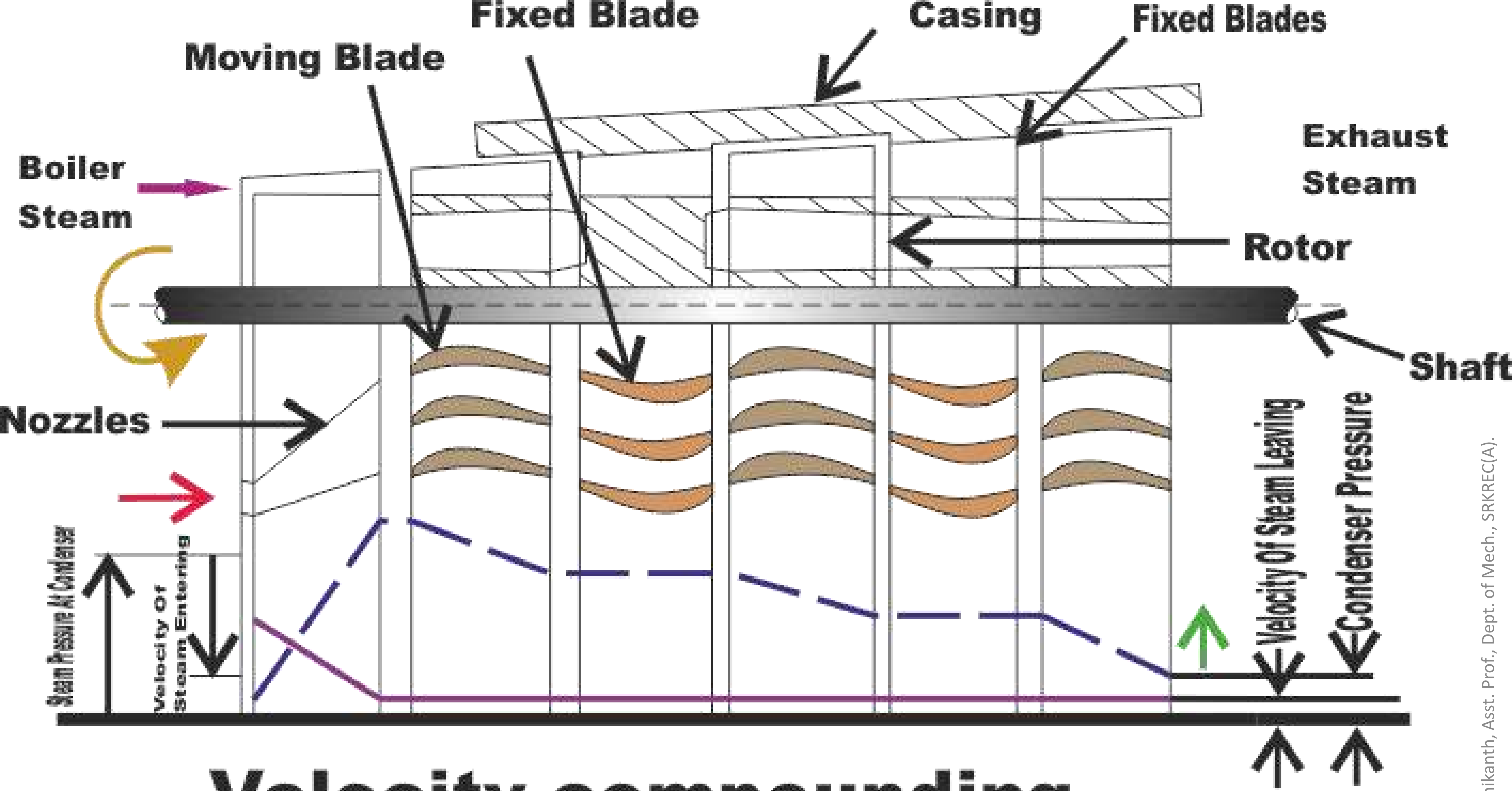


Fig. Velocity compounding.



Velocity compounding

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Advantages of Velocity Compounded Impulse Turbine

1. Owing to relatively large heat drop, a velocity-compounded impulse turbine requires a comparatively small number of stages.

2. Due to number of stages being small, its cost is less.

3. As the number of moving blades’ rows in a wheel increases, the maximum stage efficiency and optimum value of ρ decreases.

4. Since the steam temperature is sufficiently low in a two or three row wheel, therefore, cast iron cylinder may be used. This will cause saving in material cost.

Disadvantages of velocity-compounded impulse turbine :

1. It has high steam consumption and low efficiency.

2. In a single row wheel, the steam temperature is high so cast-iron cylinder cannot be used due to phenomenon of growth ; cast steel cylinder is used which is costlier than cost iron.

2. Pressure compounding

The Fig. shows rings of fixed nozzles incorporated between the rings of moving blades. The steam enters the first set of nozzles and expands partially. The kinetic energy of the steam thus obtained is absorbed by the moving blades (stage 1) and expands partially in the second set of nozzles where its pressure again falls and the velocity increases ; the kinetic energy so obtained is absorbed by the second ring of moving blades (stage 2). This is repeated in stage 3 and steam finally leaves the turbine at low velocity and pressure. The number of stages (or pressure reductions) depends on the number of rows of nozzles through which the steam must pass. This method of compounding is used in Rateau & Zoelly turbine.

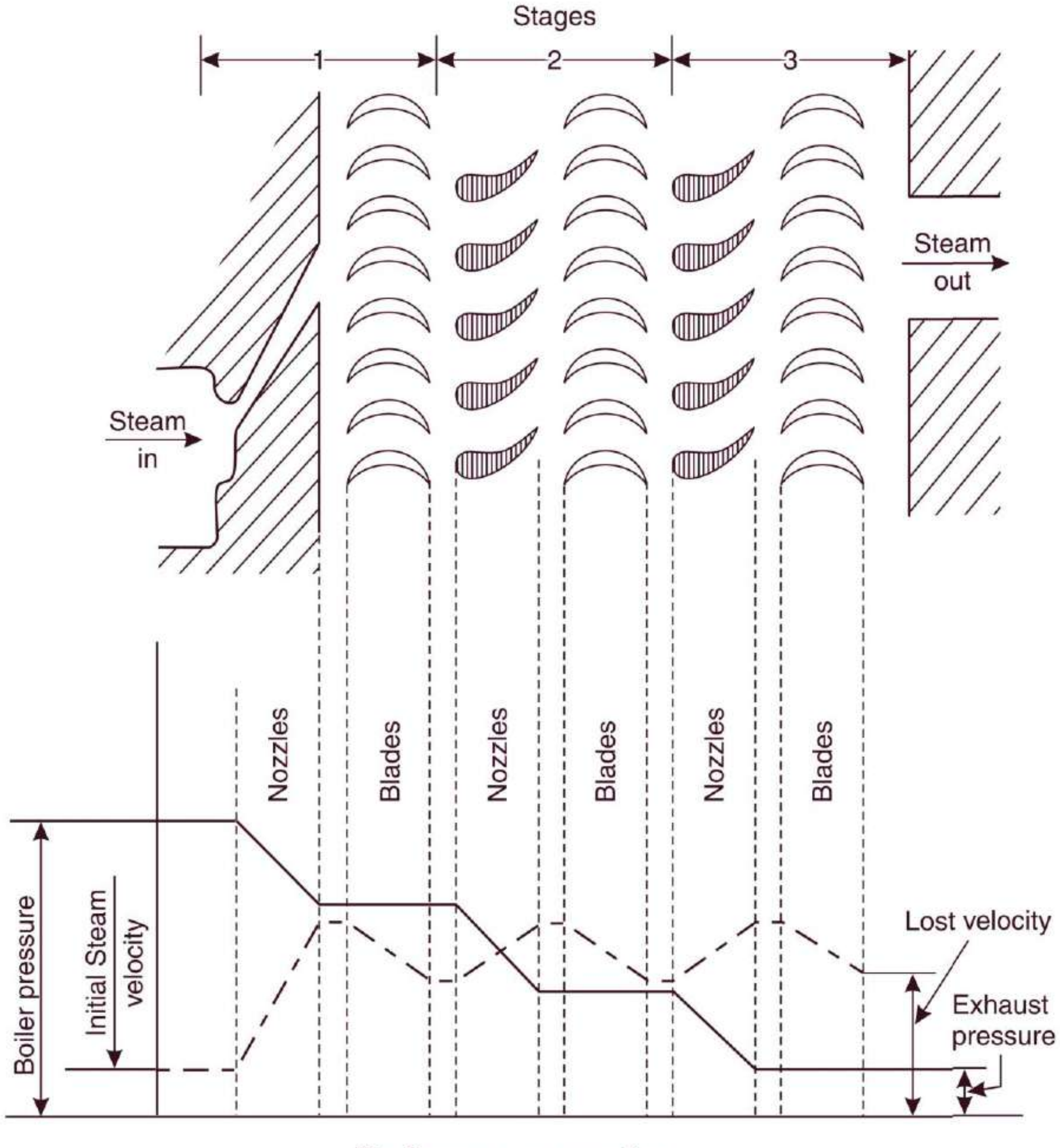
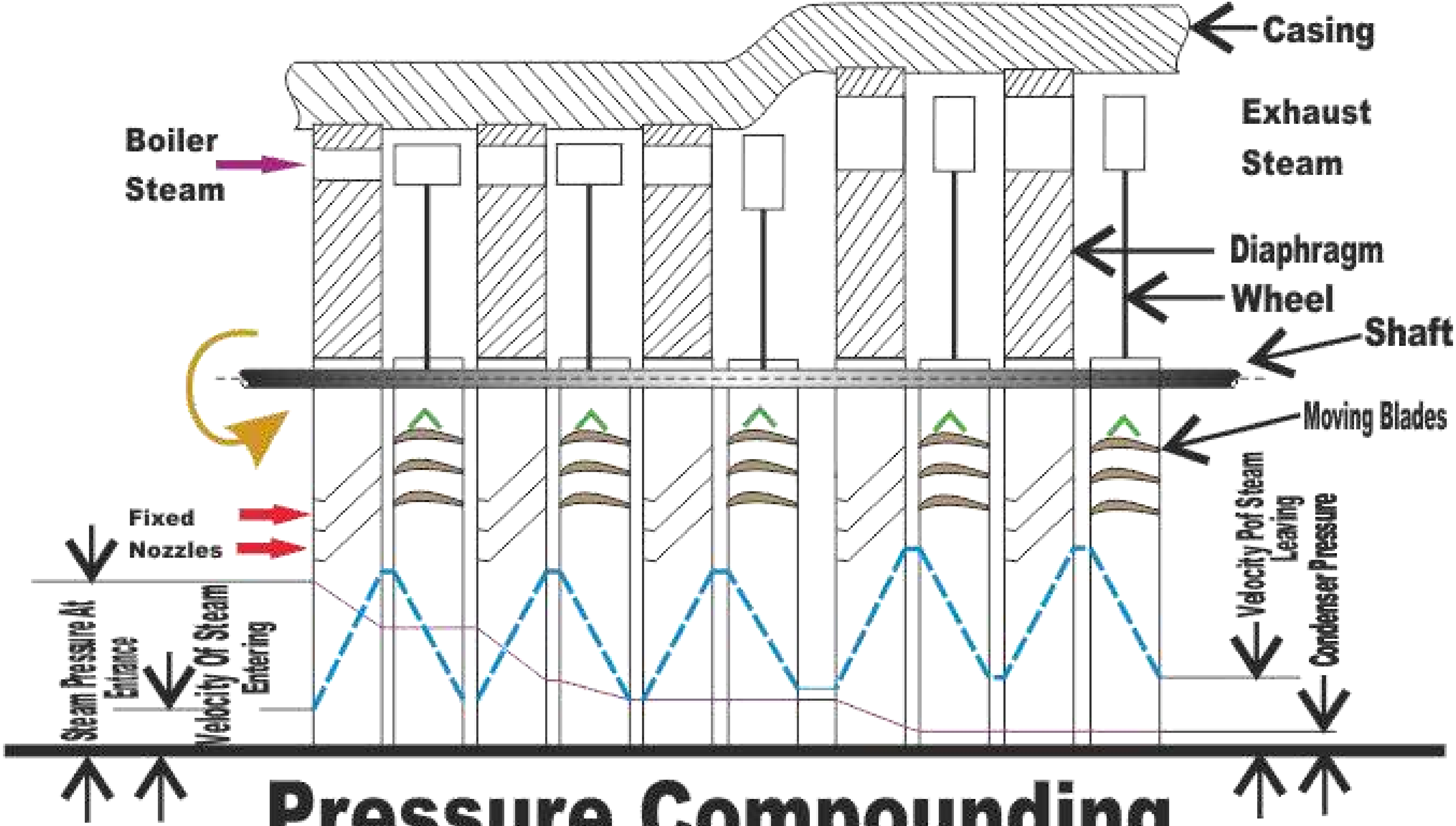


Fig. Pressure compounding.

This is most efficient turbine since the speed ratio remains constant but it is expensive owing to a large number of stages.



Pressure Compounding

3. Pressure-velocity compounding

This is a combination of pressure and velocity compoundings. The total pressure drop of steam is divided into a number of stages as done in pressure compounding. Each stage has a number of rows of fixed and moving blades working as an independent velocity compounded stage. Each stage is separated from the adjacent stage by a row of stationary ring of nozzles for expansion of steam for the next stage. The set of moving and fixed blades is used for velocity compounding and a set of nozzle rings in between stages is utilized for pressure compounding. Such type of compounding offers a larger pressure drop in each stage with less number of stages. Therefore, the turbine is simple and compact.

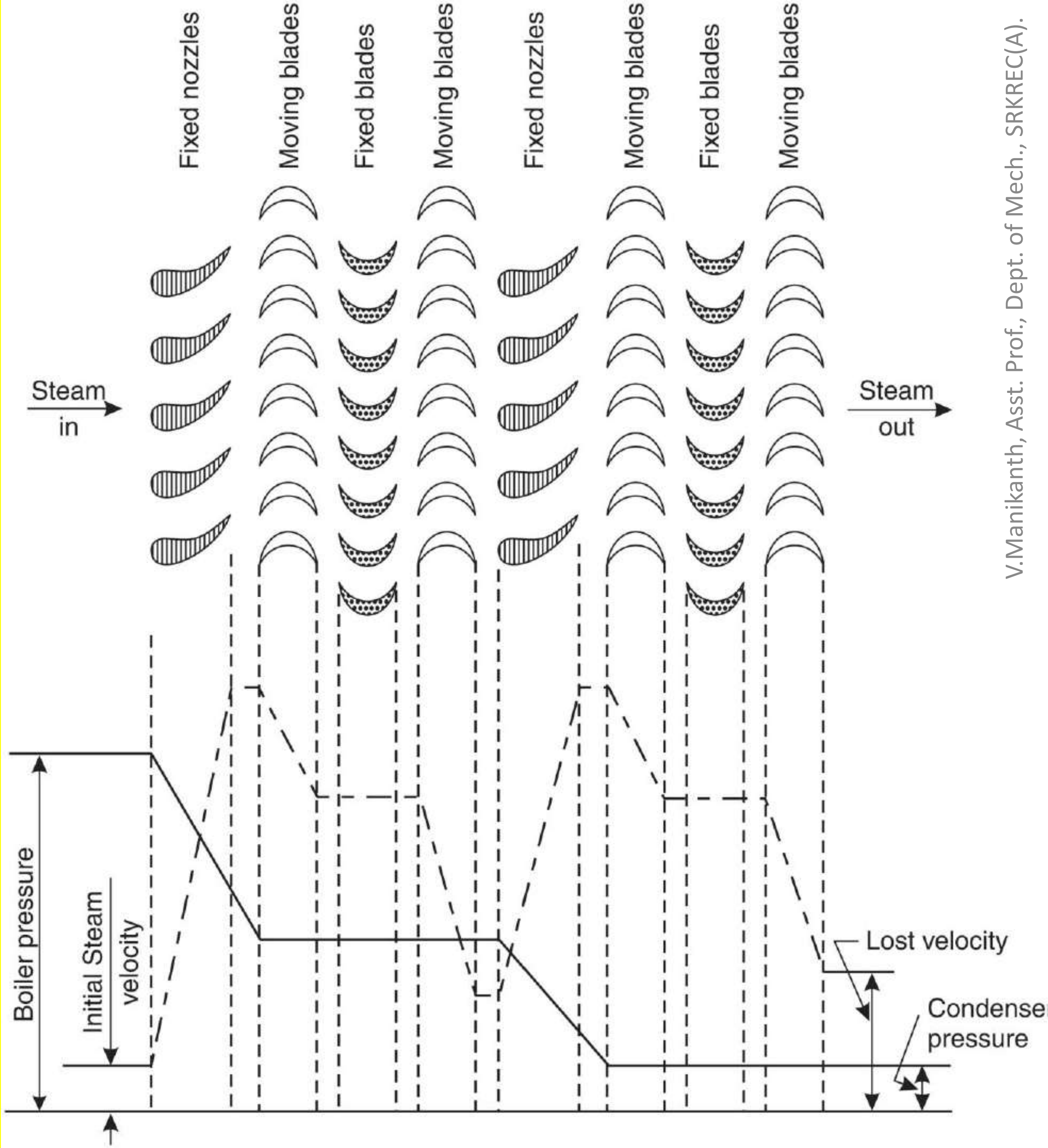
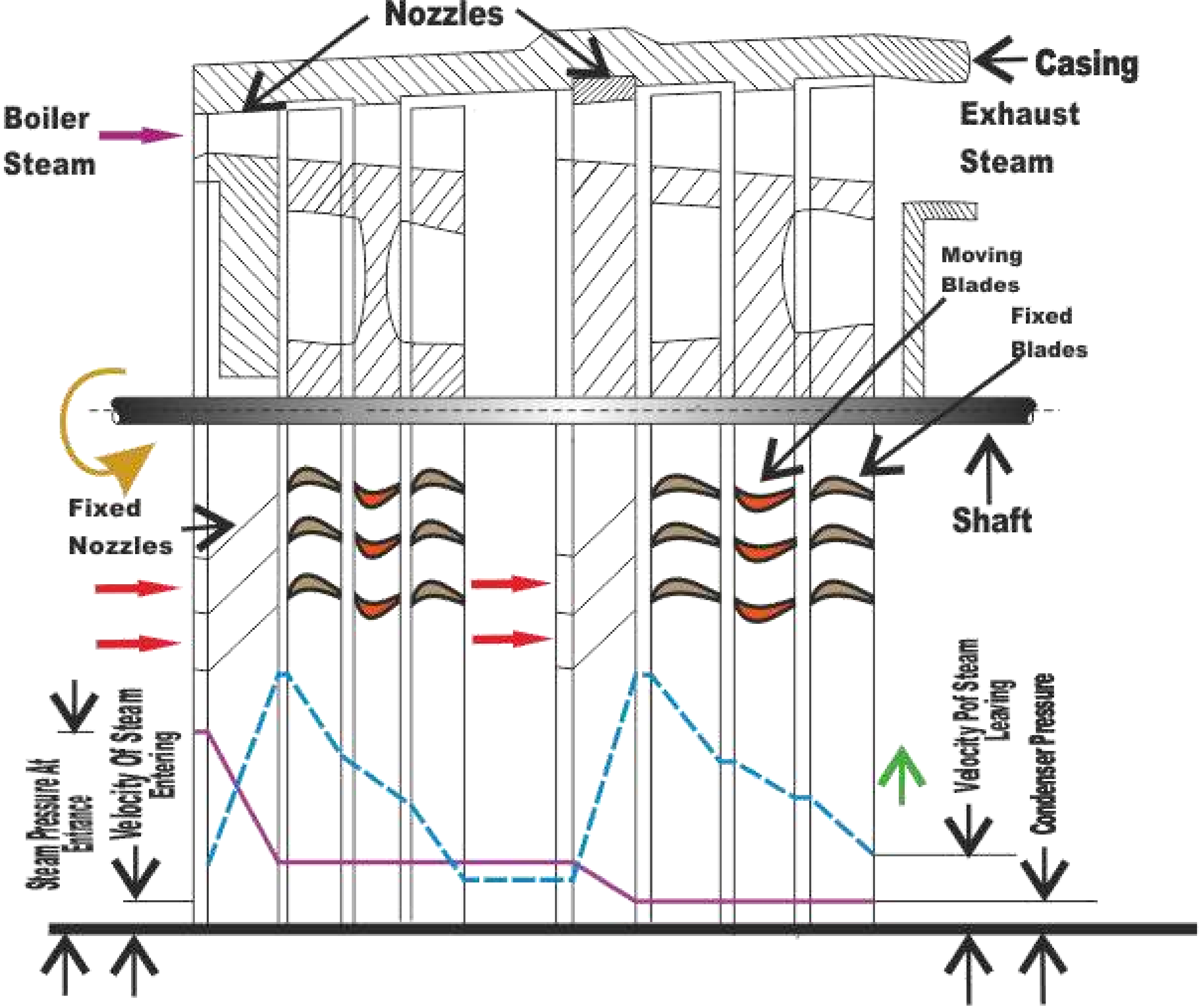


Fig. Pressure velocity compounding.



Pressure Velocity Compounding

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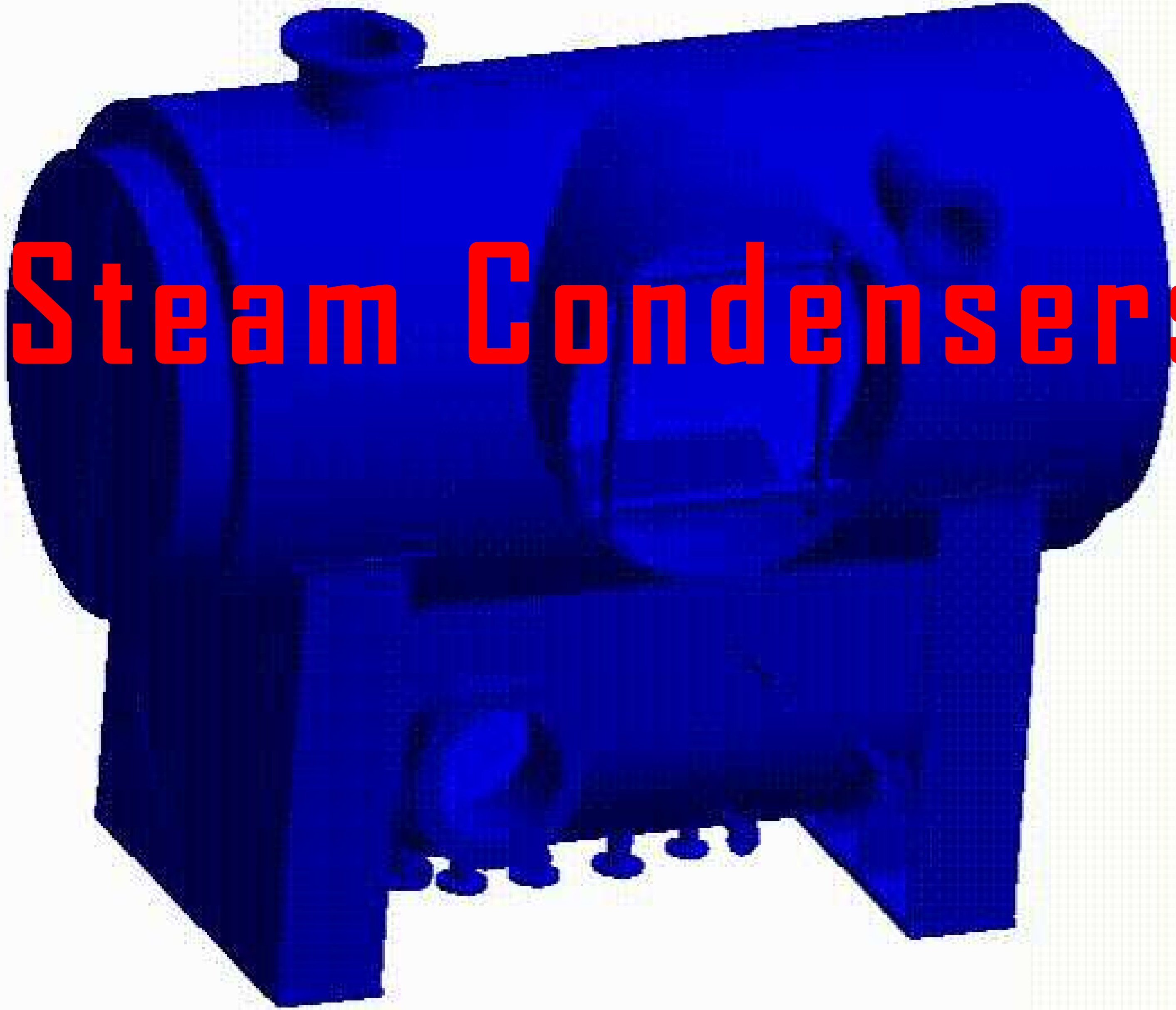
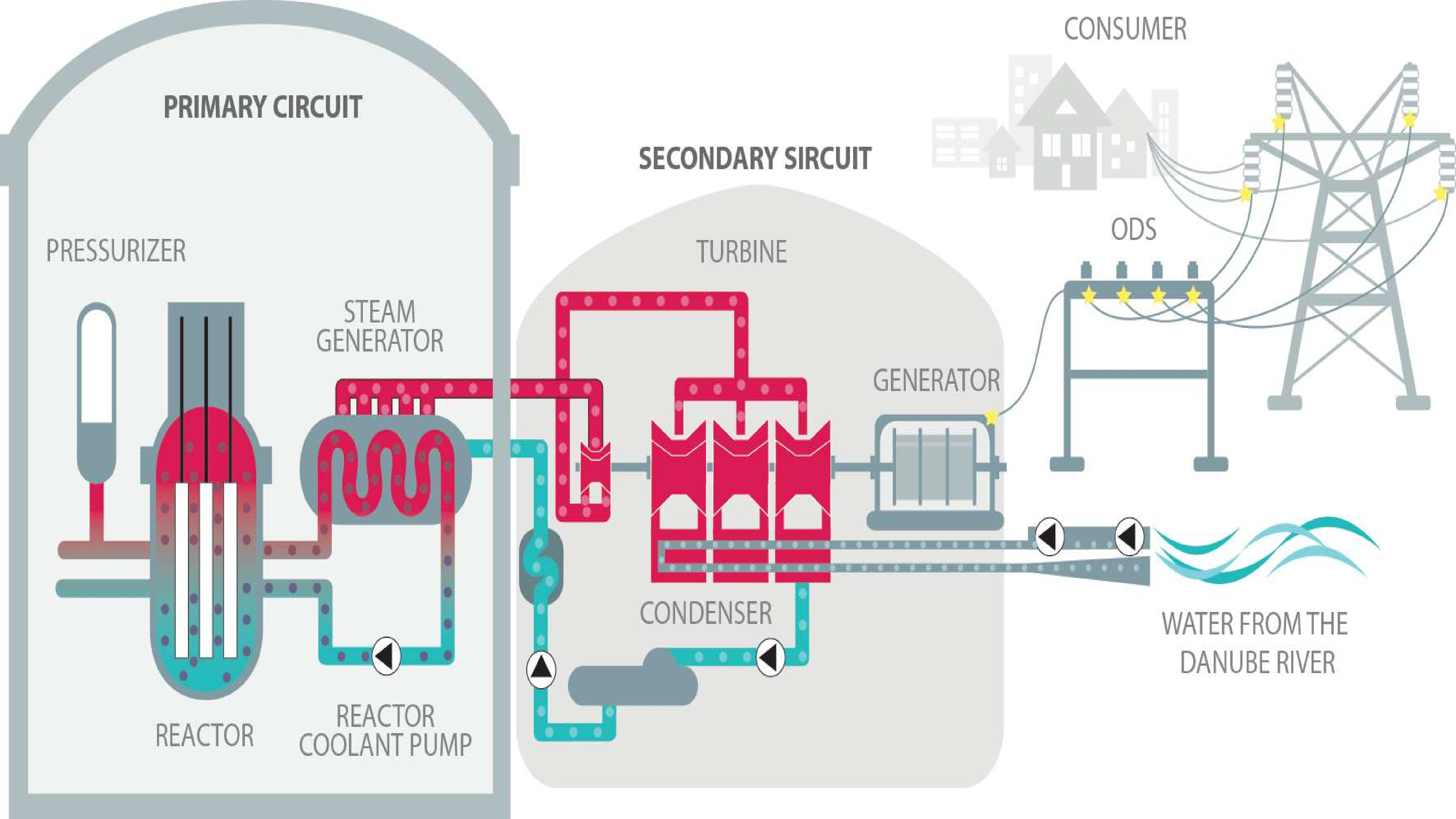
Steam Condensers

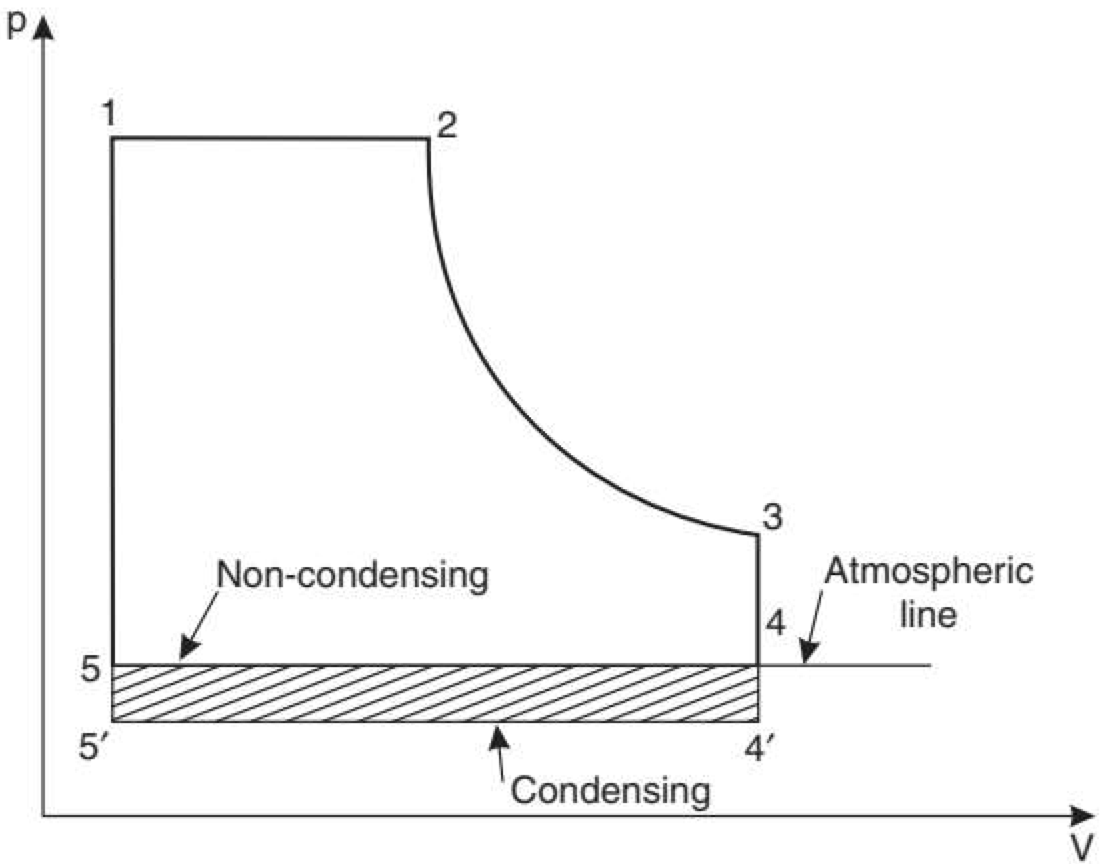



CONSUMER
PRIMARY CIRCUIT
SECONDARY SIRCUIT
ODS
PRESSURIZER
TURBINE
STEAM GENERATOR
GENERATOR
CONDENSER
WATER FROM THE DANUBE RIVER
REACTOR
REACTOR COOLANT PUMP

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

1. It *maintains a very low back pressure* on the exhaust side of the piston of the steam engine or turbine.

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



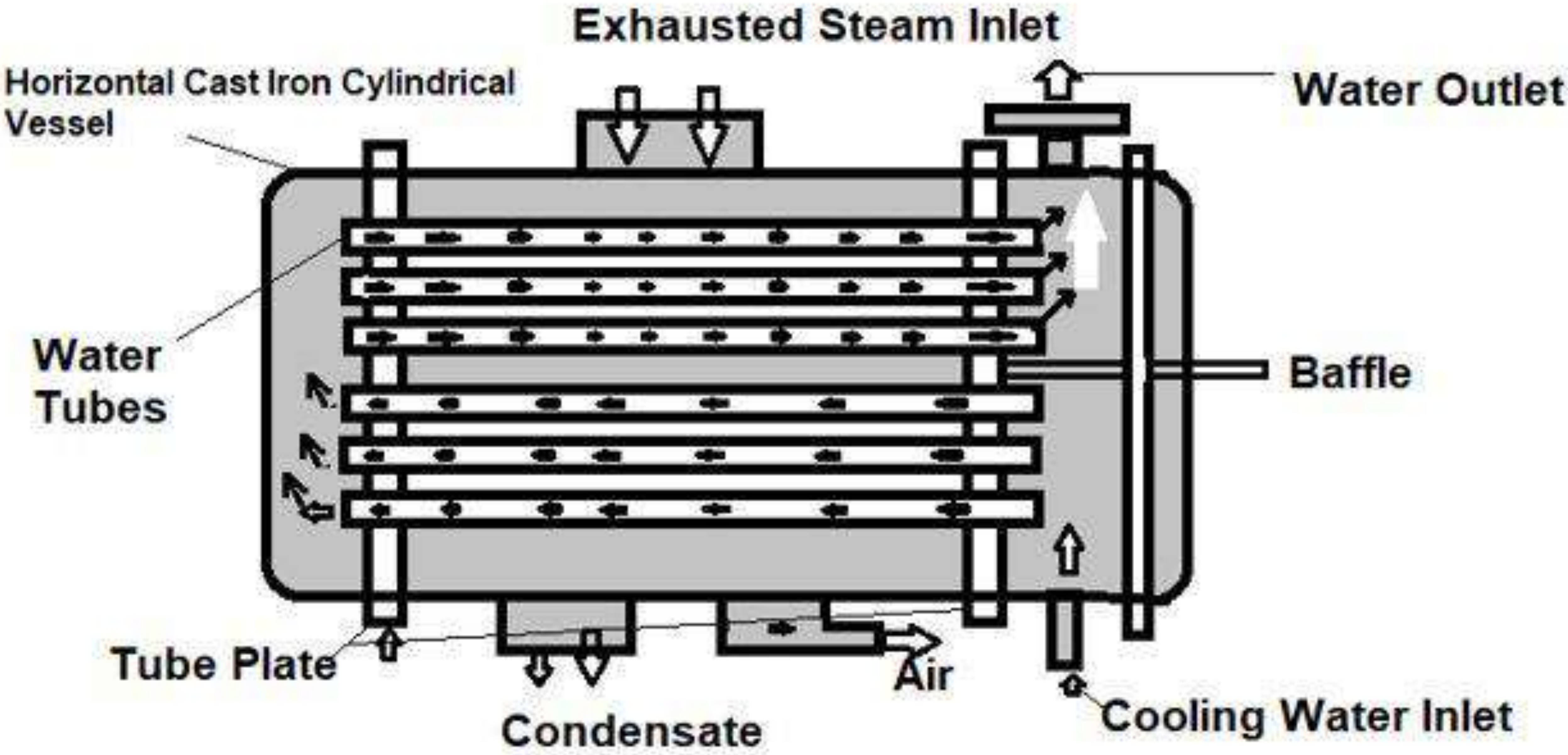
2. VACUUM

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

3. ORGANS OF A STEAM CONDENSING PLANT

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

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



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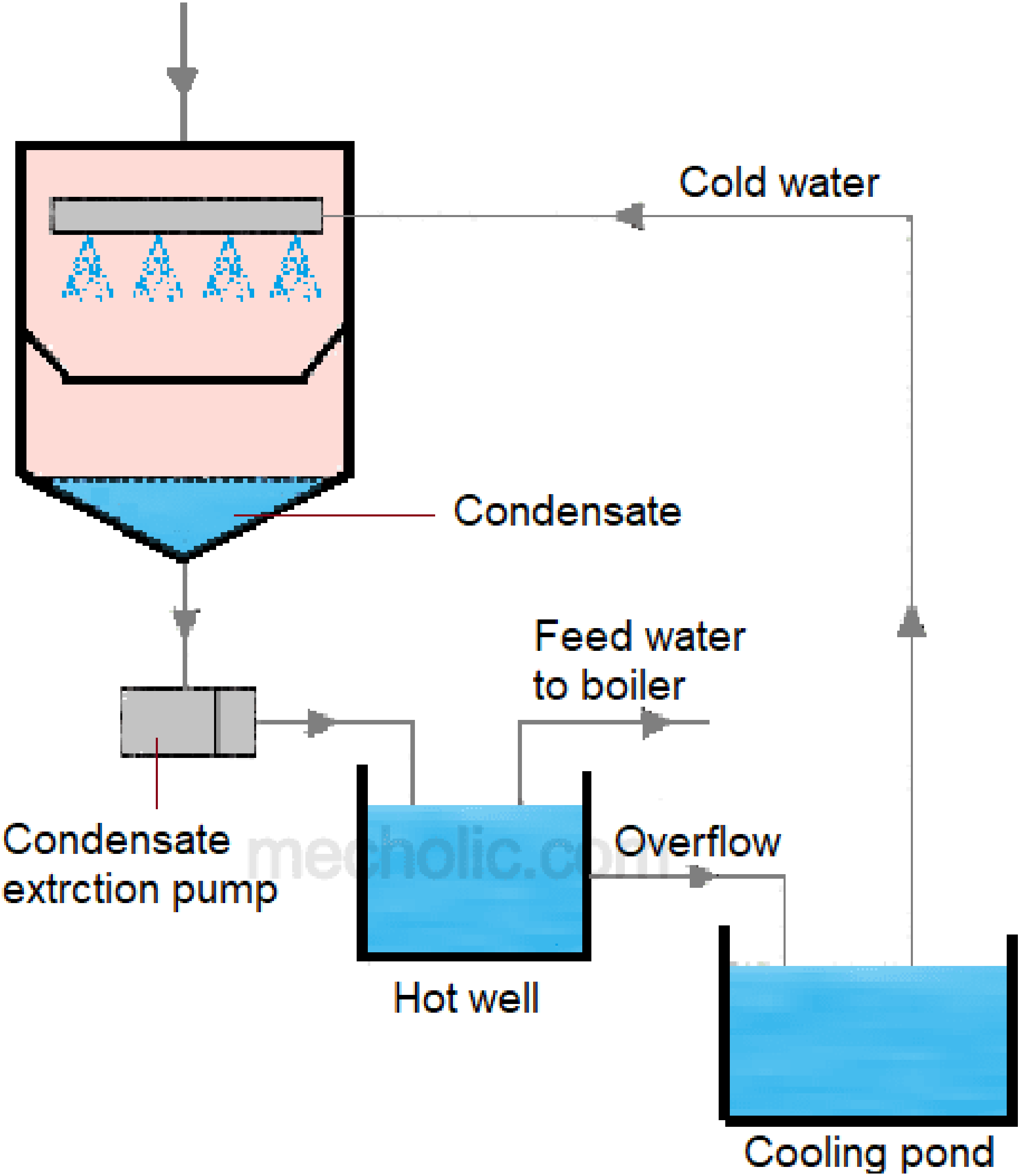

Cold water
Condensate
Feed water
to boiler
Condensate
extrction pump
Overflow
Hot well
Cooling pond

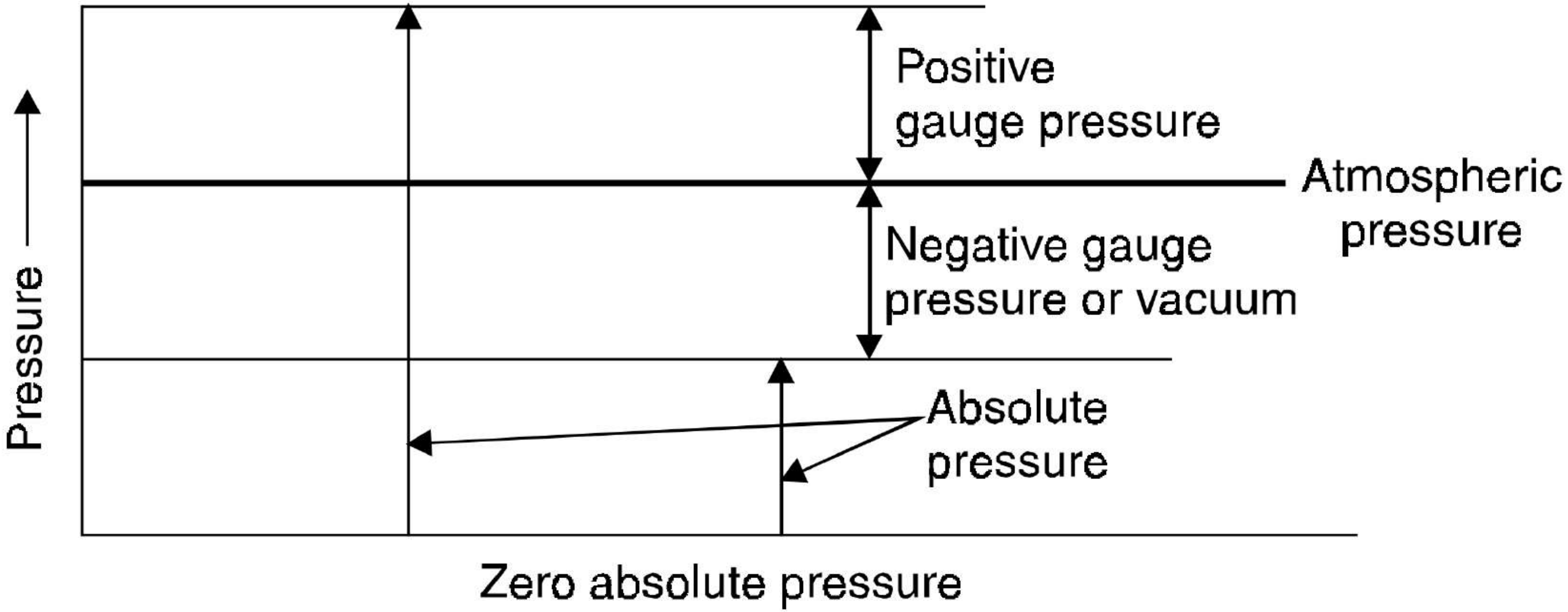


Fig. Schematic diagram showing gauge, vacuum and absolute pressures.

(*i*) Absolute pressure = Atmospheric pressure + Gauge pressure

$$p_{\text{abs.}} = p_{\text{atm.}} + p_{\text{gauge}}.$$

(*ii*) Vacuum pressure = Atmospheric pressure – Absolute pressure.

Standard atmospheric pressure = 1.01325 bar = 0.76 m (or 760 mm) Hg.

$$\text{Vacuum efficiency} = \frac{\text{Actual vacuum}}{\text{Maximum obtainable vacuum}}$$

$$= \frac{\text{Actual vacuum}}{\text{Barometer pressure} - \text{Absolute pressure of steam}}$$

$$\text{Condenser efficiency} = \frac{\text{Rise in temperature of cooling water}}{\begin{bmatrix}\text{Temp. corresponding to vacuum} \\ \text{in the condenser}\end{bmatrix} - \begin{bmatrix}\text{Inlet temp. of} \\ \text{cooling water}\end{bmatrix}}$$

$$= \frac{\text{Rise in temperature of cooling water}}{\begin{bmatrix}\text{Temp. corresponding to the absolute} \\ \text{pressure in the condenser}\end{bmatrix} - \begin{bmatrix}\text{Inlet temp. of} \\ \text{cooling water}\end{bmatrix}}$$

DETERMINATION OF MASS OF COOLING WATER

Let m_w = Mass of cooling water required in kg/h,

m_s = Mass of steam condensed in kg/h,

t_s = Saturation temperature of steam corresponding to the condenser vacuum in °C,

t_c = Temperature of the condensate leaving the condenser,

t_{w_1} = Temperature of cooling water at inlet in °C,

t_{w_2} = Temperature of cooling water at outlet in °C,

c_{pw} = Specific heat of water at constant pressure,

x = Dryness fraction of steam entering the condenser, and

h_{fg} = Latent heat of 1 kg of steam entering the condenser.

Now, heat lost by steam $= m_s[xh_{fg_2} + c_{pw}(t_s - t_c)]$ kJ/kg

and heat gained by water $= m_w \times c_{pw}(t_{w_2} - t_{w_1})$

1. A *steam condenser* is a device or an appliance in which steam condenses and heat released by steam is absorbed by water.
2. *Vacuum* is sub-atmospheric pressure. It is measured as the pressure depression below atmospheric.
3. Condensers are of two types :
   (*i*) *Jet condensers*—exhaust steam and water come in direct contact with each other and the temperature of the condensate is the same as that of cooling water leaving the condenser.
   (*ii*) *Surface condensers*—exhaust steam and water do not come into direct contact. The steam passes over the tubes through which a supply of cooling water is maintained.
4. Jet condensers may be classified as : (*i*) Parallel-flow type, (*ii*) Counter-flow type and (*iii*) Ejector type.
5. Surface condensers may be classified as : (*i*) Down-flow type, (*ii*) Central-flow type, (*iii*) Inverted-flow type, (*iv*) Regenerative type and (*v*) Evaporative type.
6. *Vacuum efficiency* is defined as the ratio of the actual vacuum to the maximum obtainable vacuum.
7. *Condenser efficiency* is defined as the ratio of the difference between the outlet and inlet temperatures of cooling water to the difference between the temperature corresponding to the vacuum in the condenser and inlet temperature of cooling water.
8. The quantity of cooling water, m_w is found by the following expression :

$$m_w = \frac{m_s[xh_{fg} + c_{pw}(t_s - t_{w_2})]}{c_{pw}(t_{w_2} - t_{w_1})} \text{ kg/h} \qquad \text{...... \textbf{Jet condenser.}}$$

$$m_w = \frac{m_s[xh_{fg} + c_{pw}(t_s - t_c)]}{c_{pw}(t_{w_2} - t_{w_1})} \text{ kg/h} \qquad \text{......\textbf{Surface condenser.}}$$

Example. *The outlet and inlet temperatures of cooling water to a condenser are 37.5°C and 30°C respectively. If the vacuum in the barometer is 706 mm of mercury with barometer reading 760 mm determine efficiency.*

Solution. $t_{w_1} = 30°\text{C}, \; t_{w_2} = 37.5°\text{C}$

Absolute pressure in the condenser = 760 – 706 = 54 mm of Hg

$= 54 \times 0.001333 = 0.072$ bar

From steam tables corresponding to 0.072 bar, $t_s \simeq 40°\text{C}$

∴ *Condenser efficiency*

$$= \frac{\text{Rise in temp. of cooling water}}{\text{(Temperature corresponding to vacuum in condenser – Inlet temperature of cooling water)}}$$

$$= \frac{(t_{w_2} - t_{w_1})}{(t_s - t_{w_1})} = \frac{37.5 - 30}{40 - 30} = \frac{7.5}{10} = \mathbf{0.75 \text{ or } 75\%.} \quad \textbf{(Ans.)}$$

Example . *The surface condenser is designed to handle 16000 kg of steam per hour. The steam enters the condenser at 0.09 bar abs. and 0.88 dryness fraction and the condensate leaves the condenser at the corresponding saturation temperature. Determine the rise in cooling water temperature if the cooling water flow rate is 8.96 × 10^5 kg/hour. Assume that the pressure is constant throughout the condenser.*

Solution. Mass of steam handled per hour = 16000 kg

Cooling water flow per hour, $m_w = 8.96 \times 10^5$ kg/hour

From steam tables corresponding to dry saturated steam, for given pressure of 0.09 bar of steam, for 1 kg steam ;

$$h_f = 183.3 \text{ kJ/kg} ; t_s = 43.8°\text{C}, h_{fg} = 2397.7 \text{ kJ/kg}$$

Total heat lost by the wet steam of dryness fraction 0.88 in one hour

$$= 16000\,(h_f + xh_{fg} - c_{pw} \times t_s) = 16000\,(183.3 + 0.88 \times 2397.7 - 4.184 \times 43.8)$$

$$= 16000\,(183.3 + 2109.9 - 183.26) = 337.59 \times 10^5 \text{ kJ}$$

Heat gained by cooling water in one hour

$$= 8.96 \times 10^5 \times 4.184 \times (t_{w_2} - t_{w_1}),\ t_{w_2} - t_{w_1} \text{ being rise of temperature}$$

∴ Heat lost by steam/hour = Heat gained by water/hour

$$337.59 \times 10^5 = 8.96 \times 10^5 \, (t_{w_2} - t_{w_1})$$

∴
$$(t_{w_2} - t_{w_1}) = \frac{337.59 \times 10^5}{8.96 \times 10^5 \times 4.184} = 8.99°\text{C}$$

Hence *rise in cooling water temperature* = **9°C. (Ans.)**

Example. *A surface condenser is designed to handle 10000 kg of steam per hour. The steam enters at 0.08 bar abs. and 0.9 dryness and the condensate leaves at the corresponding saturation temperature. The pressure is constant throughout the condenser. Estimate the cooling water flow rate per hour, if the cooling water temperature rise is limited to 10°C.*

Solution. From steam tables.

At 0.08 bar ; $t_s = 41.5°\text{C}$, $h_{fg} = 2403.1$ kJ/kg

Also $m_s = 10000$ kg/h ; $(t_{w_2} - t_{w_1}) = 10°\text{C}$.

Saturation temperature of condensate, $t_c = 41.5°\text{C}$

Assuming that there is perfect heat transfer in the condenser,

Then, heat lost by the exhaust steam = heat gained by cooling water.

$$10000\,[xh_{fg} + c_{pw}(t_s - t_c)] = m_w \times c_{pw} \times (t_{w_2} - t_{w_1})$$

$$10000\,[0.9 \times 2403.1 + 4.184 \times (41.5 - 41.5)] = m_w \times 4.184 \times 10$$

$$\therefore \qquad m_w = \frac{10000 \times 0.9 \times 2403.1}{10 \times 4.184} = \mathbf{516919\ kg/h.} \quad \textbf{(Ans.)}$$

Example . *To check the leakage of air in a condenser, the following procedure is adopted. After ruming the plant to reach the steady conditions the steam supply to the condenser and the air and condensate pump are shut down, thus completely isolating the condenser. The temperature and vacuum readings are noted at shut down and also after a period of 10-minutes. They are 39°C and 685 mm Hg and 28°C and 480 mm Hg respectively. The barometer reads 750 mm Hg. The effective volume of the condenser is 1.5 m*^{3}*. Determine (i) quantity of air leakage into the condenser during the period of observation ; (ii) the quantity of water vapour condensed during the period.*

Solution. ***At shut down :***

From steam tables, corresponding to $t_s = 39°\text{C}$:

$$p_s = 0.07 \text{ bar} = \frac{0.07}{0.001333} = 52.5 \text{ mm Hg}$$

and $$v_g = 20.53 \text{ m}^3/\text{kg}$$

The combined pressure of steam and air in the condenser,

$$p = p_a + p_s = 750 - 685 = 65 \text{ mm Hg}$$

$\therefore$ $$p_a = p - p_s = 65 - 52.5 = 12.5 \text{ mm Hg} = 12.5 \times .001333 = 0.0167 \text{ bar}$$

Now, mass of air in 1.5 m^3

$$m_a = \frac{p_a V_a}{RT_a} = \frac{0.0167 \times 10^5 \times 1.5}{287 \times (273 + 39)} = 0.028 \text{ kg}$$

and mass of steam in 1.5 m^3

$$m_s = \frac{1.5}{20.53} = 0.073 \text{ kg}$$

After 10 minutes, observed duration,

From steam tables, corresponding to $t_s = 28°\text{C}$:

$$p_s = 0.0378 \text{ bar} = \frac{0.0378}{0.001333} = 28.36 \text{ mm Hg}, \; v_g = 36.69 \text{ m}^3\text{/kg}$$

Total pressure in the condenser,

$$p = 750 - 480 = 270 \text{ mm Hg}$$

∴ Air pressure,

$$p_a = p - p_s = 270 - 28.36 = 241.64 \text{ mm Hg}$$

$$= 241.64 \times 0.001333 = 0.322 \text{ bar}$$

Mass of air, $m_a = \dfrac{p_a V_a}{RT_a} = \dfrac{0.322 \times 10^5 \times 1.5}{287 \times (273 + 28)} = 0.559$ kg.

Mass of steam, $m_s = \dfrac{1.5}{36.69} = 0.0408$ kg

$\therefore$ **Air leakage** *in 10 minutes period*

$= (0.559 - 0.028) =$ **0.531 kg. (Ans.)**

and **steam condensed** *in 10 minutes period*

$= (0.073 - 0.0408) =$ **0.0322 kg. (Ans.)**

☞ **Example 18.** *A surface condenser is required to deal with 20000 kg of steam per hour, and the air leakage is estimated at 0.3 kg per 1000 kg of steam. The steam enters the condenser dry saturated at 38°C. The condensate is extracted at the lowest point of the condenser at a temperature of 36°C. The condensate loss is made up with water at 7°C. It is required to find the saving in condensate and the saving in heat supplied in the boiler, by fitting a separate air extraction pump which draws the air over an air cooler. Assume that the air leaves the cooler at 27°C. The pressure in the condenser can be assumed to remain constant.*

Solution. The mass of air per kg of steam at entry = 0.3/1000 = 0.0003 kg

***At 38°C* :** Saturation pressure = 0.06624 bar and

$$v_g = 21.63 \text{ m}^3/\text{kg (From steam tables)}$$

For 1 kg of steam, the volume is 21.63 m^3, and this must be the volume occupied by 0.0003 kg of air when exerting its partial pressure,

i.e., $$\text{Partial pressure of air} = \frac{m_a R_a T}{V} = \frac{0.0003 \times 0.287 \times (273 + 38) \times 10^3}{21.63 \times 10^5} = 1.2 \times 10^{-5} \text{ bar}$$

This is negligibly small and may be neglected.

Condensate extraction :

***At 36°C* :** Saturation pressure = 0.0594 bar, v_g = 23.97 m^3/kg.

The total pressure in the condenser is 0.06624 bar, hence

$$0.06624 = 0.0594 + p_a$$

$$\therefore \qquad p_a = 0.00684 \text{ bar}$$

The mass of air removed per hour is

$$\frac{20000 \times 0.3}{1000} = 6 \text{ kg/h}$$

Hence the volume of air removed per hour is

$$\frac{mRT}{p} = \frac{6 \times 0.287 \times (273 + 36) \times 10^3}{0.00684 \times 10^5} = 778 \text{ m}^3\text{/h}$$

The mass of steam associated with the air removed is therefore given by

$$\frac{778}{23.97} = 32.45 \text{ kg/h.}$$

Separate extraction :

***At 27°C* :** Saturation pressure = 0.03564 bar, v_g = 38.81 m^3/kg

The air partial pressure = 0.06624 – 0.03564 = 0.0306 bar

$$\therefore \quad \text{The volume of air removed} = \frac{mRT}{p} = \frac{6 \times 0.287 \times (273 + 27) \times 10^3}{0.0306 \times 10^5} = 168.9 \text{ m}^3\text{/h}$$

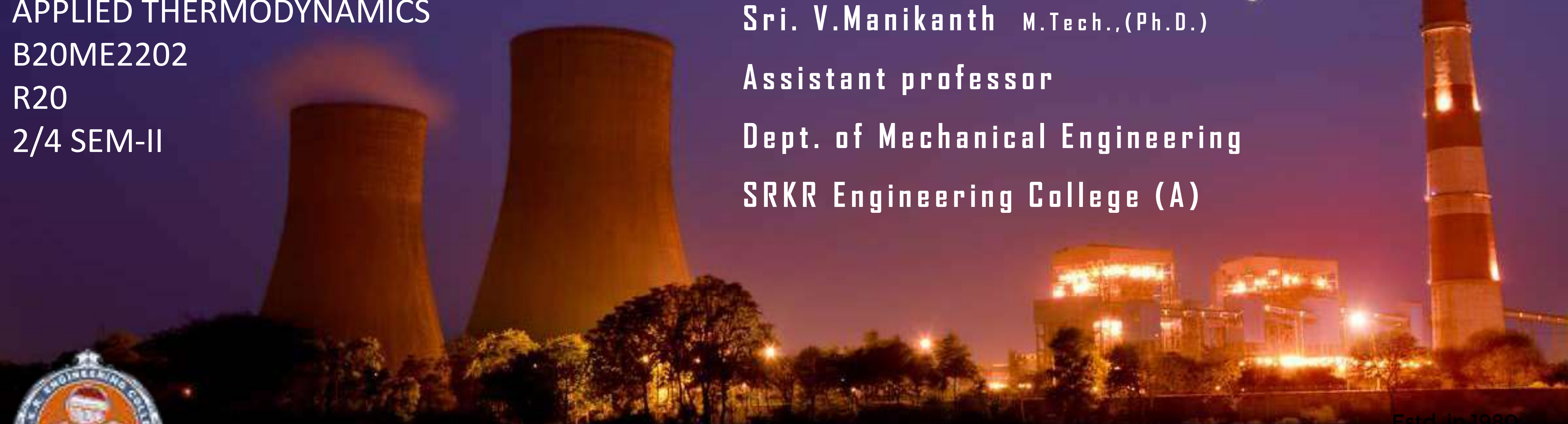
$$\therefore \quad \text{Steam removed} = \frac{168.9}{38.81} = 4.35 \text{ kg/h.}$$

Hence, *the saving in condensate by using the separate extraction method*

= 32.45 – 4.35 = **28.1 kg/h. (Ans.)**

The *saving in heat to be supplied in the boiler*

= 28.1 × 4.186 (36 – 7) = **3411 kJ/h. (Ans.)**



Vapor power cycles

APPLIED THERMODYNAMICS
B20ME2202
R20
2/4 SEM-II

By

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AUTONOMOUS INSTITUTION APPROVED BY AICTE, NEW DELHI AFFILIATED TO JNTUK, KAKINADA

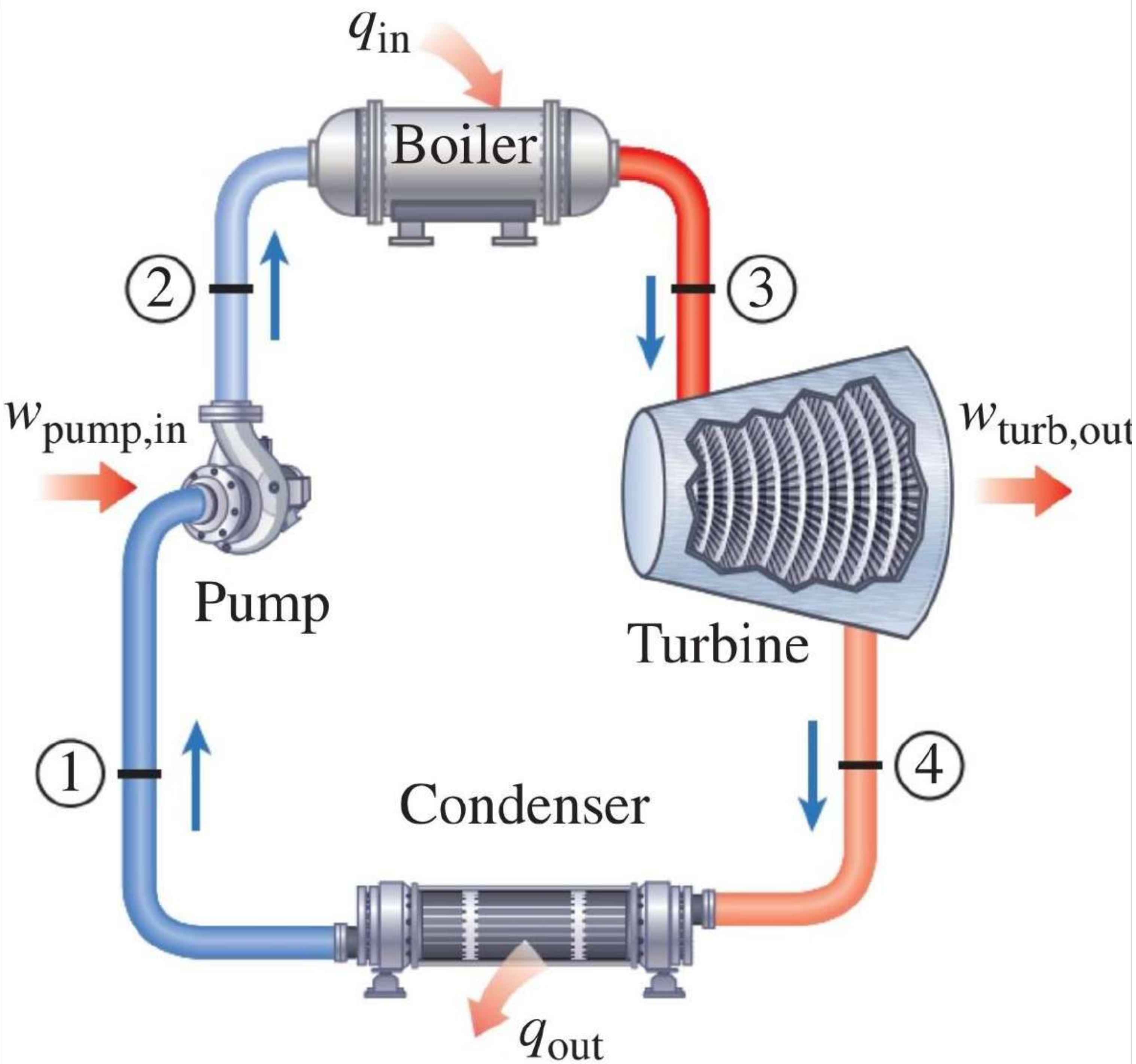
CONTENTS

Vapor power cycles: Rankine cycle- Reheat cycle(single Reheater)-Regenerative cycle- Thermodynamic variables effecting efficiency and output of Rankine and Regenerative cycles(Single open feed water heater)- Improvements of efficiency, Binary vapor power cycle.

Textbooks:

1. Thermal Engineering by Rajput, Lakshmi publications
2. Engineering Thermodynamics by PK Nag
3. Thermodynamics: An Engineering Approach by Michael A. Boles and Yunus A Çengel

The cycles on which heat engines operate to produce useful power is called power cycles. A power cycle continuously converts heat (energy released by the burning of fuel) into work , in which a working fluid repeatedly performs a succession of processes. The power cycles are categorized as ***vapor power cycle*** and ***gas power cycle***, depending upon the working fluids used in the cycle. In vapour power cycle, *the working fluid exists in vapor phase during one part of the cycle and in liquid phase during another part* whereas in a gas power cycle, the working fluid remains in the gaseous phase throughout the entire cycle. Steam power plant is working on the vapour power cycle. To study the performance of steam power plants it is important to analysis actual vapor power cycle encountered in steam power plants. However, it is difficult to analyze the actual vapor power cycle because the actual cycle consists of irreversible and non-quasi-equilibrium processes whereas we can apply thermodynamics principles only to reversible and quasi-equilibrium processes. So, to make an analytical study of an actual power cycle feasible, it is advantageous to device an idealized vapor power cycle with general characteristics of the actual vapor power cycle and then analyzes the performance of this idealized cycle.



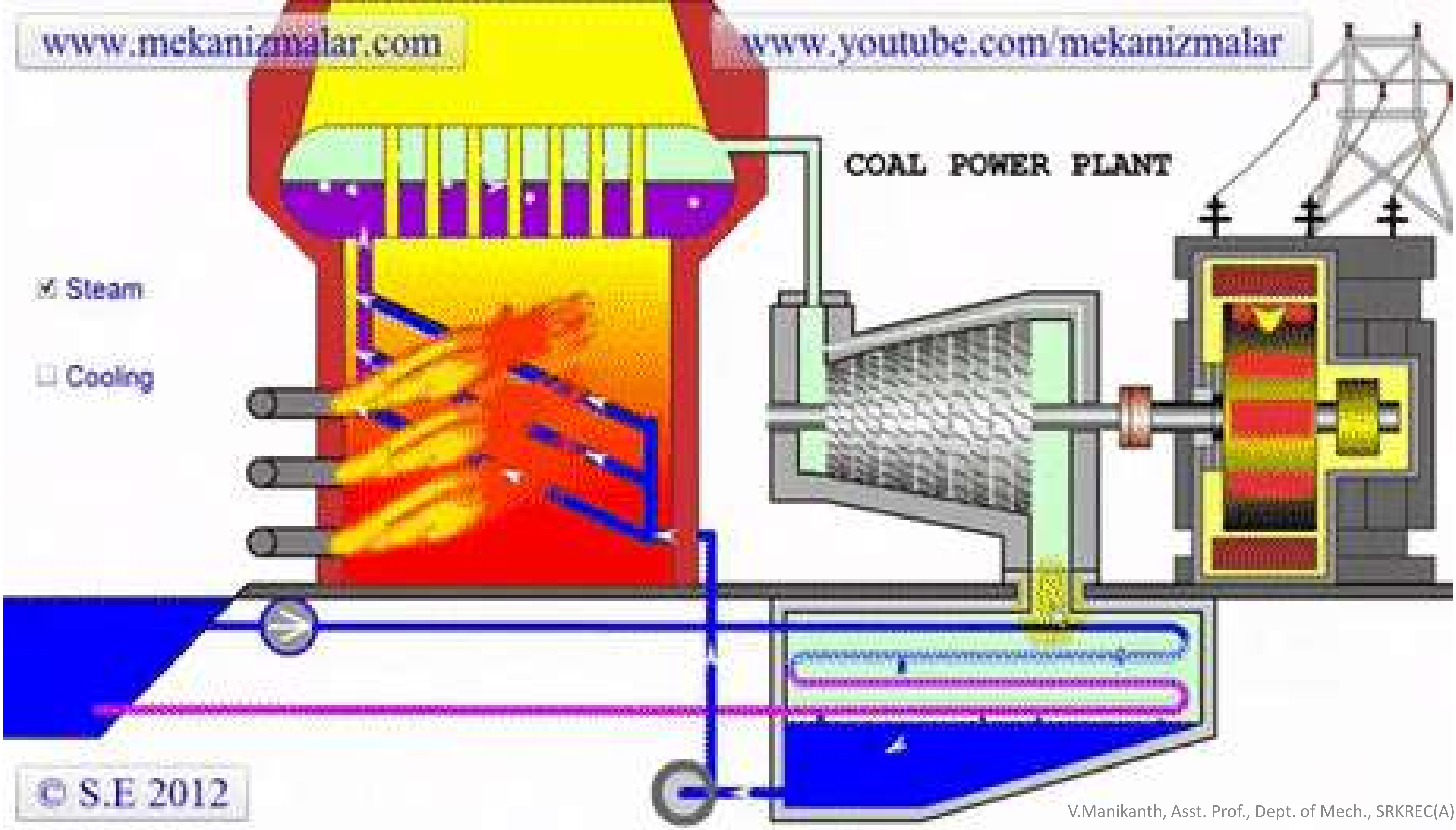
In a steam power plant the following processes take place:

1-2: The condensed steam is pumped back into the boiler to its initial state constituting a cycle.

2-3: The heat energy released by combustion of fuel in furnace is utilized to vaporize water into steam in a boiler.

3-4: The steam produced in the boiler is expanded in a steam turbine or steam engine to obtain useful work.

4-1: The vapor leaving the steam turbine or steam engine is condensed in a condenser.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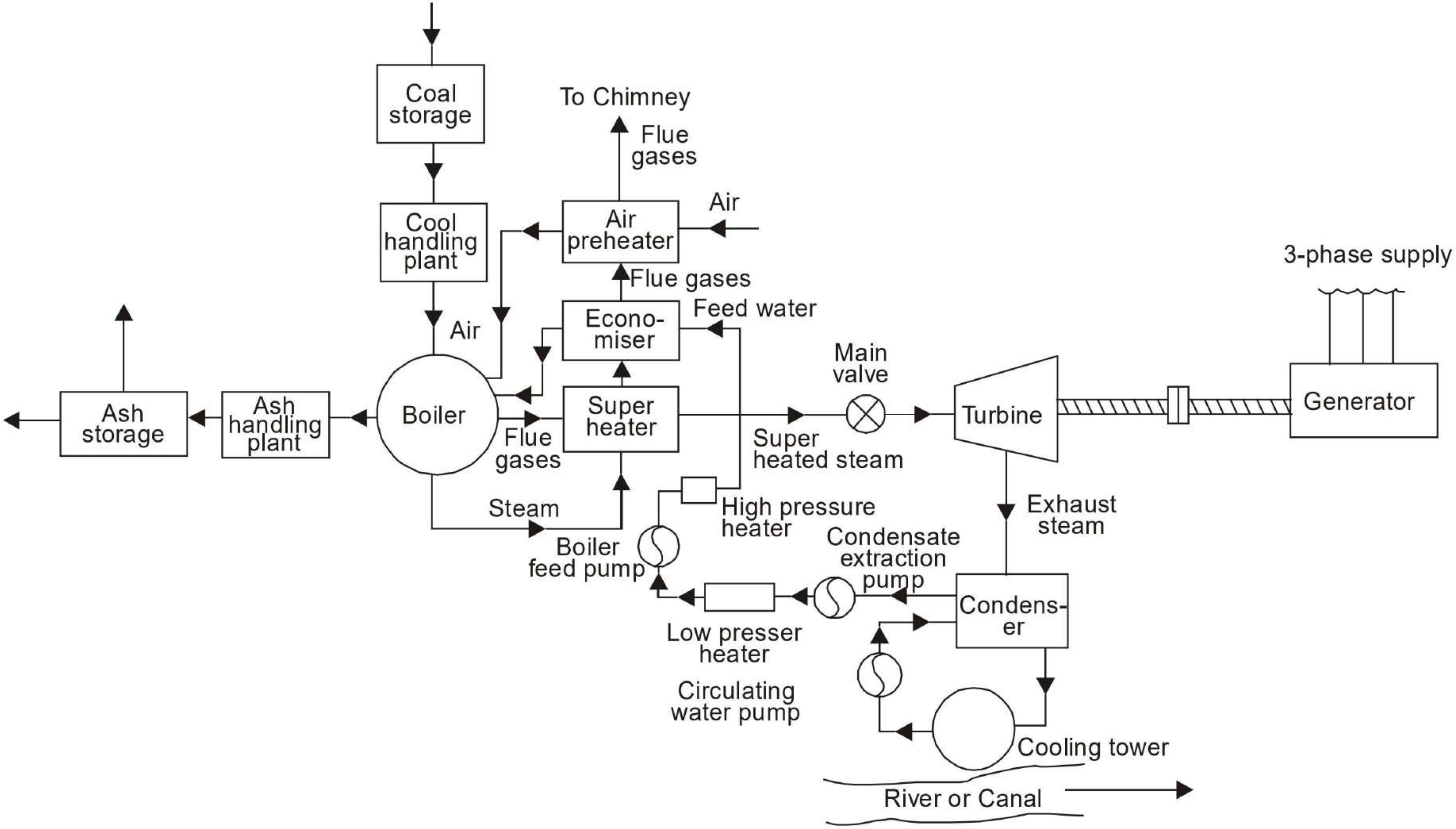


Fig. Steam Power Plant.

THE CARNOT VAPOR CYCLE

The Carnot cycle is the most efficient cycle operating between two specified temperature limits. Thus it is natural to look at the Carnot cycle first as a prospective ideal cycle for vapor power plants.

Consider a steady-flow Carnot cycle executed within the saturation dome of a pure substance, as shown in Fig.

1-2: The fluid is heated reversibly and isothermally in a boiler

2-3: Expanded isentropically in a turbine

3-4: condensed reversibly and isothermally in a condenser

4-1: compressed isentropically by a compressor to the initial state

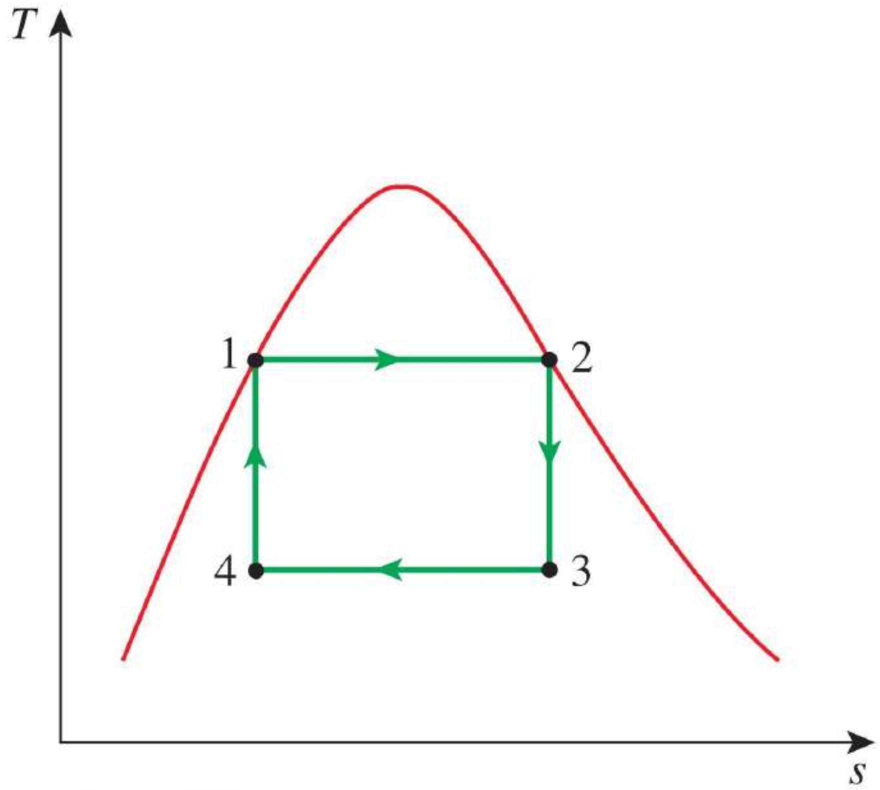


FIGURE
T-s diagram of two Carnot vapor cycles.

A Carnot cycle is not practicable for a steam power plant because of the following drawbacks:

(i) The isentropic compression process involves the compression of wet steam to a saturated liquid. It is not practical to design a pump that can handle wet steam.

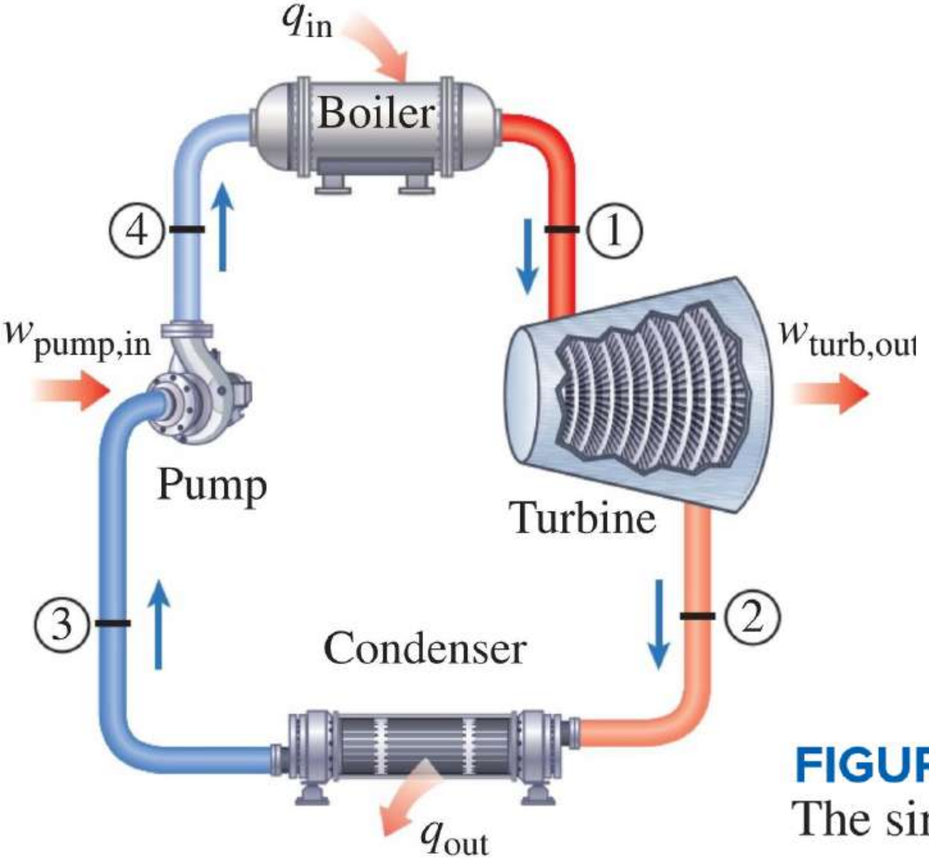
(ii) The rate of delivery of work is less, because of very large pump work.

(iii) The turbine that takes saturated vapour at the inlet produces wet steam with low quality. Thus the turbine has to handle steam with high moisture content. The impingement of liquid droplets on the turbine blades causes the erosion of the blades.

In a practical cycle, limits of pressure and volume are far more easily realized than limits of temperature so that at present no practical engine operates on the Carnot cycle, although all modern cycles aspire to achieve it.

RANKINE CYCLE: THE IDEAL CYCLE FOR VAPOR POWER CYCLES

Many of the impracticalities associated with the Carnot cycle can be eliminated by superheating the steam in the boiler and condensing it completely in the condenser. The cycle that results is the Rankine cycle, which is the ideal cycle for vapor power plants. The ideal Rankine cycle does not involve any internal irreversibilities and consists of the following four processes:



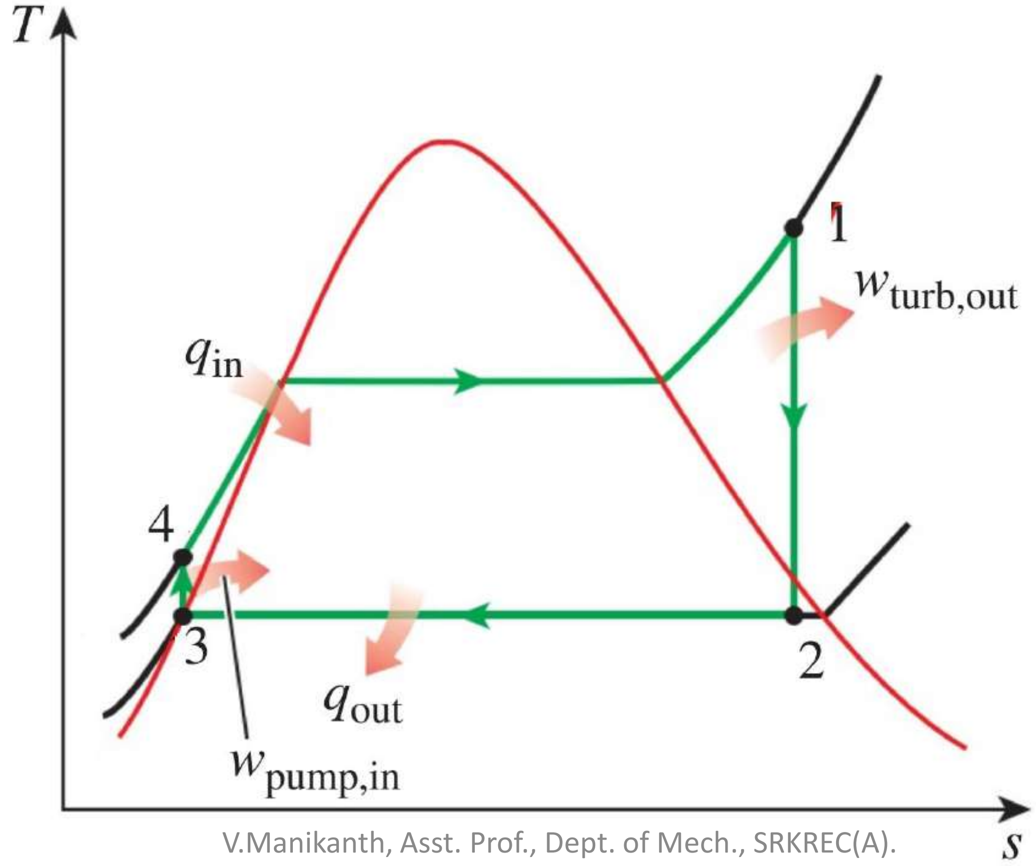


FIGURE
The simple ideal Rankine cycle.

Process 1-2 : Reversible adiabatic expansion in the turbine (or steam engine).

Process 2-3 : Constant-pressure heat rejection in the condenser.

Process 3-4 : Reversible adiabatic pumping process in the feed pump.

Process 4-1 : Constant-pressure transfer of heat in the boiler.

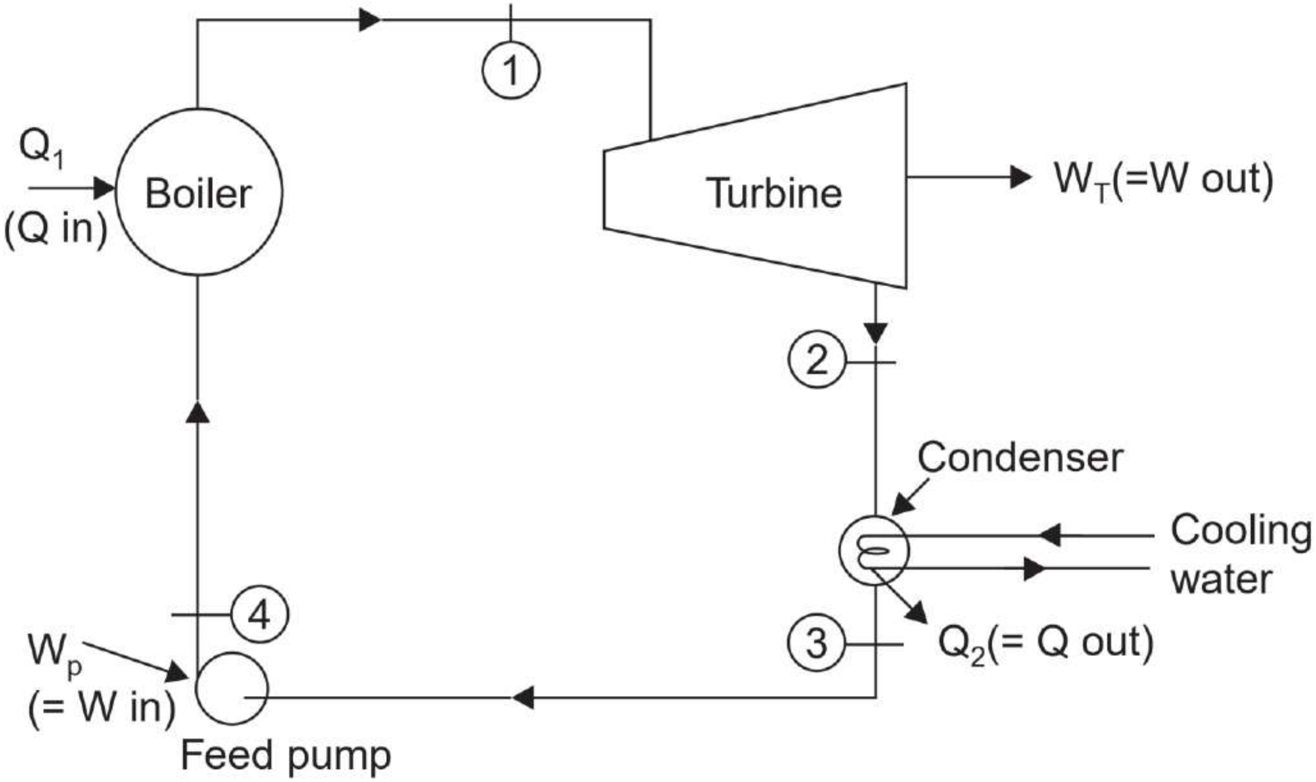


Fig. Rankine cycle.

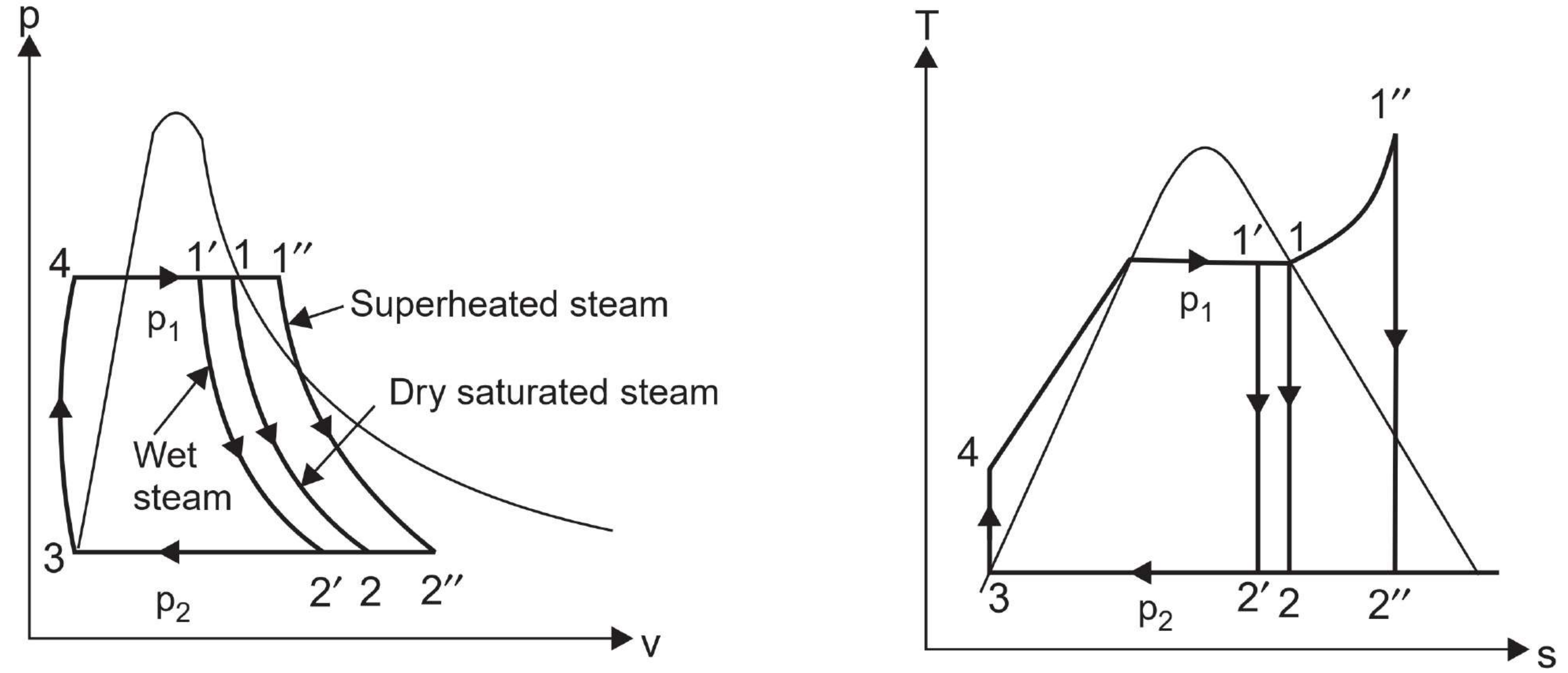


Fig. (a) p-v diagram & (b) T-s diagram of Rankine cycle

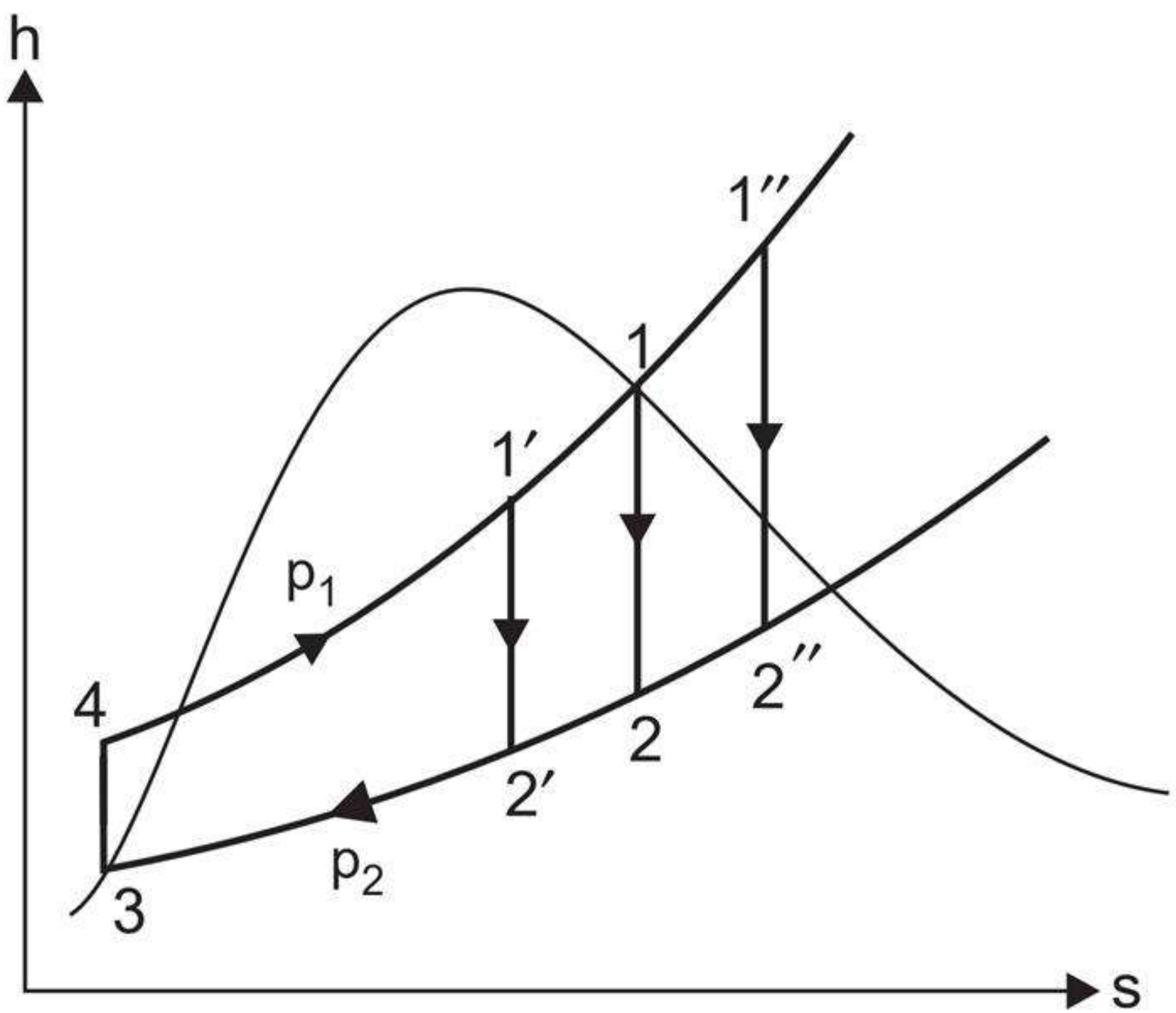


Fig.: h-s diagram of Rankine cycle

Considering 1 kg of fluid :

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

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

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

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

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

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

$$\therefore \quad \boxed{W_T = h_1 - h_2}$$

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

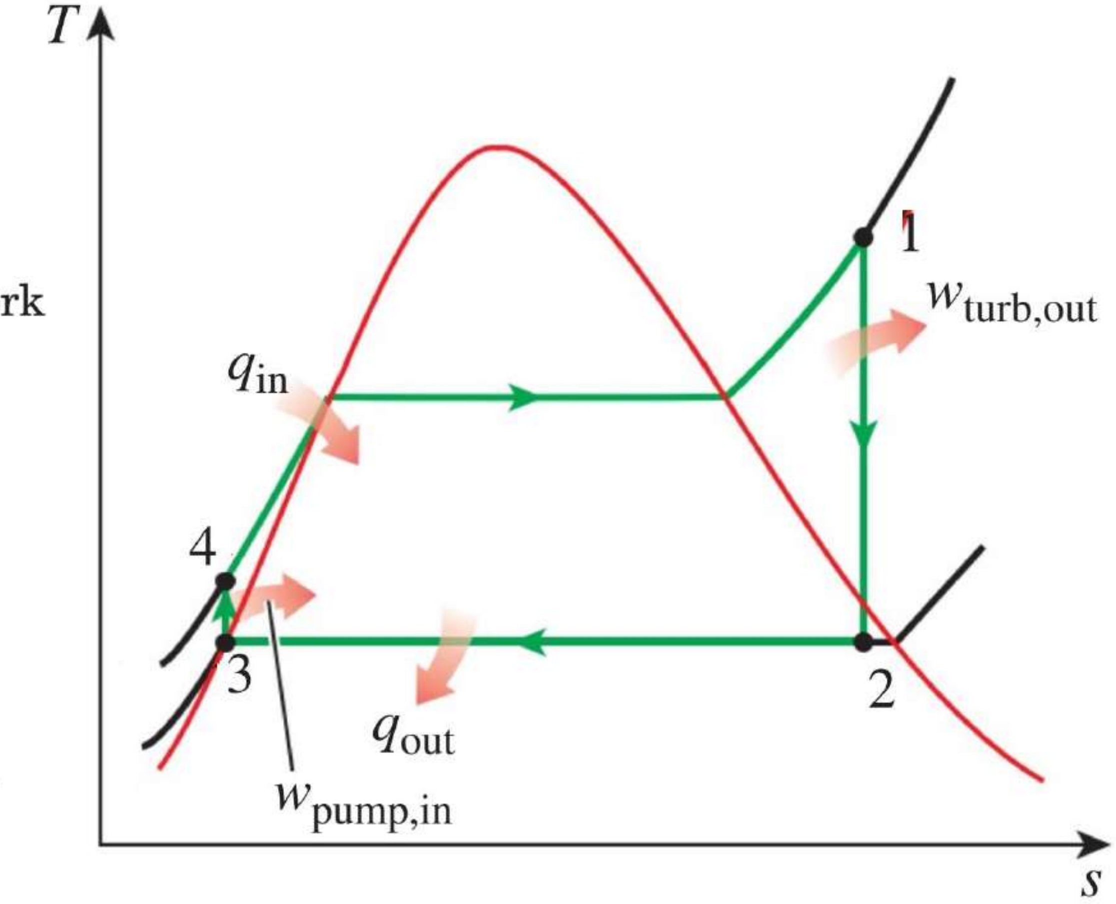
$$h_2 = Q_2 + h_{f_3}$$

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

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

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

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



Now, efficiency of Rankine cycle is given by

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

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

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

$$Tds = dh - vdp$$

$$\because \quad ds = 0$$

$$\therefore \quad dh = vdp$$

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

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

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

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

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

Then,

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

HOW CAN WE INCREASE THE EFFICIENCY OF THE RANKINE CYCLE?

Steam power plants are responsible for the production of most electric power in the world, and even small increases in thermal efficiency can mean large savings from the fuel requirements. Therefore, every effort is made to improve the efficiency of the cycle on which steam power plants operate.

The basic idea behind all the modifications to increase the thermal efficiency of a power cycle is the same: *Increase the average temperature at which heat is transferred to the working fluid in the boiler, or decrease the average temperature at which heat is rejected from the working fluid in the condenser*. That is, the average fluid temperature should be as high as possible during heat addition and as low as possible during heat rejection.

The thermal efficiency of the Rankine cycle is improved by the following methods :

(i) By regenerative feed heating.

(ii) By reheating of steam.

(iii) By water extraction.

(iv) By using binary-vapour

THE IDEAL REHEAT RANKINE CYCLE

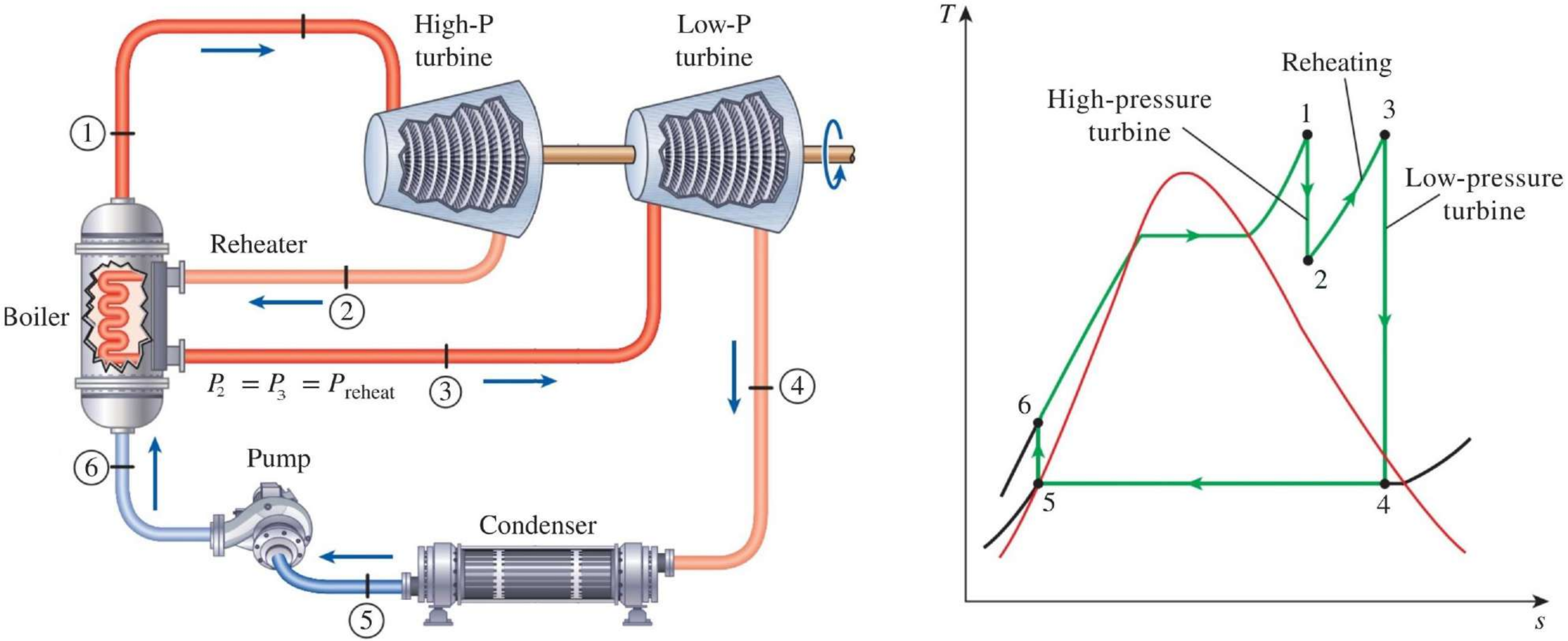
We know that, Increasing the boiler pressure increases the thermal efficiency of the Rankine cycle, but it also increases the moisture content of the steam to unacceptable levels. Then it is natural to ask the following question:

How can we take advantage of the increased efficiencies at higher boiler pressures without facing the problem of excessive moisture at the final stages of the turbine?

Two possibilities come to mind:

1. **Superheat** the steam to very high temperatures before it enters the turbine. This would be the desirable solution since the average temperature at which heat is added would also increase, thus increasing the cycle efficiency. This is not a viable solution, however, since it requires raising the steam temperature to metallurgically unsafe levels.

2. Expand the steam in the turbine in two stages, and ***reheat*** it in between. In other words, modify the simple ideal Rankine cycle with a reheat process. Reheating is a practical solution to the excessive moisture problem in turbines, and it is commonly used in modern steam power plants.



FIGURE

The ideal reheat Rankine cycle.

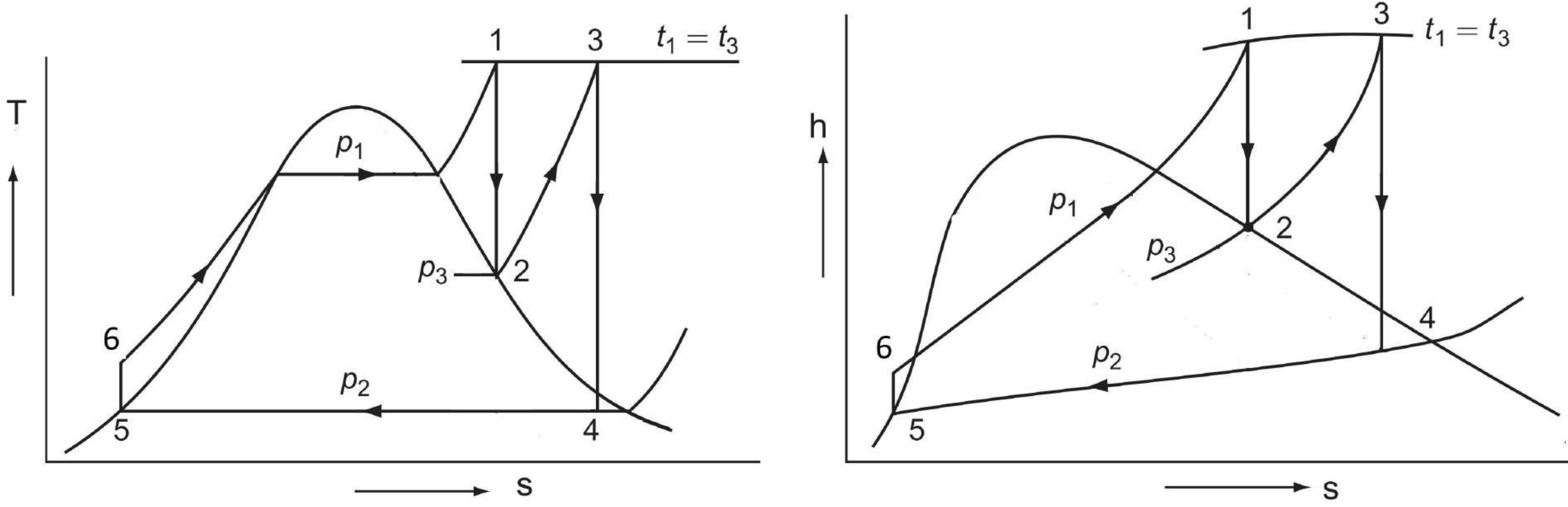


Fig. *Reheat cycle*

The steam as at state point 1 (i.e., pressure p1 and temperature T1) enters the turbine and expands isentropically to a certain pressure p3 and temperature T2. From this state point 2 the whole of steam is drawn out of the turbine and is reheated in a reheater to a temperature T3. This reheated steam is then readmitted to the low pressure turbine where it is expanded to condenser pressure isentropically.

for 1 kg of steam

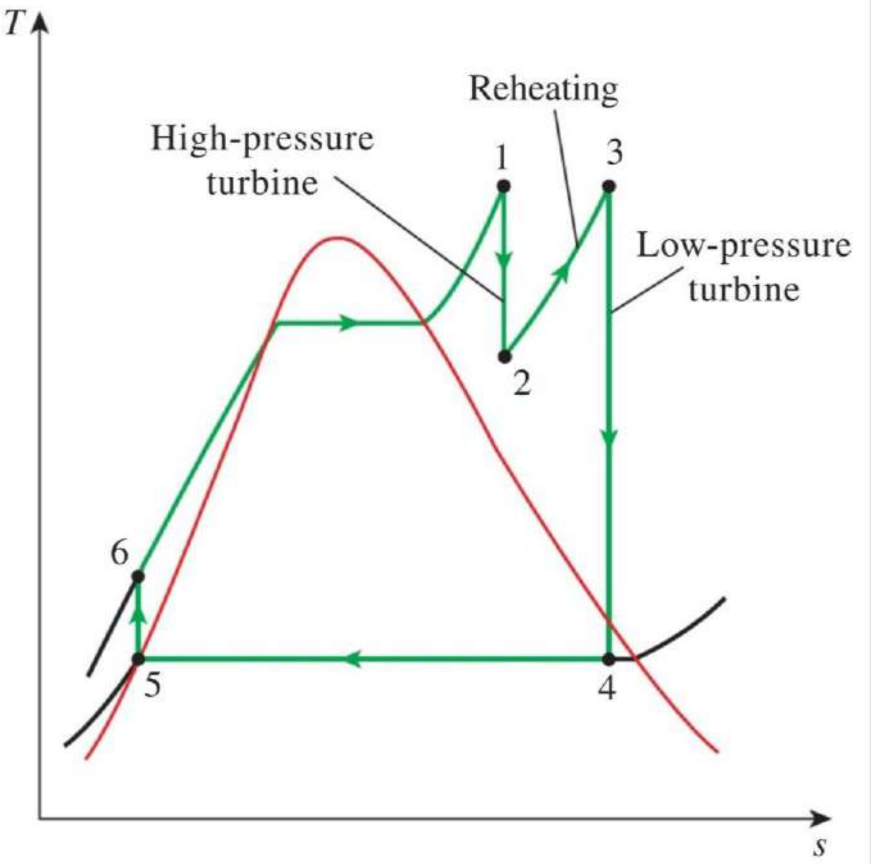
$$Q_1 = (h_1 - h_6) + \underbrace{(h_3 - h_2)}_{\text{Reheating}}$$

$$Q_2 = h_4 - h_5$$

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

$$W_P = h_6 - h_5$$

$$\eta = \frac{W_T - W_P}{Q_1} = \frac{(h_1 - h_2) + (h_3 - h_4) - (h_6 - h_5)}{(h_1 - h_6) + (h_3 - h_2)}$$



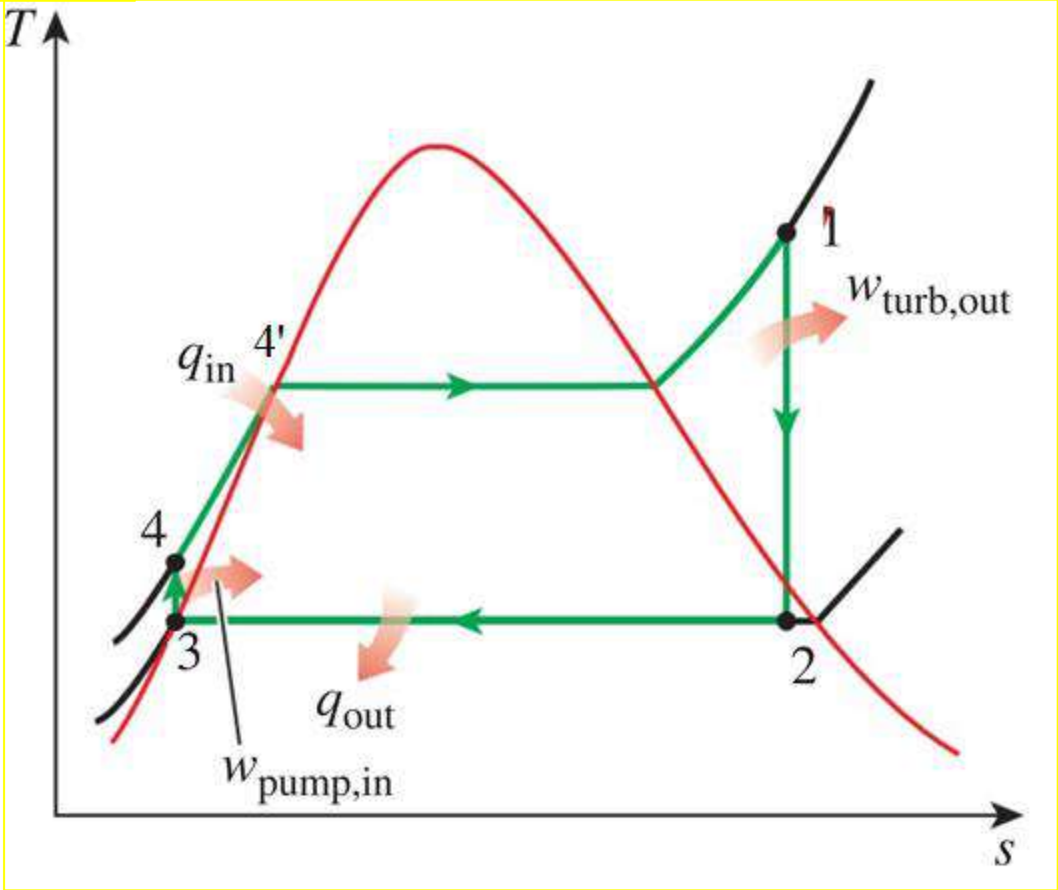
Advantages of ‘Reheating’ :

1. There is an increased output of the turbine.
2. Erosion and corrosion problems in the steam turbine are eliminated/avoided.
3. There is an improvement in the thermal efficiency of the turbines.
4. Final dryness fraction of steam is improved.
5. There is an increase in the nozzle and blade efficiencies.

Disadvantages :

1. Reheating requires more maintenance.
2. The increase in thermal efficiency is not appreciable in comparison to the expenditure incurred in reheating.

THE IDEAL REGENERATIVE RANKINE CYCLE



A careful examination of the T-s diagram of the Rankine cycle reveals that the heat is transferred to the working fluid during process 4-4′ at a relatively low temperature. This lowers the average heat-addition temperature and thus the cycle efficiency. To remedy this shortcoming, we look for ways to raise the temperature of the liquid leaving the pump (called the feedwater) before it enters the boiler.

One such possibility is to transfer heat to the feedwater from the expanding steam in a counterflow heat exchanger built into the turbine, that is, to use **regeneration.** This solution is also impractical because it is difficult to design such a heat exchanger and because it would increase the moisture content of the steam at the final stages of the turbine.

A practical regeneration process in steam power plants is accomplished by extracting, or "**bleeding**," steam from the turbine at various points. This steam, which could have produced more work by expanding further in the turbine, is used to heat the feedwater instead. The device where the feedwater is heated by regeneration is called a **regenerator**, or a **feedwater heater** (FWH). The resulting cycle is known as **regenerative** or **bleeding cycle**. A feedwater heater is basically a heat exchanger where heat is transferred from the steam to the feedwater either by mixing the two fluid streams (open feedwater heaters) or without mixing them (closed feedwater heaters).

Regeneration not only improves cycle efficiency, but also provides a convenient means of deaerating the feedwater (removing the air that leaks in at the condenser) to prevent corrosion in the boiler. It also helps control the large volume flow rate of the steam at the final stages of the turbine (due to the large specific volumes at low pressures). Therefore, regeneration has been used in all modern steam power plants since its introduction in the early 1920s.

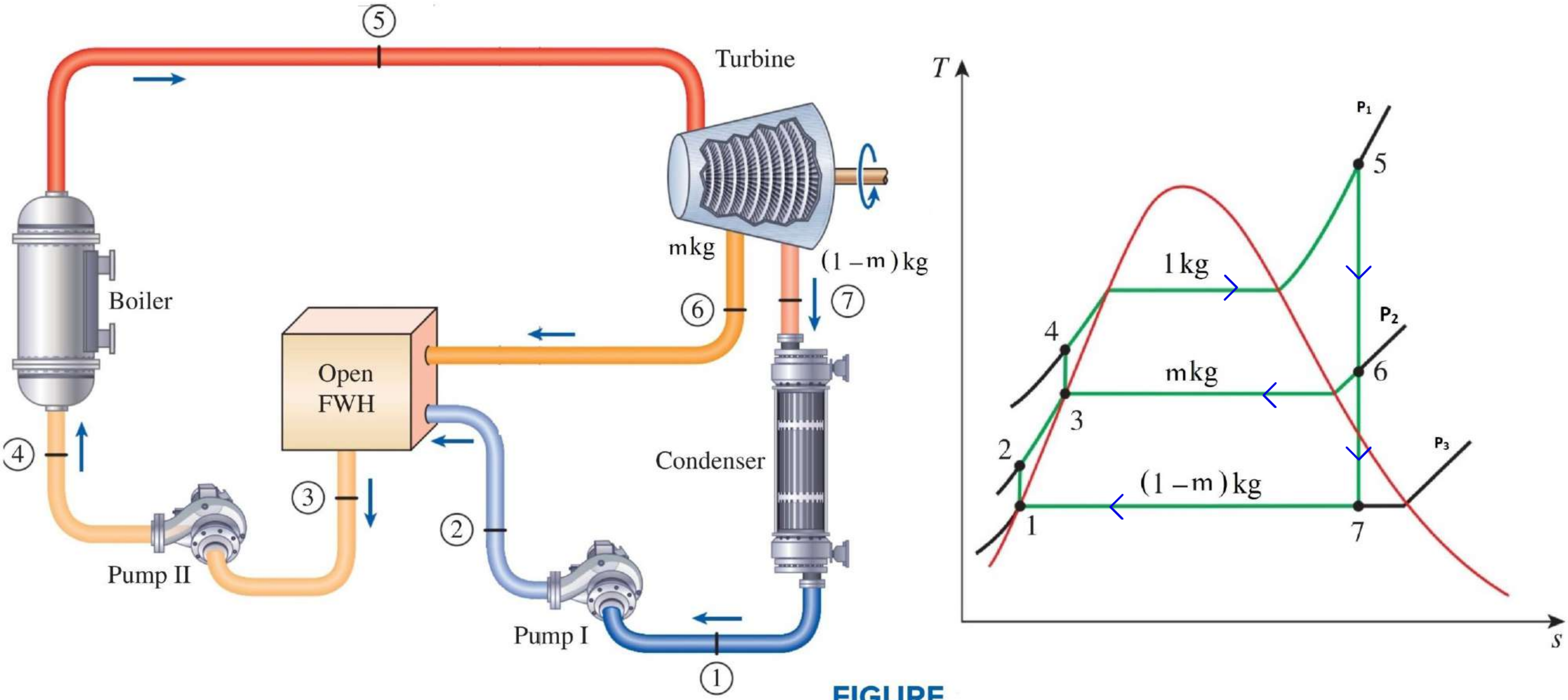


FIGURE
The ideal regenerative Rankine cycle with an open feedwater heater.

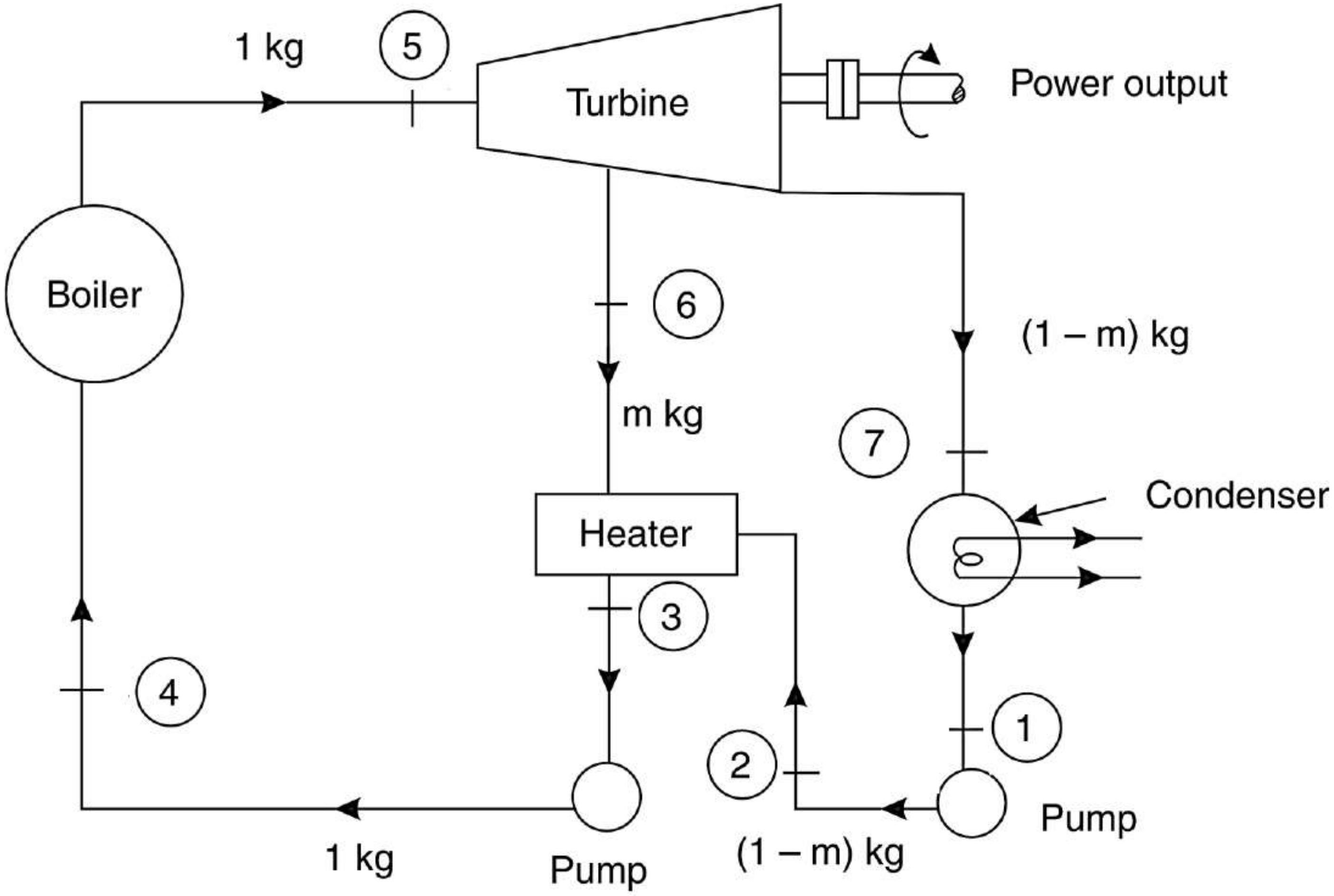


Fig: Plant diagram of regenerative cycle

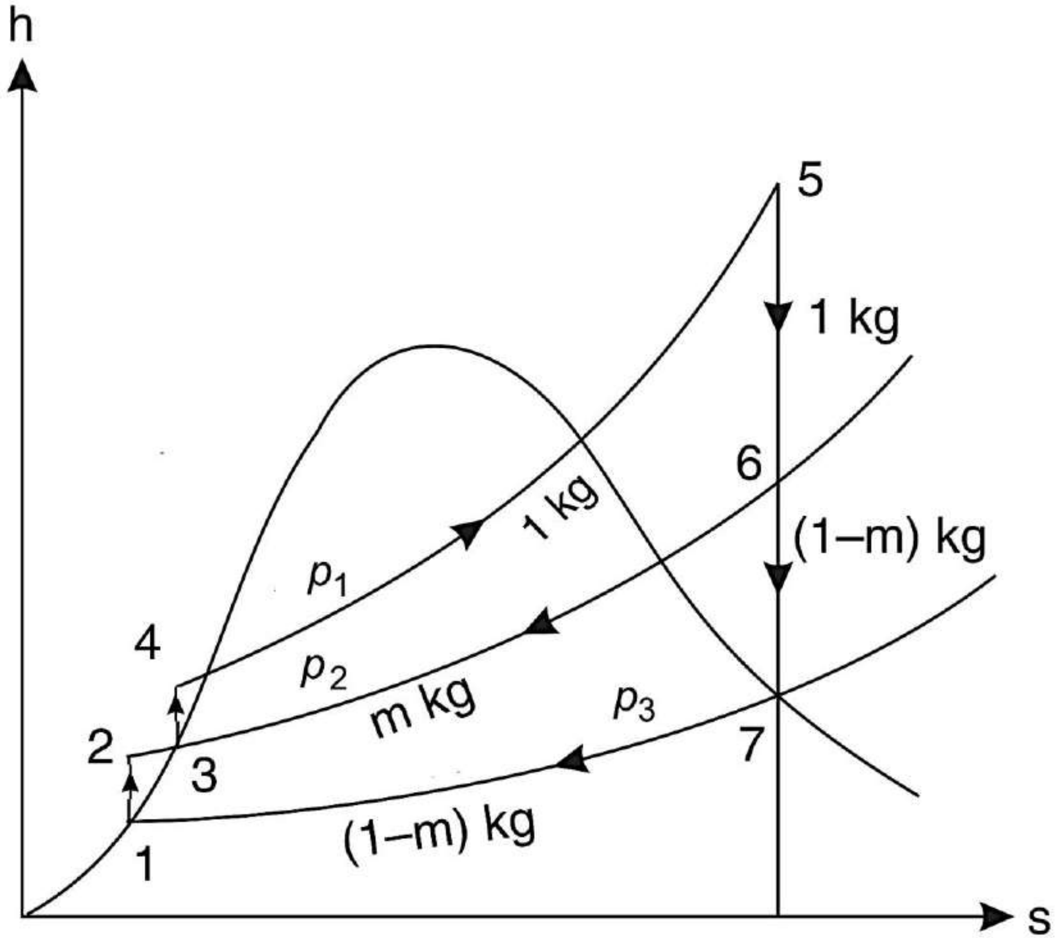


Fig: h-s diagram of regenerative cycle

In the analysis of steam power plants, it is more convenient to work with quantities expressed per unit mass of the steam flowing through the boiler. For each 1 kg of steam leaving the boiler, m kg expands partially in the turbine and is extracted at state 6. The remaining (1 – m) kg expands completely to the condenser pressure. Therefore, the mass flow rates are different in different components. If the mass flow rate through the boiler is m* , for example, it is (1 – m)m* through the condenser. This aspect of the regenerative Rankine cycle should be considered in the analysis of the cycle as well as in the interpretation of the areas on the T–s diagram. The heat and work interactions of a regenerative Rankine cycle with one feedwater heater can be expressed per unit mass of steam flowing through the boiler as follows:

For every 1 kg of steam

$$Q_{in} = h_5 - h_4$$

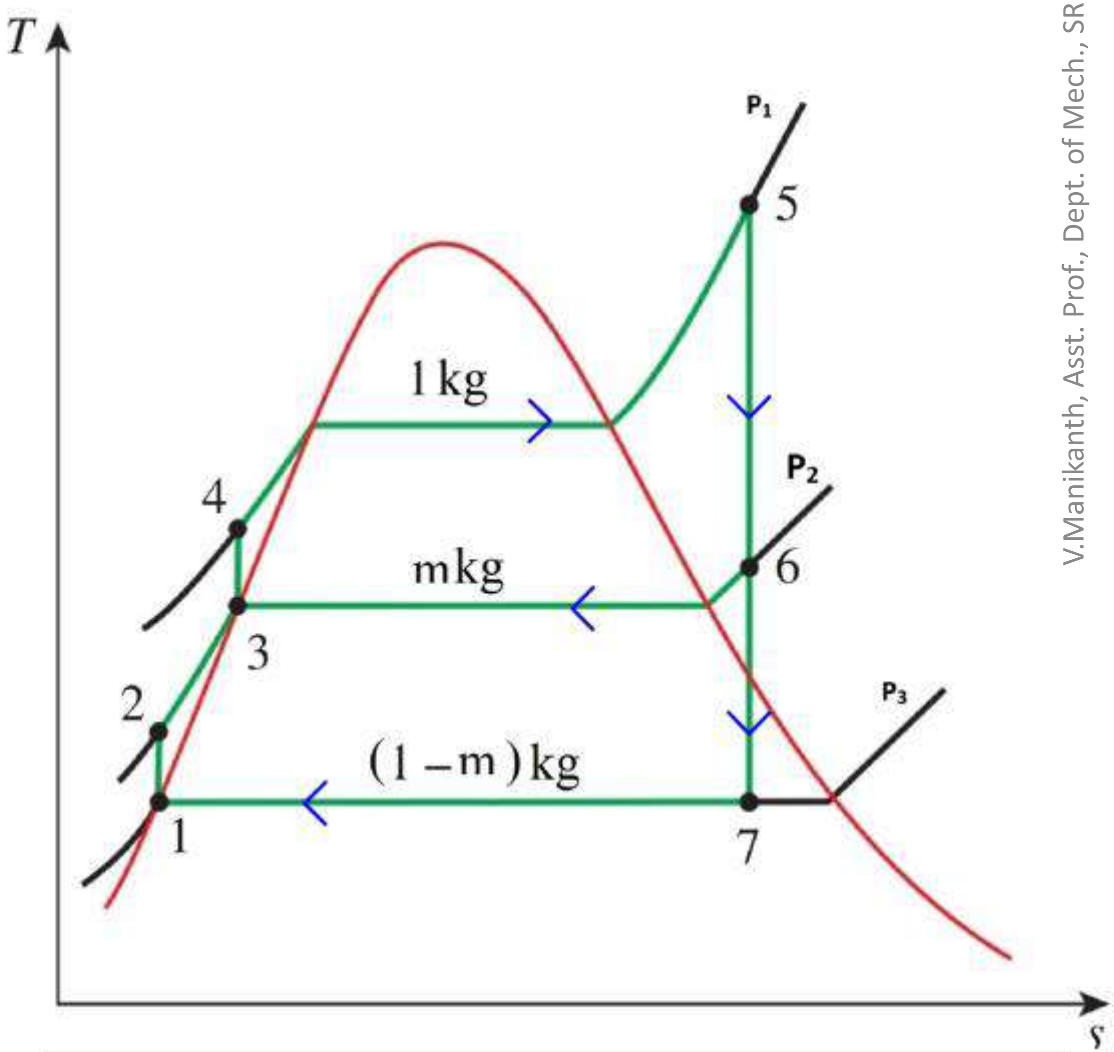
$$Q_{Out} = (1-m)\,(h_7 - h_1)$$

$$W_T = (h_5 - h_6) + (1-m)\,(h_6 - h_7)$$

$$W_P = (1-m)\,W_{P1} + W_{P2}$$

$$= (1-m)\,(h_2 - h_1) + (h_4 - h_3)$$

$$\eta_{Reg} = \frac{W_{net}}{Q_{in}} = \frac{W_T - W_P}{Q_{in}}$$



$$\boxed{\text{Steam rate} = \frac{3600}{W_T - W_P}\ \text{kg/kW h}}$$

Advantages of Regenerative cycle over Simple Rankine cycle :

1. The heating process in the boiler tends to become reversible.
2. The thermal stresses set up in the boiler are minimized. This is due to the fact that temperature ranges in the boiler are reduced.
3. The thermal efficiency is improved because the average temperature of heat addition to the cycle is increased.
4. Heat rate is reduced.
5. The blade height is less due to the reduced amount of steam passed through the low pressure stages.
6. Due to many extractions there is an improvement in the turbine drainage and it reduces erosion due to moisture.
7. A small size condenser is required.

Disadvantages :

1. The plant becomes more complicated.
2. Because of addition of heaters greater maintenance is required.
3. For given power a large capacity boiler is required.
4. The heaters are costly and the gain in thermal efficiency is not much in comparison to the heavier costs.

Note: *In the absence of precise information (regarding actual temperature of the feed water entering and leaving the heaters and of the condensate temperatures) the following assumption should always be made while doing calculations :*

1. Each heater is ideal and bled steam just condenses.
2. The feed water is heated to saturation temperature at the pressure of bled steam.
3. Unless otherwise stated the work done by the pumps in the system is considered negligible.
4. There is equal temperature rise in all the heaters (usually 10°C to 15°C).

Binary Vapor Power Cycle

The binary cycle is an attempt to overcome some of the shortcomings of water and to approach the ideal working fluid by using two fluids. Before we discuss the binary cycle, let us list the characteristics of a working fluid most suitable for vapor power cycles:

1. A *high critical temperature and a safe maximum pressure*. A critical temperature above the metallurgically allowed maximum temperature (about 620°C) makes it possible to transfer a considerable portion of the heat isothermally at the maximum temperature as the fluid changes phase. This makes the cycle approach the Carnot cycle. Very high pressures at the maximum temperature are undesirable because they create material-strength problems.

2. *Low triple-point temperature*. A triple-point temperature below the temperature of the cooling medium prevents any solidification problems.

3. *A condenser pressure that is not too low*. Condensers usually operate below atmospheric pressure. Pressures well below the atmospheric pressure create air-leakage problems. Therefore, a substance whose saturation pressure at the ambient temperature is too low is not a good candidate.

4. A *high enthalpy of vaporization* (h_{fg}) so that heat transfer to the working fluid is nearly isothermal and large mass flow rates are not needed.

5. A saturation dome that resembles an *inverted U*. This eliminates the formation of excessive moisture in the turbine and the need for reheating.

6. Good heat transfer characteristics (*high thermal conductivity*).

7. Other properties such as being inert, inexpensive, readily available, and nontoxic.

Not surprisingly, no fluid possesses all these characteristics. Water comes the closest, although it does not fare well with respect to characteristics 1, 3, and 5. We can cope with its subatmospheric condenser pressure by careful sealing, and with the inverted V-shaped saturation dome by reheating, but there is not much we can do about item 1. Water has a low critical temperature (374°C, well below the metallurgical limit) and very high saturation pressures at high temperatures (16.5 MPa at 350°C) we cannot change the way water behaves during the high-temperature part of the cycle, but we certainly can replace it with a more suitable fluid. The result is a power cycle that is actually a combination of two cycles, one in the high-temperature region and the other in the low-temperature region. Such a cycle is called a ***binary vapor cycle***. In binary vapor cycles, the condenser of the high-temperature cycle (also called the *topping cycle*) serves as the boiler of the low-temperature cycle (also called the *bottoming cycle*). That is, the heat output of the high-temperature cycle is used as the heat input to the low temperature one.

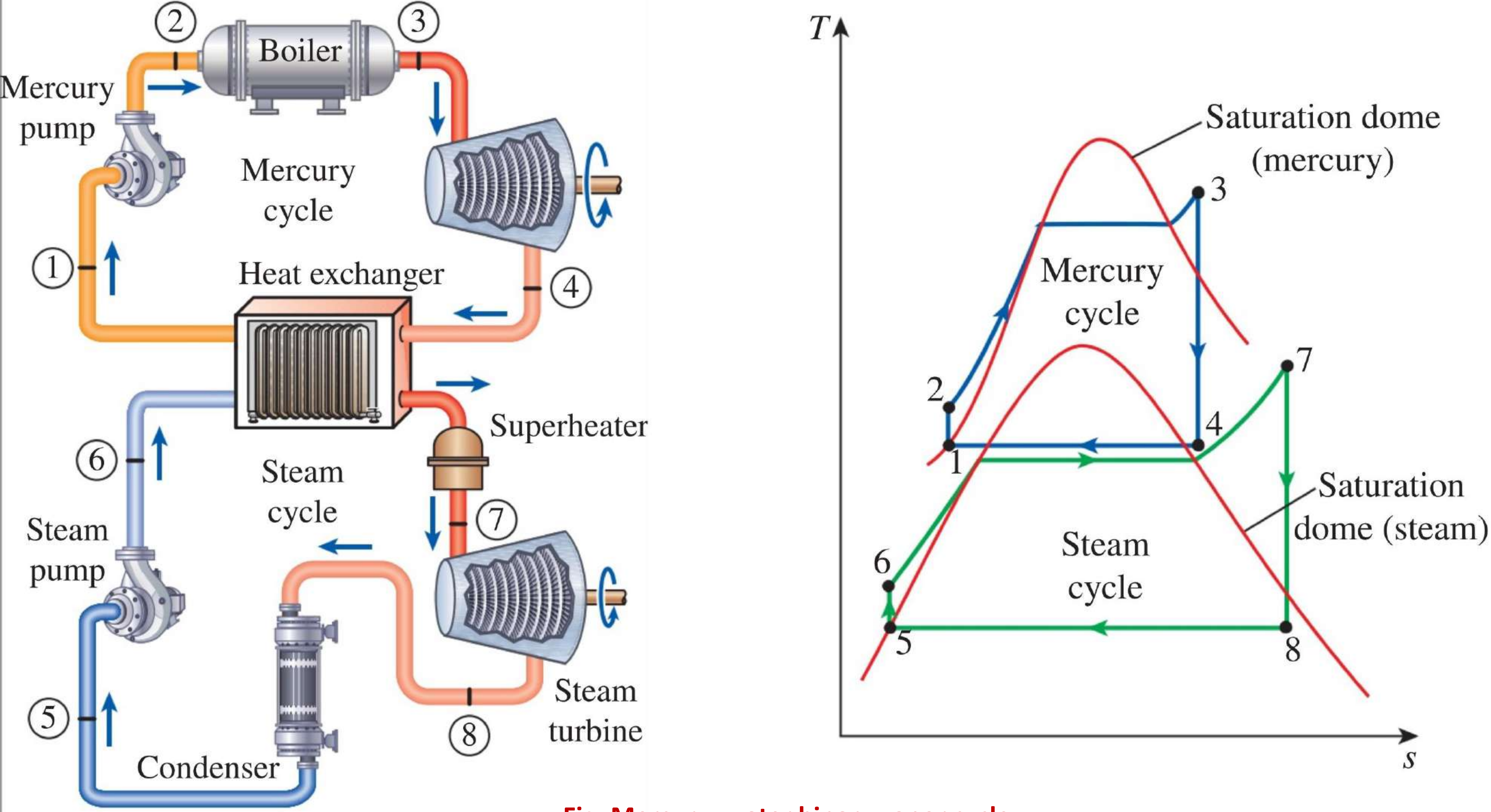


Fig: Mercury–water binary vapor cycle

The critical temperature of mercury is 898°C (well above the current metallurgical limit), and its critical pressure is only about 18 MPa. This makes mercury a very suitable working fluid for the topping cycle. Mercury is not suitable as the sole working fluid for the entire cycle, however, since at a condenser temperature of 32°C its saturation pressure is 0.07 Pa. A power plant cannot operate at this vacuum because of air-leakage problems. At an acceptable condenser pressure of 7 kPa, the saturation temperature of mercury is 237°C, which is too high as the minimum temperature in the cycle. Therefore, the use of mercury as a working fluid is limited to the high-temperature cycles. Other disadvantages of mercury are its toxicity and high cost. The mass flow rate of mercury in binary vapor cycles is several times that of water because of its low enthalpy of vaporization.

Although mercury does not have all the required properties, it is more favourable than any other fluid investigated. It is most stable under all operating conditions. Although, mercury does not cause any corrosion to metals, but it is extremely dangerous to human life, therefore, elaborate precautions must be taken to prevent the escape of vapour. The major disadvantage associated with mercury is that it does not wet surface of the metal and forms a serious resistance to heat flow. This difficulty can be considerably reduced by adding magnesium and titanium (2 parts in 100000 parts) in mercury.

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

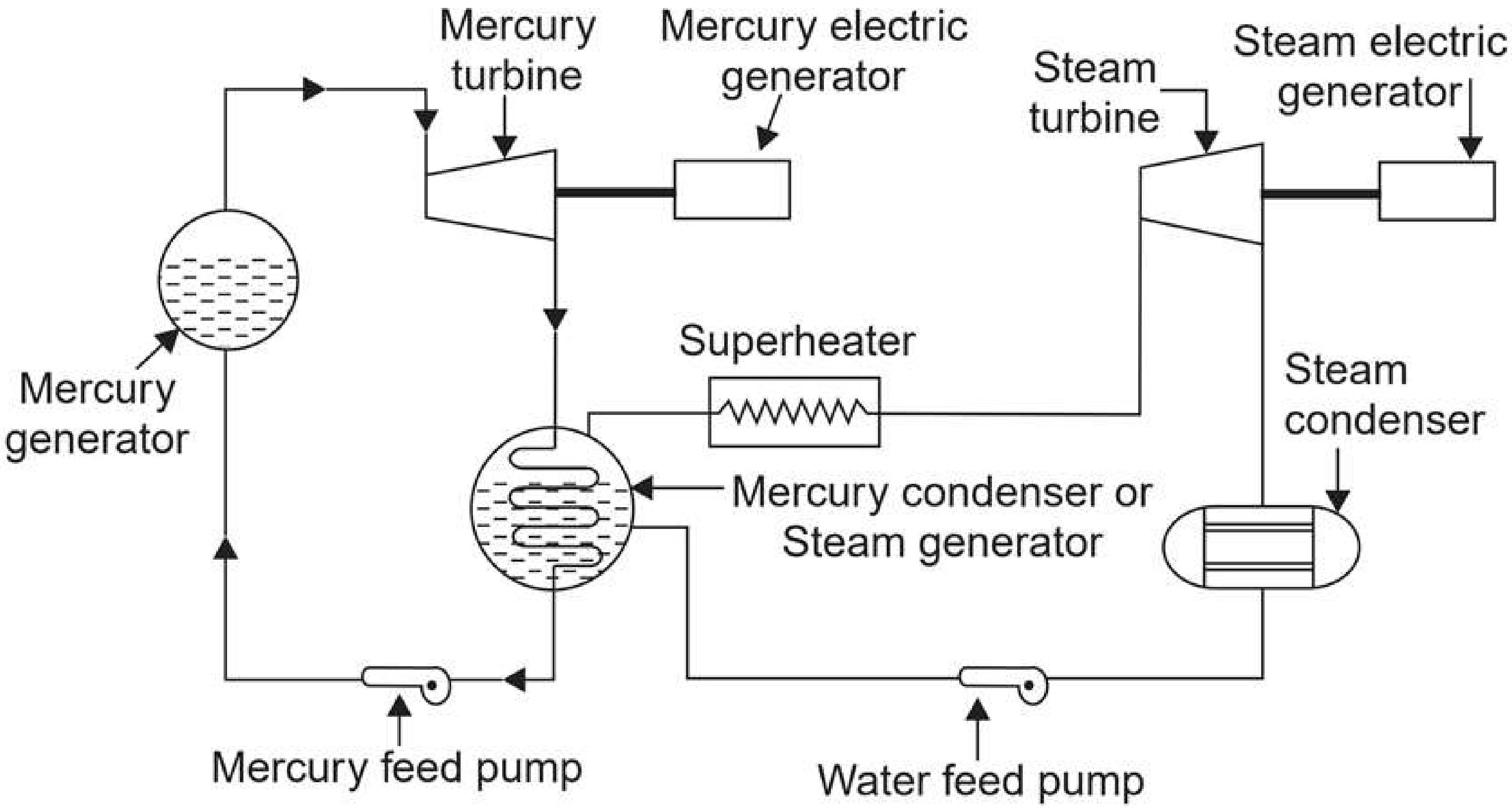


Fig. Line diagram of binary vapour cycle.

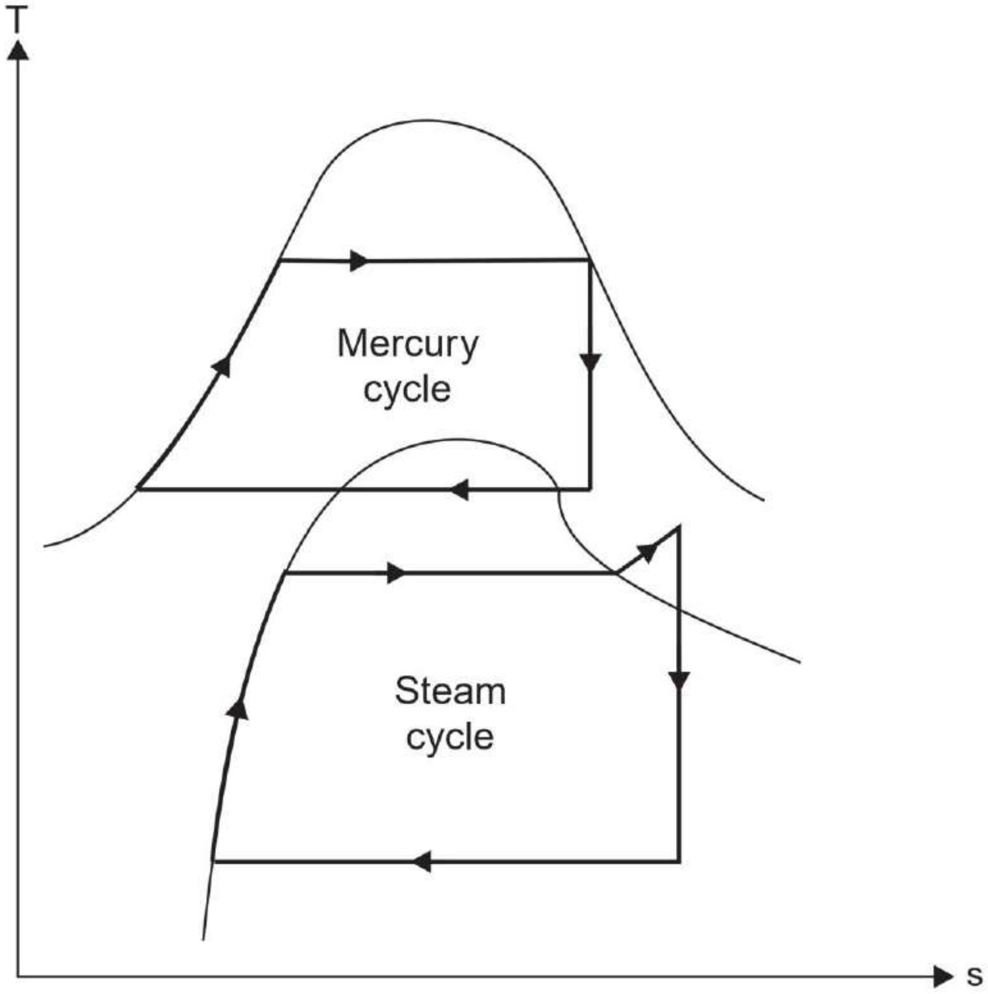


Fig. Binary vapour cycle on *T*-*s* diagram.

Analysis of Binary vapour cycle :

h_{hg_1} = Heat supplied per kg of Hg (mercury) vapour formed in the mercury boiler.

h_{hg_2} = Heat lost by one kg of Hg vapour in the mercury condenser.

h_s = Heat given per kg of steam generated in the mercury condenser or steam boiler.
W_{hg} = Work done per kg of Hg in the cycle.
W_s = Work done per kg of steam in the steam cycle.
η_s = Thermal efficiency of the steam cycle.
η_{hg} = Thermal efficiency of the Hg cycle.
m = Mass of Hg in the Hg cycle per kg of steam circulated in the steam cycle.

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

Heat supplied in the Hg boiler

$$h_t = m \times h_{hg_1} \quad \text{...(12.14)}$$

Work done in the mercury cycle

$$= m \,.\, W_{hg} \quad \text{...(12.15)}$$

Work done in the steam cycle

$$= 1 \times W_s \quad \text{...(12.16)}$$

Total work done in the binary cycle is given by

$$W_t = m\, W_{hg} + W_s \quad \text{...(12.17)}$$

∴ Overall efficiency of the binary cycle is given by

$$\eta = \frac{\text{Work done}}{\text{Heat supplied}} = \frac{W_t}{h_t} = \frac{mW_{hg} + W_s}{mh_{hg_1}} \quad \text{...(12.18)}$$

Thermal efficiency of the mercury cycle is given by

$$\eta_{hg} = \frac{mW_{hg}}{mh_{hg_1}}$$

$$= \frac{W_{hg}}{h_{hg_1}} = \frac{h_{hg_1} - h_{hg_2}}{h_{hg_1}} = 1 - \frac{h_{hg_2}}{h_{hg_1}} \quad \text{...(12.19)}$$

$$= \frac{mh_{hg_1} - h_s}{mh_{hg_1}} = 1 - \frac{1}{m} \cdot \frac{h_s}{h_{hg_1}} \quad \text{...(12.20)}$$

Heat lost by mercury vapour = Heat gained by steam

$$\therefore \quad m\,h_{hg_2} = 1 \times h_s \quad \text{...(12.21)}$$

Substituting the value of m from eqn. (12.21) into eqn. (12.20), we get

$$\eta_{hg} = 1 - \frac{h_{hg_2}}{h_{hg_1}} \quad \text{...(12.22)}$$

The thermal efficiency of the steam cycle is given by

$$\eta_s = \frac{W_s}{h_s} = \frac{h_{s_1} - h_{s_2}}{h_{s_1}} = \frac{h_{s_1} - h_{s_2}}{mh_{hg_2}} \quad \text{...(12.23)}$$

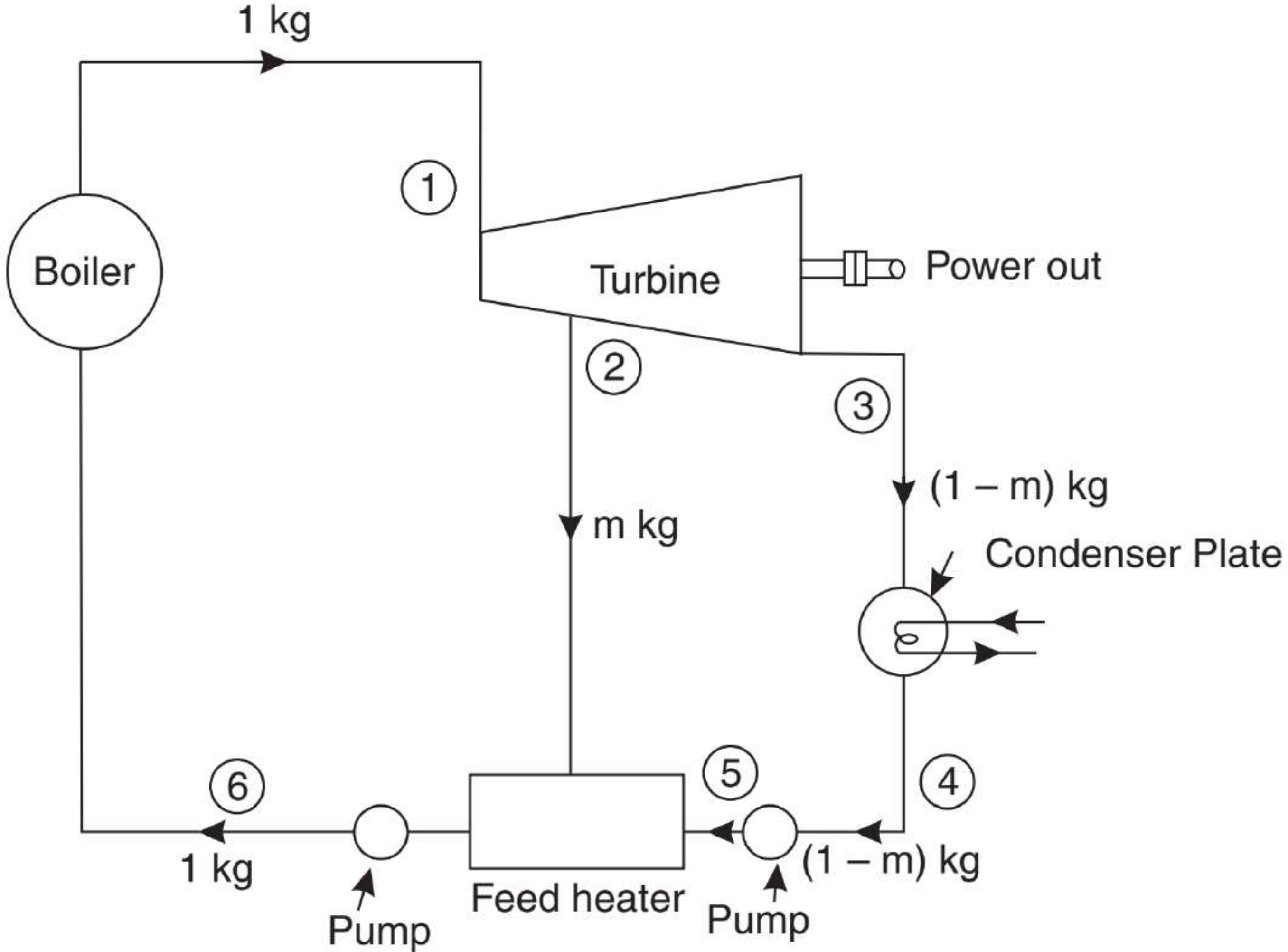
From the eqns. (12.18), (12.20), (12.21), (12.22) and (12.23), we get

$$\eta = \eta_{hg}(1 - \eta_s) + \eta_s \qquad \ldots(12.24)$$

To solve the problems eqns. (12.19), (12.23), (12.24) are used.

In the design of binary cycle, another important problem is the limit of exhaust pressure of

Example *A steam turbine is fed with steam having an enthalpy of 3100 kJ/kg. It moves out of the turbine with an enthalpy of 2100 kJ/kg. Feed heating is done at a pressure of 3.2 bar with steam enthalpy of 2500 kJ/kg. The condensate from a condenser with an enthalpy of 125 kJ/kg enters into the feed heater. The quantity of bled steam is 11200 kg/h. Find the power developed by the turbine. Assume that the water leaving the feed heater is saturated liquid at 3.2 bar and the heater is direct mixing type. Neglect pump work.*



At 3.2 bar, $h_{f_2} = 570.9$ kJ/kg.

Consider m kg out of 1 kg is taken to the feed heater (Fig.).

Energy balance for the feed heater is written as :

$$mh_2 + (1 - m)\ h_{f_5} = 1 \times h_{f_2}$$

$$m \times 2100 + (1 - m) \times 125 = 1 \times 570.9$$

$$2100\ m + 125 - 125\ m = 570.9$$

$$1975\ m = 570.9 - 125$$

∴ $m = 0.226$ kg per kg of steam supplied to the turbine

∴ Steam supplied to the turbine per hour

$$= \frac{11200}{0.226} = 49557.5 \text{ kg/h}$$

Net work developed per kg of steam

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

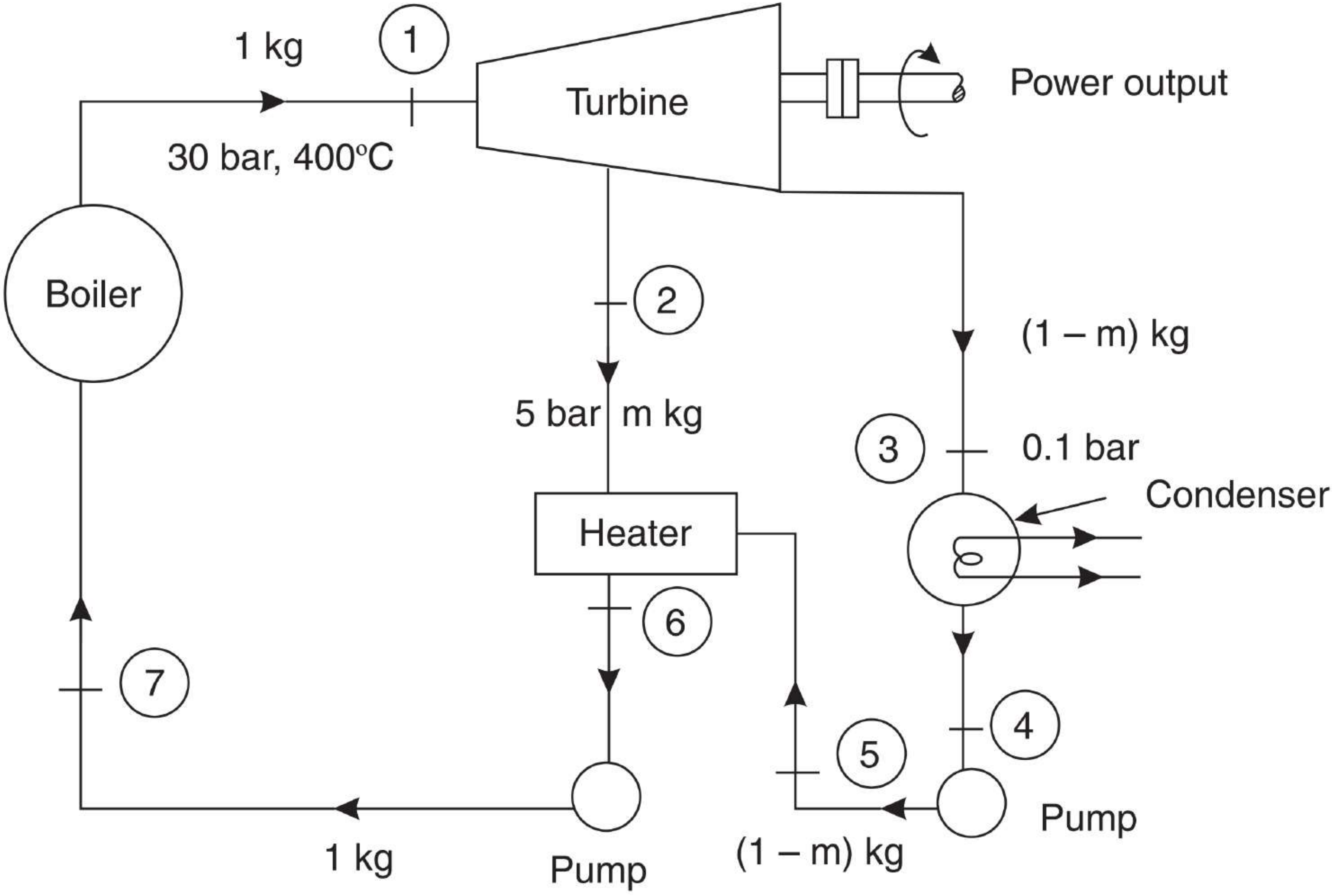
$$= (3100 - 2500) + (1 - 0.226)(2500 - 2100)$$

$$= 600 + 309.6 = 909.6 \text{ kJ/kg}$$

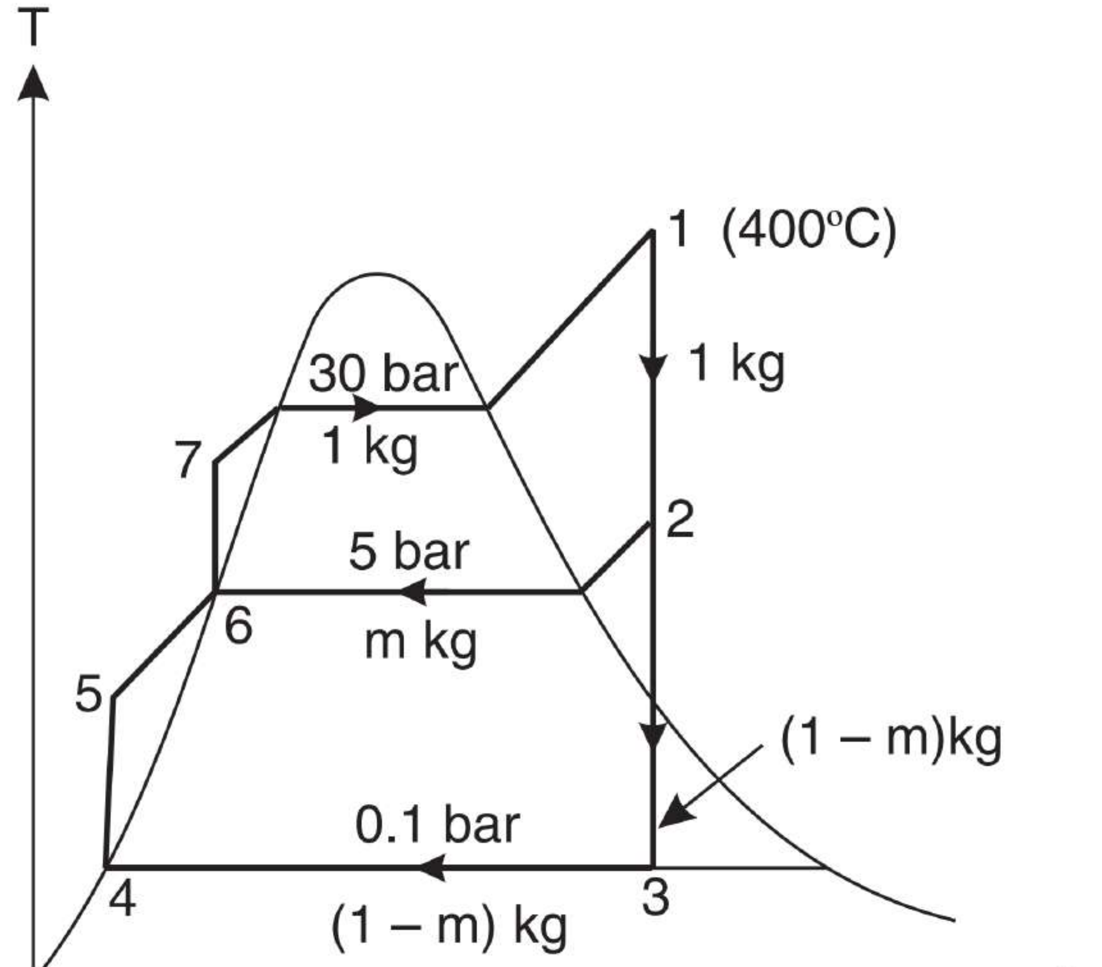
∴ **Power developed by the turbine**

$$= 909.6 \times \frac{49557.5}{3600} \text{ kJ/s}$$

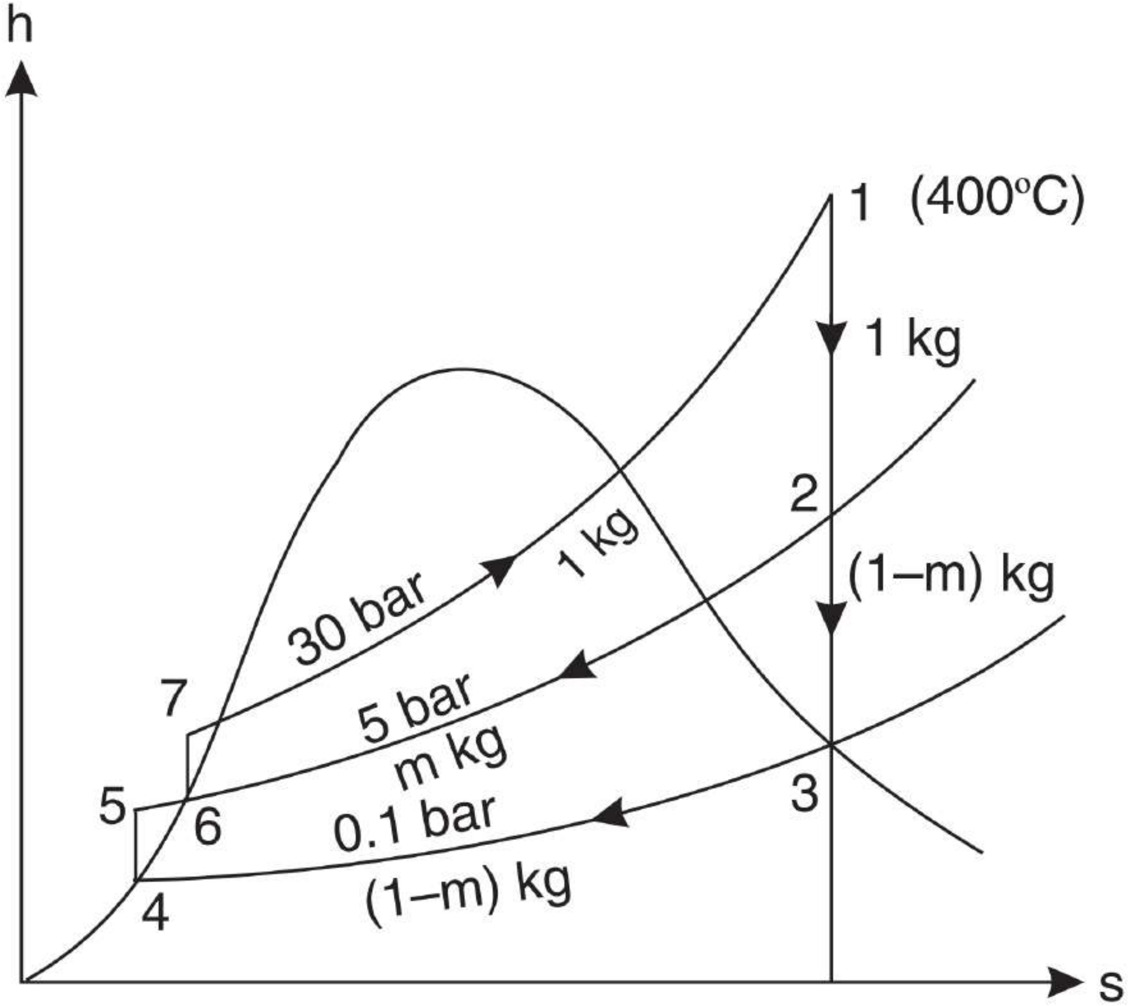
$$= \mathbf{12521.5 \text{ kW.}} \quad \textbf{(Ans.)} \qquad (\because 1 \text{ kJ/s} = 1 \text{ kW})$$


1 kg
①
Turbine
Power output
30 bar, 400ºC
Boiler
②
(1 – m) kg
5 bar
m kg
③
0.1 bar
Condenser
Heater
⑥
⑦
④
⑤
Pump
1 kg
Pump
(1 – m) kg

(*a*)


T
1 (400ºC)
1 kg
30 bar
1 kg
7
2
5 bar
6
m kg
5
(1 – m)kg
0.1 bar
4
3
(1 – m) kg
s

(*b*)


h
1 (400ºC)
1 kg
2
1 kg
(1–m) kg
30 bar
5 bar
7
m kg
5
6
0.1 bar
3
4
(1–m) kg
s

(*c*)