# Chapter 20

# Steam Condensers

**Learning objectives :** Attention has been focussed in this chapter to make the reader conversant with :

- *steam condenser and its utility*
- *elements of a condensing plant and types of condensers*
- *condenser vacuum, vaccum efficiency and condenser efficiency*
- *air leakage and loss of vacuum, construction and working of Edward air pump*
- *cooling towers and cooling ponds*

## 20.1. CONDENSER AND ITS UTILITY

The efficiency of a steam power plant working on Carnot cycle is given by

$$\eta = \frac{T_1 - T_2}{T_2} = 1 - \frac{T_2}{T_1}$$

where $T_1$ is the temperature at which heat is supplied and $T_2$ is the temperature at which heat is rejected. This expression reveals that for obtaining maximum efficiency, it is necessary that the difference between the temperature $T_1$ and the temperature $T_2$ should be as large as possible. The temperature $T_1$ can have the maximum optimum value consistent with metallurgical considerations and so for maximum efficiency, the temperature $T_2$ should have the minimum value. There is a definite relation between steam temperature and pressure. Low exhaust temperature means low exhaust pressure. Steam cannot be exhausted to atmosphere if it is expanded in the steam turbine or engine to a back pressure which is lower than the atmospheric pressure. However, the steam by exchanging heat with water in a vessel can be condensed resulting in a fall in its temperature and pressure. However, the condensation of steam will cause reduction in pressure only when water is contained in a closed vessel. With the vessel open to atmosphere the condensation will of course be there but the pressure will not drop below the atmospheric pressure.

The closed vessel in which steam is condensed by abstraction of heat and in which vacuum is maintained is called *condenser*.

Condensation of steam enables expansion of steam to a lower back pressure. The available enthaply drop increases. Consequently more work is done and the plant efficiency improves.

Due to increased cost involved in the creation and maintenance of higher vacuum, a limit is imposed beyond which the reduction of back pressure does not prove economical. For steam engines the lowest practical exhaust pressure is 65 cm of mercury vacuum. The steam engines being of positive displacement have to provide piston displacement equivalent to volume of exhaust steam and as such the cylinder dimensions impose a limit on the engine condenser pressure. Steam turbines being steady flow machines may be designed to operate at 73.25 cm of mercury vacuum or even more depending upon the capacity of the plant and cooling water available.





The hatched areas in Fig. 20.1(*a*) and 20.1(*b*) represent the increase in work done by a steam engine and a steam turbine by exhausting the steam into a condenser.

The following are the advantages obtained by incorporating a condenser in a steam engine or steam turbine plant :

(1) Improvement in the efficiency of the plant due to increased available enthalpy drop.

(2) Reduction in steam consumption per kW/hour. Increase in vacuum from 71 to 73.5 cm. of Hg gives about 45% reduction in steam consumption.

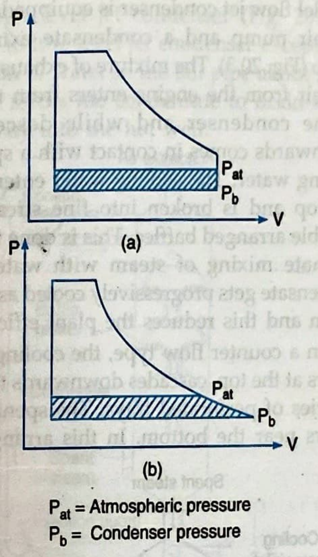


**Fig. 20.1.** Increase in work

(3) The condensed steam, called *condensate* collected in the hot well may be pumped back to the boiler as feed water. Recovery of condensate reduces the make up water that must be added to the system from 100% when non-condensing to 1.5% when condensing. For steam power plants where sufficient quantity of good quality boiler feed water is not available, recovery of condensate is very important. For marine practice where sea water is treated before being used in the boiler, recovery is a necessity.

(4) To prevent the encrustation of boiler, the feed water if not available in pure form has to be treated first in the water softening plant. The recovered condensate reduces the capital and running cost of the water softening plant.

(5) Provision for the supply of hot water to the boiler results in fuel economy and safety from thermal stresses.

## 20.2. ELEMENTS OF A CONDENSING PLANT

Fig. 20.2 shows the essential elements which comprise a condensing unit:

(1) A closed vessel in which steam is condensed.

(2) A condensate pump to extract the condensed steam from the condenser and feed it to hot well.

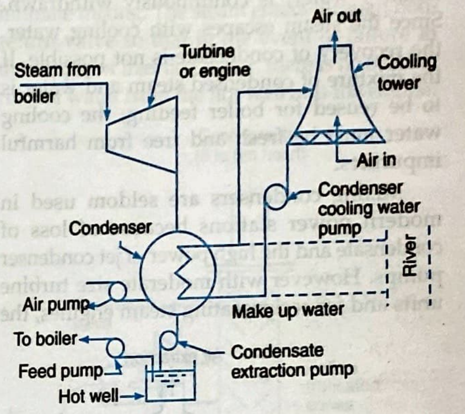


**Fig. 20.2.** Elements of a condensing unit

(3) A dry air pump to remove air and non-condensable gases. Sometimes a single pump known as wet air pump serves to remove both air and condensate.

(4) A feed water pump to force the condensate from the hot well to the boiler.

(5) A cooling water pump for circulating cooling water.

(6) An arrangement such as cooling tower or spray pond to recool the circulating water after it gets heated in the condenser. This is necessary when the supply of cooling water is scarce and same water has to be used over and over again for cooling purposes.

(7) An atmospheric relief valve for relieving the pressure in the condenser when the condenser does not function properly. The steam then escapes through the valve and engine operates as non-condensing.

## 20.3. TYPES OF CONDENSERS

The condensers can be classified into two groups namely jet condensers and surface condensers. These two types are discussed briefly in the following sections.

(A) *Mixing* or *jet condensers*. There is a direct contact between the steam and cooling water and the heat exchange is by direct conduction. The steam quickly condenses in water introduced in the form of spray or jet and the stream of warm water (condensed steam + water) is continuously withdrawn. Since the steam escapes with cooling water, the recovery of condensate is not possible. If the mixture of condensed steam and water is to be reused for boiler feeding, the cooling water must be fresh and free from harmful impurities.

Mixing condensers are seldom used in modern power stations because of loss of condensate and the high power of jet condenser pumps. However with moderate size turbine units and for reciprocating steam engines, the jet condensers are used especially where an abundant supply of good feed water is available.

The jet condensers are divided into:

(*a*) Parallel flow in which the steam and the cooling water flow are in the same direction.

(*b*) Counter flow in which steam flows in the opposite direction to the cooling water.

Depending upon the arrangement of the removal of condensate, the jet condensers are further subdivided into the following three categories:

(*i*) *Low level jet condenser*. A low water level parallel flow jet condenser is equipped with a dry air pump and a condensate extraction pump (Fig. 20.3). The mixture of exhaust steam and air from the engine enters from the top of the condenser and while descending downwards comes in contact with a spray of cooling water. The cooling water enters from the top and is broken into fine streams by suitable arranged baffles. This is done to have intimate mixing of steam with water. The condensate gets progressively cooled as it goes down and this reduces the plant efficiency.

In a counter flow type, the cooling water enters at the top, cascades downwards through a series of perforated trays. The spent steam enters near the bottom. In this arrangement

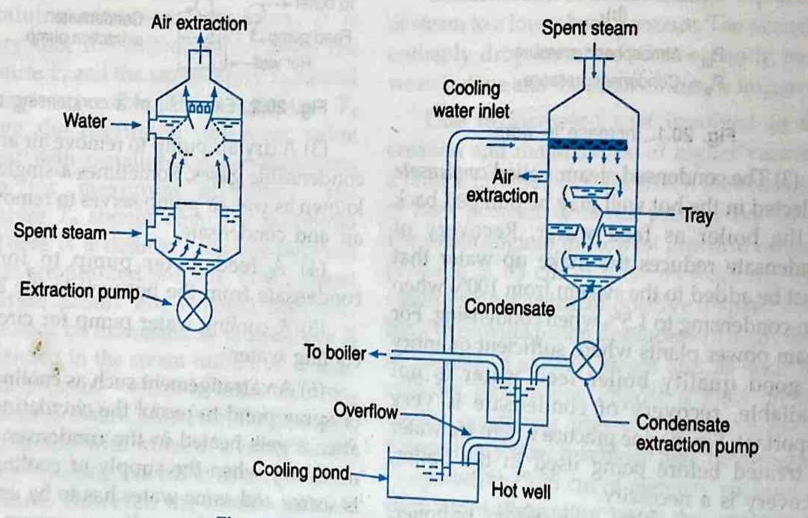

Fig. 20.3. Jet condensers (Low-Level)



there is no under-cooling of condensate and the temperature of the condensate may approach that of the inconing steam. To bring about condensation of steam less quantity of cooling water will be required. With air extraction at the top, the air while ascending up increases in density due to progressive cooling and this reduces the required capacity of air extraction pump.

The vacuum created by air pump is sufficient to draw the cold water into the condenser shell and as such there is no need of cooling water pump.

(*ii*) *Barometric* or *high level jet condenser*. This high level jet condenser (Fig. 20.4) is similar to low level jet condenser except that a column of water in the tail pipe about 10.36 m high forces the condensate to drain away by gravity into the hot well.

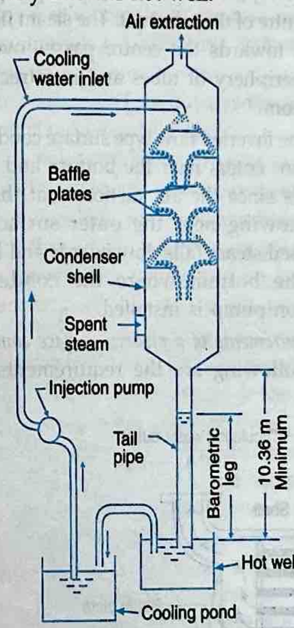

Fig. 20.4. Barometric jet condenser

Theoretically the height $H$ of the tail pipe is given by :

$H$ = (Atmospheric pressure
– condenser pressure)
+ friction loss in tail pipe
+ water velocity head

Unless a supply of fresh water under pressure is available, an injection pump has to be installed to pump the cooling water into the condenser shell. Its installation is appropriate only where enough head required for tail pipe is available.

(*iii*) *Ejector condenser*. The distinctive feature of ejector condenser (Fig. 20.5) is that a momentum of flowing water removes the mixture of condensate and cooling water against the atmospheric pressure.

The cooling water under a head of about 6 m enters the condenser from the top and passes through a series of convergent nozzles. At the nozzle throat the velocity increases with a corresponding reduction in pressure. This reduction in pressure draws the spent steam into the nozzle through ports and provides intimate mixing. The steam enters through non-return valve so that in the event of failure in the supply of injection water, there is no sudden rush of water from the hot well into the engine.

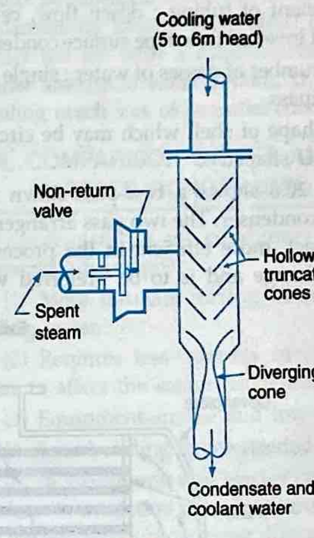

Fig. 20.5. Ejector condenser

In the diverging cone, the kinetic energy is partially transformed into pressure energy. This pressure is enough to overcome the resistance of the atmospheric pressure and so the water discharges automatically into the hot well.

The condenser is used for moderate vacuum and dispenses with extraction pump.

(*B*) *Surface condensers*. The heat is convectively transferred through a wall interposed between steam and water. The steam is drawn across a nest of tubes which are arranged in certain pattern and are maintained at a temperature lower than that of steam by a flow of cooling water through them.

Because the steam and circulating water do not mix, the condensate can be recovered. Moreover cooling water need not be up to a high standard of purity. This fact is of great advantage for ships which can carry only a limited amount of pure treated water for stream raising and use sea water for cooling purposes. All the marine installations are equipped with surface condensers.

The surface condensers may be classified according to:

(*a*) direction of flow of condensate and arrangement of tubing : down flow, central flow and inverted flow type surface condensers

(*b*) number of passes of water : single pass or multipass

(*c*) shape of shell which may be circular, oval or *U* shaped

Fig. 20.6 shows a two pass down flow surface condenser. The two pass arrangement is compact, more efficient in the process of heat exchange and is to be preferred when the supply of cooling water is limited. The suction of the extraction pump, installed at the bottom, causes the spent steam entering from the top to flow downwards over a nest of tubes. The cooling water enters at one end of the bottom-set of tubes, flows through them till it reaches the other end of the shell. The water then rises up and flows in the opposite direction through the upper half of the tubes and finally leaves through the outlet.

A section of tubes near the air pump suction is screened off by providing a baffle [Fig. 20.7(*a*)] This is done to reduce the amount of water vapour going along with air. Moreover the lowest temperature maintained in this section increases the density of air and so we need an extraction pump of a small capacity.

In the central flow type surface condenser [Fig. 20.7(*b*)] the air extraction pump is placed in the centre of the tube nest. The steam flowing radially towards the centre passes over the entire periphery of tubes and is extracted at the bottom.

In the inverted flow type surface condenser, the steam enters near the bottom and flows upwards since the air suction is at the top. After flowing near the outer surface the condensed steam falls downwards and leaves from the bottom where the condensate extraction pump is installed.

*Requirements of a modern surface condenser* : The following are the requirements of a modern surface condenser, sometimes referred to as an ideal condenser :

(1) The steam should be well distributed in the vessel and the pressure drop should be minimum.

(2) The steam should enter the condenser with least possible resistance.

(3) Circulating water should pass through the surface condenser with least friction but at a velocity consistent with high efficiency.

(4) The steam should lose only its latent heat and there should be no condensate depression (undercooling of condensate). To achieve this the quantity of water circulating through the tubes should be so regulated that temperature of leaving water is the same as the saturation temperature of steam.

(5) The water is to flow inside the tubes and vapour outside so that outside surface of the tubes (which is rather difficult to clean) does not get deposited with sediments. The cooling water, if dirty, would leave such deposits on the inside of the tubes. By removing the end cover plates and passing motor driven brushes, the inside surface of tubes can be cleaned.

(6) There should be no leakage of air into the condenser and if any, arrangements must be made to extract it rapidly and with least expenditure of mechanical energy. The presence of air in the condenser would destroy the vacuum and will hamper the rate of heat transfer from the steam to the coolant because of its poor conductivity.

(7) The air extraction should be at the coolest section of the tubes and the air exit should be shielded from the down flowing condensate by means of a baffle. This is to extract the air with a comparatively much smaller amount of water vapour, *i.e.*, without entailing much loss of potential condensate.

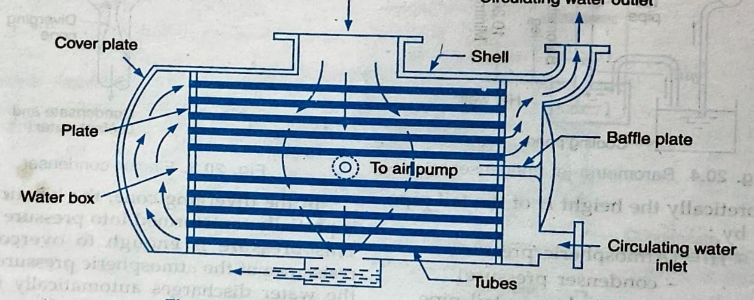


Fig. 8.6. Two-pass downflow surface condenser



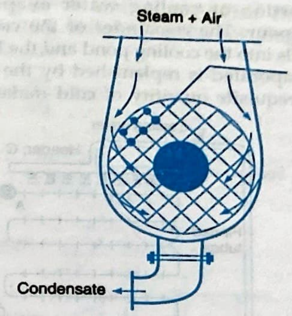


Fig. 8.7. (a) Central flow surface condender

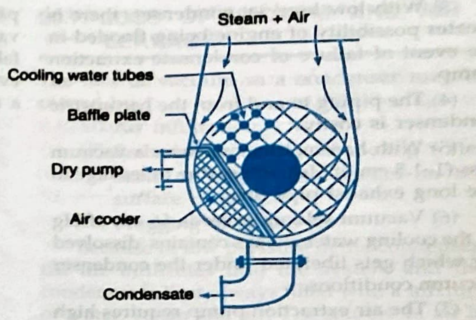


Fig. 8.7. (b) Down flow surface condender

## 20.4. COMPARISON OF JET AND SURFACE CONDENSERS

*Jet condenser* : A jet condenser has the following advantages :

(1) More intimate mixing of steam and cooling water.

(2) Requires less quantity of circulating water to affect the steam condensation.

(3) Equipment simple and low in cost.

(4) Less building space needed.

(5) In barometric and ejecter condensers, condensate extraction pump is dispensed with and the low level jet condenser does not require cooling water pump.

The disadvantage of jet condenser are :

(1) There is waste of condensate.

(2) If the condensate has to be salvaged, the cooling water should be clean and free from harmful impurities.

(3) With low level jet condenser, there is greater possibility of engine being flooded in the event of failure of condensate extraction pump.

(4) The piping to and from the barometric condenser is costly.

(5) With barometric condenser a vacuum loss (1–1.5 cm of Hg) occurs due to leakage in the long exhaust pipe line.

(6) Vacuum seldom exceeds 66 cm of Hg as the cooling water always contains dissolved air which gets liberated under the condenser vacuum conditions.

(7) The air extraction pump requires high power which may be about two times of that required for surface condenser.

*Surface condenser* : A surface condenser has the following advantages :

(1) High vacuum can be attained and greater plant efficiency is achieved.

(2) Any kind of cooling water can be used and so the cost of water softening plant is considerably reduced.

(3) Chances of losing vacuum are minimum because water supply is not affected by drop in vacuum.

(4) The condensate can be salvaged and used for boiler feed.

(5) The arrangement can be conveniently adapted to weigh the condensate for tests in laboratory.

(6) Suitable for high capacity units.

The surface condenser is, however, bulky and so requires considerable floor space for its erection. Moreover, the unit has a high capital and maintenance costs.

## 20.5. EVAPORATIVE CONDENSER

The steam to be condensed enters a coiled finned pipe system at *A* (Fig. 20.8). The water from a cooling pond is pumped to a horizontal header *C* which is fitted with spray nozzles. The cooling water sprayed over the finned tubes gets evaporated. The steam loses its heat both to the cooling water and the current of air circulating over the water film. The heated air moves upwards carrying along with it a portion of cooling water evaporated into vapour. The remainder of the cooling water falls into the cooling pond and the loss of water evaporated is replenished by the addition of a requisite quantity of cold make up water.

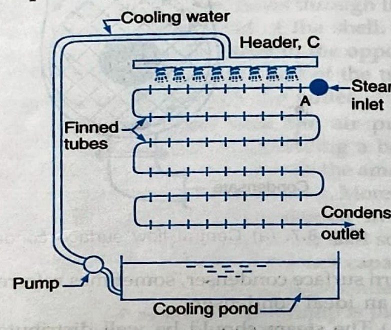

**Fig. 20.8.** Evaporative condenser

The arrangement is simple, cheap, does not require large quantity of cooling water and so needs a cooling water pump of small capacity.

The vacuum created is, however, not so high as in the surface condenser. Moreover the cloud of evaporated water is also a nuisance to the surroundings.

## 20.6. CONDENSER VACUUM

The vacuum obtainable from a condenser depends upon tightness of valves and joints, amount of air infiltration and the temperature of steam after condensation. The vacuum is not uniform throughout the condenser, being least at the air pump suction, high in the body of the condenser and still higher at the engine exhaust valve.

The degree of vacuum measured by means of a vacuum gauge (Fig. 20.9) can be expressed in the following different ways :

(*a*) Excess of atmospheric pressure over observed vacuum. A 65 cm of vacuum means that the pressure of atmosphere is 65 cm of Hg above the condenser pressure.

(*b*) Percentage of vacuum, *i.e.*, ratio of observed vacuum to atmospheric pressure. If the gauge reads 65 cm of Hg with the barometer reading standing at 75 cm, then percentage vacuum

$$= \frac{65}{75} \times 100$$

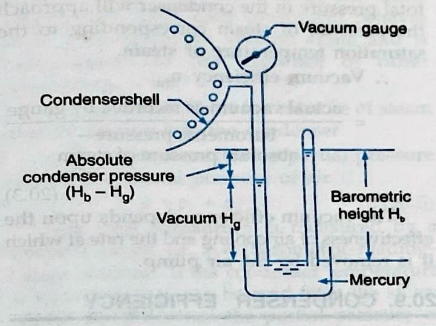

**Fig. 20.9.** Vacuum measurement

(*c*) Absolute pressure

Let $H_b$ = barometric height in cm of Hg

$H_g$ = vacuum gauge reading in cm of Hg

Then absolute pressure in the condenser

$= (H_b - H_g)$ cm of Hg

The vacuum in the condenser is thus a function of both absolute pressure as well as the barometric pressure. The barometric height is a variable quantity, changing from place to place. Accordingly the gauge reading would also vary, if the absolute pressure is to remain constant. For purposes of comparison it is more convenient to refer the vacuum readings to a standard barometric height of 76 cm of Hg.

$$\therefore \text{ Corrected vacuum} = 76 - (H_b - H_g) \quad ...(20.1)$$

Standard atmospheric pressure

$= 76$ cm of Hg $= 1.01325$ bar

$\therefore$ Pressure equivalent of 1 cm of Hg

$$= \frac{1.01325}{76} = 0.01333 \text{ bar}$$



## 20.7. LOSS OF VACUUM AND AIR LEAKAGE

The loss of vacuum in a condenser may be caused by :

(*i*) Air infiltration

(*ii*) Reduced circulation of cooling water

(*iii*) Scale or lime accumulation on the surface of tubes

(*iv*) Plugging of ejector jet

Because of low pressure in the condenser, the air infiltrates into the system and the condenser is thus always filled with a mixture of water, steam and air. The sources of air in a condenser are:

(*i*) Leakage through packing glands and microscopic holes in the shell and the joints

(*ii*) Leakage through vents from atmospheric relief valve and other accessories

(*iii*) In jet condenser, the air enters with injection water and gets liberated at low pressure

(*iv*) The air contained in spent up steam is also liberated at low pressure.

In a well designed and properly maintained surface condenser, the amount of air leakage is about :

= 5 kg per 10,000 kg of steam condensed for a steam turbine plant.

= 15 kg per 10,000 kg of steam condensed for a steam engine plant.

The presence of air in the condenser affects its performance in the following ways :

(1) Back pressure in the stream power plant increases and correspondingly the work output decreases.

(2) For the same absolute pressure the partial pressure of steam decreases with air infiltration. From steam tables, we see that at lower pressures the steam has more latent heat. To remove this greater quantity of heat, more cooling water has to be supplied.

(3) Because of poor thermal conductivity of air, the rate of heat transfer from the vapour is reduced and the surface area of the tubes has to be increased for a condenser duty.

(4) The abstraction of heat by the cooling water is partly from steam and partly from air. This aspect also reduces the rate of steam condensation.

(5) For the maintenance of proper vacuum, we have to install an air extraction pump which would remove the air continuously from the shell. Inspite of shielding the air extraction section, some steam does escape with air resulting in loss of high temperature feed water to the boiler. Moreover, the condensate is under-cooled and more heat has to be supplied to water in the boiler. Loss of condensate and its undercooling lowers the overall plant efficiency.

The following procedure is adopted to check whether there is air leakage in the condenser.

(*i*) Run the plant until the temperature and pressure conditions are steady in the condenser.

(*ii*) Isolate the condenser by shutting off steam supply and simultaneously closing the condensate and air extraction pumps.

If there is a leakage, the readings of vacuum gauge and thermometer will record a fall.

Source of air leakage can be checked by adopting the following methods:

(*i*) The condenser is put under air pressure and its effect on soap water is noted at the points where infiltration is likely to occur.

(*ii*) When the condenser is operating, pepprament oil is put on the suspected joint and a check is made on the pepprament odour in the air ejector discharge.

(*iii*) With condenser under vacuum, large leakages can be detected by passing a candle flame over possible openings.

## 20.8. VACUUM EFFICIENCY

The vacuum efficiency is a measure of the degree of perfection in achieving the aim of maintaining a desired vacuum in condenser. The vacuum efficiency may be defined as the ratio of actual vacuum as recorded by the vacuum gauge to the ideal vacuum.

$$\text{Vacuum efficiency} = \eta_{vac} = \frac{\text{actual vacuum as recorded by gauge}}{\text{ideal vacuum}} \qquad ...(20.2)$$

The ideal vacuum means the vacuum due to steam alone when air is absent. In that case total pressure in the condenser will approach the pressure of steam corresponding to the saturation temperature of steam.

$\therefore$ Vacuum efficiency $\eta_{vac}$

$$= \frac{\text{actual vacuum as recorded by gauge}}{\text{barometric pressure} - \text{absolute pressure of steam}} \qquad ...(20.3)$$

The vacuum efficiency depends upon the effectiveness of air cooling and the rate at which it is removed by the air pump.

## 20.9. CONDENSER EFFICIENCY

The purpose of an ideal condenser is to remove only the latent heat so that temperature of condensate equals the saturation temperature corresponding to condenser pressure. In other words there is no under cooling of condensate. Further, the maximum temperature to which the cooling water can be raised is the condensate temperature. The condenser efficiency is then defined as the ratio of actual rise in the temperature of cooling water to the maximum possible rise.

$$\text{Condenser efficiency} = \frac{t_2 - t_1}{t_3 - t_1} \qquad ...(20.4)$$

where $t_3$ = saturation temperature corresponding to condenser pressure, $t_2$ and $t_1$ are the outlet and inlet temperatures of cooling water.

## 20.10. DALTON'S LAW AND AIR-VAPOUR MIXTURE

Dalton's law forms the basis for the analytical treatment of the problems dealing with mixture of gases or of gas and vapour. The law states:

*"The total pressure exerted by a mixture of gases or a mixture of gas and vapour (which have no chemical action on each other) is equal to the sum of partial pressure of the constituents".*



The partial pressure of each constituent of the mixture is the pressure exerted by the constituent taken separately if the quantity of the constituent occupies alone the same volume as that of the mixture and at the same temperature. Thus each constituent behaves as if it occupies the volume alone and is independent of the presence of other constituent.

In a condenser, there is a mixture of steam and air leaking into the condenser.

$\therefore$ Pressure of mixture = partial pressure of steam + partial pressure of air

$$p = p_s + p_a \qquad ...(20.5)$$

The total pressure $p$ is measured by a vacuum gauge, *i.e.*, $p$ = (barometric reading– gauge reading). If the condenser temperature is known than $p_s$ can be read from the steam tables. By difference the partial pressure of air $p_a$ can be obtained.

### EXAMPLE 20.1

*The following observations were recorded during a test on a steam condenser :*

*Recorded condenser vacuum* = 71 *cm of Hg*

*Barometric reading* = 76.5 *cm of Hg*

*Mean condenser temperature* = 34 °C

*Temperature of hot well* = 28.5°C

*Condensate collected* = 1800 *kg/hour*

*Flow rate of cooling water* = 57500 *kg/hour*

*Inlet temperature of cooling water* = 8.5°C

*Outlet temperature of cooling water* = 26°C

*Calculate: (a) vacuum corrected to standard barometer of 76 cm of Hg, (b) vacuum efficiency, (c) under cooling of condensate, (d) condenser efficiency, (e) state of steam entering the condenser and (f) mass of air present per cubic meter of condenser volume and the mass of air present per kg of uncondensed steam.*

**Solution :** (*a*) Corrected vacuum

$= 76 - (H_b - H_g)$

$= 76 - (76.5 - 71) =$ **70.5 cm of Hg**

(*b*) From steam tables, the absolute pressure of steam corresponding to condenser temperature of 34°C

= 0.053 bar

$= \frac{0.0532}{0.01333} = 3.99$ cm of Hg

Vacuum efficiency $\eta_{vac}$

$$= \frac{\text{actual vacuum as recorded by gauge}}{\text{barometric pressure} - \text{absolute pressure of steam}}$$

$$= \frac{71}{76.5 - 3.99}$$

$$= \frac{71}{72.51} = 0.979 \simeq \mathbf{98\%}$$

(*c*) Condensate under-cooling

= condenser temperature – temperature of hot well

= 34 – 28.5 = **5.5°C**

(*d*) Absolute condenser pressure

= barometer reading – vacuum reading

= 76.5 – 71 = 5.5 cm of Hg

= 5.5 × 0.01333 = 0.0733 bar

Saturation temperature corresponding to 0.0733 bar = 39.95°C

Therefore the maximum temperature to which cooling water can be raised is 39.95°C.

Condenser efficiency

$$= \frac{\text{actual rise in temperature of cooling water}}{\text{maximum permissible temperature rise}}$$

$$= \frac{26 - 8.5}{39.95 - 8.5} = \mathbf{55.7\%}$$

(*e*) From steam tables for a pressure of 0.0733 bar

$h_f$ = 167.23 kJ/kg

and $h_{fg}$ = 2406 kJ/kg

The enthaply of condensate ($h_c$) corresponding to hot well temperature of 28.5°C is 119.4 kJ/kg.

Heat absorbed by cooling water

= Heat given up by steam

$$m_w c_w(t_2 - t_1) = m_3\left[(h_f + h_{fg}) - h_c\right]$$

$$57500 \times 4.186\ (26.0 - 8.5)$$
$$= 1800\ [(167.23 + x \times 2406) - 119.4]$$

or $2340 = 2406 + 47.83$

$$\therefore \quad x = \frac{2340 - 47.83}{2406} = 0.953$$

Thus steam at entry to the condenser has a dryness fraction of 0.953.

(*f*) At condenser temperature of 34°C,

Partial pressure of steam,

$$p_s = 0.0532 \text{ bar}$$

$\therefore$ From Dalton's law, partial pressure of air

$$p_a = 0.0733 - 0.0532 = 0.0201 \text{ bar}$$

From characteristic gas equation,

$$pV = mRT$$

$\therefore$ Mass of air present per m$^3$ of condenser volume

$$= \frac{0.0201 \times 10^5 \times 1}{287 \times 307} = 0.0228 \text{ kg}$$

Volume of 1 kg of steam at 34°C = 26.61 m$^3$

Air associated with 1 kg of steam will have the same volume.

$\therefore$ Mass of air present per kg of uncondensed steam

$$= \frac{0.0201 \times 10^5 \times 26.61}{287 \times 307} = 0.607 \text{ kg}$$

**EXAMPLE 20.2**

*A condenser deals with 1000 kg of steam per hour with a dryness fraction of 0.9. The mean condenser temperature is 45°C. The air associated with the steam in the condenser is 250 kg/hour. What would be the vacuum reading? Barometer reading is 75cm of Hg. Correct this vacuum to a standard barometer reading of 76 cm of mercury.*

**Solution :** From steam tables :

Partial pressure of steam at 45°C,

$$p_s = 0.0958 \text{ bar}$$

Volume of 1 kg of steam at 45°C,

$$v_s = 15.28 \text{ m}^3$$

$\therefore$ Total volume of steam

$$= m \times (x \times v_s)$$
$$= 1000(0.9 \times 15.28)$$
$$= 13752 \text{ m}^3/\text{hr}$$

According to Dalton's law this is also the volume of 250 kg of air at 45°C.

From characteristic gas equation,

$$pV = mRT$$

partial pressure of air,

$$p_a = \frac{250 \times 287 \times (45 + 273)}{13752}$$
$$= 1659 \text{ N/m}^2 = 0.01659 \text{ bar}$$

$\therefore$ Total condenser pressure

$$= 0.0958 + 0.01659 = 0.11239 \text{ bar}$$
$$= \frac{0.11239}{0.01333} = 8.43 \text{ cm of Hg}$$

$\therefore$ Vacuum reading

$$= 75 - 8.43 = 66.57 \text{ cm of Hg}$$

Corrected vacuum

$$= 76 - (75 - 66.57)$$
$$= 67.57 \text{ cm of Hg}$$

**EXAMPLE 20.3**

*In a surface condenser test, the vacuum gauge reading was 65 cm of Hg when the barometer read 75 cm of Hg. The temperature in the condenser was 50°C. If the quantity of air entering the condenser was reduced to 80kg/hr by suitable means, find the resultant alteration in vacuum. If the condensate weighs 1000kg/hour, calculate quantities of air and vapour which the pump has to deal with.*

**Solution :** Absolute condenser pressure

$$= (76 - 65) \times 0.01333 = 0.1333 \text{ bar}$$

From steam tables, at 50°C

partial pressure of steam

$$= 0.1233 \text{ bar}$$

and volume of 1 kg of steam

$$= 12.04 \text{ m}^3$$

$\therefore$ Partial pressure of air

$$= 0.1333 - 0.1233 = 0.01 \text{ bar}$$

volume of steam/hr

$$= 12.04 \times 1000 = 12040 \text{ m}^3$$

This is also the volume of air



$\therefore$ Mass of air,

$$m = \frac{pV}{RT} = \frac{0.01 \times 10^5 \times 12040}{287 \times 327}$$
$$= 128.29 \text{ kg/hr}$$

When the mass of air is reduced to 80 kg/hr, the partial pressure of air will become

$$= \frac{80}{128.29} \times 0.01 = 0.00623 \text{ bar}$$

$\therefore$ Absolute pressure of condenser will change to

$$0.1233 + 0.00623 = 0.12953 \text{ bar}$$
$$= \frac{0.12953}{0.01333}$$
$$= 9.717 \text{ cm of Hg}$$

New vacuum reading

$$= 75 - 9.717$$
$$= 65.283 \text{ cm of Hg}$$

$\therefore$ Alteration of vacuum

$$= 65.283 - 65$$
$$= 0.283 \text{ cm of Hg}$$

Capcaity of pump

$$= 1\,000 + 80 = 1080 \text{ kg/hr}$$

**EXAMPLE 20.4**

*A steam turbine discharges 5000 kg/hr of steam at 40°C and 0.85 dry. The air leakage in the condenser is estimated to be 15 kg/hr. The temperature at the suction of air pump is 32°C and the temperature of condensate is 35°C. Determine : (a) vacuum gauge reading, (b) capacity of air pump, (c) loss of condensate in kg/hr and (d) quantity of cooling water if its temperature rise is limited to 10°C.*

**Solution :** At 40°C temperature

$$v_s = 19.54 \text{ m}^3/\text{kg}$$
$$p_s = 0.0737 \text{ bar}$$
$$h_f = 167.6 \text{ kJ/kg}$$

and $h_{fg} = 2406.7$ kJ/kg

Then enthalpy of steam being discharged by the turbine is

$$h_1 = h_f + x\, h_{fg}$$
$$= 167.6 + 0.85 \times 2406.7$$
$$= 2213.29 \text{ kJ/kg}$$

For 1 kg of steam in the condenser,

Specific volume

$$v_1 = 0.85 \times 19.54 = 16.609 \text{ m}^3/\text{kg}$$

This is also the volume occupied by

$$\frac{15}{5000} = 3 \times 10^{-3} \text{ kg of air}$$

Applying characteristic gas equation

$pv = mRT$, we get

Partial pressure of air in the condenser

$$p_a = \frac{3 \times 10^{-3} \times 287 \times (273 + 40)}{16.609}$$
$$= 16.22 \text{ N/m}^2 = 1.622 \times 10^{-4} \text{ bar}$$

Then absolute pressure in the condenser

$$p = p_a + p_s$$
$$= 1.622 \times 10^{-4} + 0.0737$$
$$= 0.07386 \text{ bar}$$
$$= \frac{760}{1.01327} \times 0.07386$$
$$= 55.4 \text{ mm of Hg}$$

(*a*) Taking standard atmospheric pressure

$$= 760 \text{ mm of Hg}$$

Vacuum gauge reading

$$= 760 - 55.4 = 704.6 \text{ mm of Hg}$$

(*b*) At the air pump suction where temperature is 32°C

Steam vapour pressure $p_s$ = 0.0476 bar

$\therefore$ Partial pressure of air

$$= p - p_s$$
$$= 0.07386 - 0.0476 = 0.02626 \text{ bar}$$

Capacity air pump

$$V = \frac{mRT}{p}$$
$$= \frac{15 \times 287 \times (273 + 32)}{0.02626 \times 10^5} = 500 \text{ kg/hr}$$

This is also the volume of vapour mixed with air.

(*c*) The loss of condensate in air pump equals the mass of water for the volume (capacity) of air pump. Then

$$m_w = \frac{p_s V}{R_w T}$$
$$= \frac{(0.0476 \times 10^5) \times 500}{\left(\frac{8314}{18}\right) \times (273 + 32)} = 16.89 \text{ kg/hr}$$

(*d*) From heat energy balance
heat lost by steam = heat gained by water

Corresponding to 35°C temperature, enthalpy of condensate = 146.7 kJ/kg

Then : $5000(2213.29 - 146.7)$

$$= m_w \times 4.186 \times 10$$

∴ Quantity of cooling water

$$= \frac{5000(2213.29-146.7)}{4.186\times 10}$$

$$= 246845 \text{ kg/hr}$$

### EXAMPLE 20.5

*The following procedure was adopted to check the leakage of air into a condenser.*

*After running the plant to reach steady state conditions, the condenser was completely isolated by shutting down the steam supply and also the condensate and air extraction pumps. The temperature and vacuum readings were noted to be 39°C and 68.5 cm mercury at shut down, and 28°C and 27 cm mercury respectively 5 minutes after the shut down. If effective volume of the condenser is 1 m³, calculate*

*(a) the quantity of air leakage into the condenser during the period of observation and*

*(b) the quantity of water vapour condensed during the period.*

*The barometer is stated to read 75 cm of mercury.*

**Solution :** (*i*) At shut down :

Absolute pressure in the condenser

$$= (75 - 68.5) = 6.5 \text{ cm of Hg}$$

$$= \frac{6.5}{760} \times 1.01325 = 0.08665 \text{ bar}$$

Steam vapour pressure corresponding to 39°C = 0.06999 bar

∴ Partial pressure of air

$$p_a = 0.08665 - 0.06999$$

$$= 0.01666 \text{ bar}$$

Mass of air present in the condenser volume of 1 m³

$$m_{a_1} = \frac{p_a \times V}{RT}$$

$$= \frac{0.01666\times 10^5 \times 1}{287(273+39)} = 0.0186 \text{ kg}$$

Specific volume of steam at 39°C

$$= 20.54 \text{ m}^3/\text{kg}$$

∴ Mass of vapour (steam) present

$$m_{s_1} = \frac{1}{20.54} = 0.0487 \text{ kg}$$

(*ii*) *Five minutes after shut down :*

Absolute pressure

$$= (75 - 27) = 48 \text{ cm of Hg}$$

$$= \frac{48}{760} \times 1.01325 = 0.63984 \text{ bar}$$

Steam vapour pressure corresponding 28°C

$$= 0.03782 \text{ bar}$$

∴ Partial pressure of air

$$= 0.63984 - 0.03782 = 0.6020 \text{ bar}$$

Mass of air present in the condenser volume of 1 m³

$$m_{a_2} = \frac{0.6020\times 10^5 \times 1}{287(273+28)} = 0.697 \text{ kg}$$

Specific volume of steam at 28°C

$$= 36.69 \text{ m}^3/\text{kg}$$

∴ Mass of vapour present

$$m_{s_2} = \frac{1}{36.69} = 0.0272 \text{ kg}$$

That gives :

Mass of air leaked in 5 minutes

$$= 0.697 - 0.0186 = 0.6784 \text{ kg}$$

Mass of steam condensed in 5 minutes

$$= 0.0487 - 0.0272 = 0.0215 \text{ kg}$$

### EXAMPLE 20.6

*A steam turbine develops 2500 kW using 15000 kg of steam per hour which is supplied at a pressure of 30 bar and 300°C. The exhaust from the turbine is condensed at a vacuum of 725 mm of mercury while the barometer reading is 758 mm of mercury. The condensate is removed from the condenser at a temperature of 28°C and the temperature of circulating water increases from 7 to 27°C. Assuming that there is no heat loss by convection and radiation, determine*

*(a) dryness fraction of steam entering the condenser*

*(b) mass circulating water/hr and the cooling ratio*



*(c) minimum amount of cooling water required per kg of steam*

*(d) overall thermal efficiency*

**Solution :** Enthalpy of steam at 30 bar and 300°C,

$$h_1 = 2995 \text{ kJ/kg}$$

Enthalpy drop

$$(h_1 - h_2) = \text{heat equivalent of work done}$$

$$= 2500 \times 3600$$

$$= 90 \times 10^5 \text{ kJ/hr} \quad ...(i)$$

Condenser pressure

$$p = (758 - 725) = 33 \text{ mm of mercury}$$

$$= \frac{33}{760} \times 1.01325 = 0.04399 \text{ bar}$$

Corresponding to 0.04399 bar, it is noted from steam tables that :

$$h_f \equiv 128 \text{ kJ/kg and } h_{fg} \equiv 2430 \text{ kJ/kg}$$

(*a*) If $x$ is the dryness fraction of steam entering the condenser then enthalpy drop in the turbine

$$15000\left[2995 - (128 + 2430x)\right] \text{ kJ/hr} \quad ...(ii)$$

From identities (*i*) and (*ii*), we get that

$$15000\left[2995-(128+2430)x\right] = 90 \times 10^5$$

$$\text{or } 2430x = 2995 - \frac{90\times 10^5}{1500} - 120 = 2067$$

∴ Dryness fraction of steam

$$= \frac{2275}{2430} = \mathbf{0.936}$$

(*b*) From heat energy balance,

heat lost by steam

= heat gained by water

$$m_s\left[h_f + x h_{fg} - h_w\right] = m_w \times c_{pw} \times \Delta T$$

where $h_w$ is the enthalpy of condensate at 28°C = 117.2 kJ/kg

$$15000\left[(128+0.936\times 2430)-117.2\right]$$

$$= m_w \times 4.186 \times 20$$

Solution gives : cooling water flow rate

$$m_w = \mathbf{409450 \text{ kg/hr}}$$

Cooling ratio represents the amount of cooling water required per kg of steam

$$\therefore \text{Cooling ratio} = \frac{409450}{15000} = \mathbf{27.297}$$

(*c*) Maximum possible temperature of cooling water at outlet may correspond to saturation temperature corresponding to condenser pressure of 0.04399 bar which is approximately 30.7°C.

Then : maximum rise in temperature of circulating water

$$= 30.7 - 7 = 23.7°C$$

Cooling water flow rate

$$= 409450 \times \frac{20}{23.7} = 345527 \text{ kg/hr}$$

$$= \frac{345527}{15000} = \mathbf{23.03 \text{ kg/kg of steam}}$$

(*d*) Work output per kg of steam

$$= \frac{2500\times 3600}{15000} = 600 \text{ kJ}$$

Heat added per kg of steam

$$= 2995 - 117.2 = 2877.8 \text{ kJ}$$

∴ Overall efficiency

$$= \frac{600}{2877.8} = 0.2085 \text{ or } \mathbf{20.85\%}$$

### EXAMPLE 20.7

*A surface condenser, fitted with separate air and condensate extraction pump deals with 12000 kg of steam per hour at a pressure of 7.5 kN/m². The steam entering the condenser is 0.95 dry and the temperature at the air and condensate extraction pumps is 30°C. If the air leakage is estimated to be 10 kg/hr, make calculations for*

*(a) the surface area required if the average rate of heat transfer is 4 kJ/cm²/s.*

*(b) the cylinder diameter of the single acting reciprocating dry pump if it runs at 75 rpm. The volumetric efficiency of the pump is 0.9 and the stroke equals 1.25 times the cylinder bore.*

**Solution :** Mass of steam dealt with

$$= \frac{12000}{3600} = 3.33 \text{ kg/s}$$

From steam tables : At 7.5 kN/m² or 0.075 bar pressure

$$h_f = 169 \text{ kJ/kg}$$

$$\text{and} \quad h_{fg} = 2405 \text{ kJ/kg}$$

Further, at 30°C temperature :

$$h_{f1} = 126 \text{ kJ/kg}$$

Then : heat lost by steam

$$= m_s\left[(h_f + x f_{fg}) - h_{f1}\right]$$
$$= 3.33\,[(169 + 0.95 \times 2405) - 126]$$
$$= 7751.4 \text{ kJ/s}$$

$\therefore$ Surface area required

$$= \frac{7751.4}{4} = 1937.85 \text{ cm}^2$$

(b) Total pressure of air and steam in the condenser = 0.075 bar

Partial pressure of steam at 30°C temperature,

$$p_s = 0.0425$$

$\therefore$ Partial pressure of air,

$$p_a = 0.075 - 0.0425 = 0.0325 \text{ bar}$$

Mass of air leakage

$$= \frac{10}{60} = 0.1667 \text{ kg/min.}$$

Then from characteristic gas equation, the volume of air dealt with

$$V_a = \frac{m_a R_a T_a}{p_a}$$
$$= \frac{0.1667 \times 287 \times (273+30)}{0.0325 \times 10^5}$$
$$= 4.46 \text{ m}^3\text{/min}$$

Air volume dealt per stroke

$$= \frac{4.46}{0.9 \times 75}$$
$$= 0.0661 \text{ m}^3 = 66.1 \times 10^3 \text{ cm}^3$$

The volume of air dealt per stroke of pump is also given by $\frac{\pi}{4}\, d^2 l$

$$\therefore \frac{\pi}{4}\, d^2 \times 1.25\, d = 66.1 \times 10^3\, ; \; d^3 = 67.34 \times 10^3$$

Hence the diameter of dry air pump,

$$d = \left[67.34 \times 10^3\right]^{\frac{1}{3}} = 40.53 \text{ cm}$$

## EXAMPLE 20.8

*At a thermal power station, a surface condenser is required to deal with 15000 kg of steam per hour, and the air leakage is estimated at 0.6 kg per 1500 kg of steam. At the air pump suction, the vacuum reading is 705 mm of water and the temperature is 35°C. Determine :*

*(a) the discharge capacity of wet air pump which removes both air and condensate. The volumetric efficiency of the pump is 92 percent.*

*(b) the dimensions of the pump. The pump is single acting, runs at 60 rev/min and the piston stroke equals 1.25 times the pump diameter.*

**Solution :** Total pressure of steam and air in the condenser, *i.e.*, the condenser pressure is

$$p = 760 - 705$$
$$= 55 \text{ mm of mercury}$$
$$= \frac{55}{760} \times 1.01325 = 0.0733 \text{ bar}$$

Corresponding to 35°C, the partial pressure of steam

$$p_s = 0.0562 \text{ bar} \quad \text{(from steam tables)}$$

$\therefore$ Partial pressure of air

$$p_a = 0.0733 - 0.0562 = 0.0171 \text{ bar}$$

Mass of air leakage into the condenser

$$m_a = \frac{0.6}{1500} \times \frac{15000}{60} = 0.1 \text{ kg/min}$$

Then from characteristic gas equation, the volume of air leakage is

$$V_a = \frac{m_a RT}{p_a}$$
$$= \frac{0.1 \times 287 \times (35+273)}{0.0171 \times 10^5}$$
$$= 5.169 \text{ m}^3\text{/min}$$

Mass of steam condensed

$$m_s = \frac{15000}{60} = 250 \text{ kg/min}$$

Taking density of water as 1000 kg/m³, the volume of condensate is

$$V_c = \frac{250}{1000} = 0.25 \text{ m}^3\text{/min}$$

$\therefore$ Volume of air-condensate mixture to be dealt with

$$= 5.169 + 0.25 = 5.419 \text{ m}^3\text{/min}$$

Mixture volume dealt per stroke

$$= \frac{5.419}{0.9 \times 60} = 0.1 \text{ m}^3 = 100000 \text{ cm}^3$$



The discharge capacity of the wet air pump is also given by $\frac{\pi}{4}\, d^2 l$

$$\therefore \frac{\pi}{4}\, d^2 \times 1.25\, d = 100000\, ; \; d^3 = 101910$$

Hence : diameter of pump,

$$d = 46.53 \text{ cm}$$

Stroke length,

$$l = 46.56 \times 1.25 = 58.16 \text{ cm}$$

## EXAMPLE 20.9

*A surface condenser is designed to handle 2500 kg of steam per hour and the air leakage is estimated as 0.8 per 1000 kg of steam. The steam enters into the condenser dry saturated at 38°C and the condensate is extracted at the lowest point of the condenser at a temperature of 37°C. The system is provided with a separate air extraction pump which draws air over an air cooler, and the air leaves the cooling section at 31°C. Assuming that condenser pressure remains practically constant, compute :*

*(a) mass of steam condensed in the air cooler section*

*(b) volume of air to be handled by the dry air pump*

*(c) reduction in air pump capacity resulting from the cooling of air.*

*The following data is taken from the steam tables :*

| | | | |
|---|---|---|---|
| *Temperature in °C* | 38 | 37 | 31 |
| *Pressure in bar* | 0.0676 | 0.0640 | 0.0459 |
| *Specific volume in m³/kg* | 21.63 | 22.77 | 31.20 |

**Solution :** Air infiltration

$$= \frac{0.8}{1000} \times 2500 = 2 \text{ kg/hr}$$

Total volume of dry steam that enters the condenser at 38°C

$$= 25000 \times 21.63 = 54075 \text{ m}^3\text{/hr}$$

This is also the volume of 2 kg of air and it exerts its own partial pressure.

Partial pressure of air

$$p_a = \frac{m_a R T_a}{V_a} = \frac{2 \times 287 \times (273+38)}{54075}$$
$$= 3.30 \text{ N/m}^2 = 0.33 \times 10^{-4} \text{ bar}$$

**This is negligibly small and can be neglected. Obviously then the total pressure in the condenser is the saturation pressure corresponding to 38°C (the temperature at which steam enters the condenser. Thus**

$$p = 0.0676 \text{ bar}$$

*At condensate extraction* (37°C)

Partial pressure of steam

$$p_s = 0.0640 \text{ bar}$$

Partial pressure of air

$$p_a = p - p_s$$
$$= 0.0676 - 0.0640 = 0.0036 \text{ bar}$$

Applying the characteristic gas equation to 2 kg of air that is to be extracted per hour at partial pressure of 0.0036 bar, we have

Volume of air removed

$$= \frac{2 \times 287 \times (273+37)}{0.0036 \times 10^5}$$
$$= 494.28 \text{ m}^3\text{/hr}$$

By Dalton's law this will also be the volume of steam at 37°C

$\therefore$ Mass of steam going with air

$$= \frac{494.28}{22.77} = 21.71 \text{ kg/hr}$$

*At air cooler section* (31°C)

Partial pressure of steam

$$p_s = 0.0459 \text{ bar}$$

Partial pressure of air

$$p_a = p - p_s$$
$$= 0.0676 - 0.0459 = 0.0217 \text{ bar}$$

Applying characteristic gas equation, Volume of air removed

$$V_a = \frac{2 \times 287 \times (273+31)}{0.0217 \times 10^5}$$
$$= 80.4 \text{ m}^3\text{/hr}$$

Mass of steam associated with this air and which is partially condensed in the air cooler

$$= \frac{80.4}{31.2} = 2.577 \text{ kg/hr}$$

$\therefore$ Mass of steam condensed in air cooler section (or saving in condensate)

$$= 21.71 - 2.577 = 19.133 \text{ kg/hr}$$

Air pump capacity (volume of air handled by dry air pump) = 494.28 m³/hr

Air pump capacity with air cooler
$= 80.4\ m^3/hr$

$\therefore$ Percentage reduction in air pump capacity due to air cooling

$$= \frac{494.28 - 80.4}{494.28}\ 0.837 \text{ or } 83.7\%$$

**EXAMPLE 20.10.**

*A two-pass surfaces condenser handles 20 × 10³ kg per hour of steam which is 0.87 dry. The condenser maintains a vacuum of 70 cm of mercury when the barometer reads 76 cm of mercury. The condensate is removed from the condenser at 35°C and the rise in the temperature of cooling water is limited to 22°C. If pressure head equivalent to 8 m of water is required to force water through the condenser and pipe, calculate :*

*(a) flow area required for a water velocity of 65 m/min, heat transfer rate is*

*15 × 10⁵ kJ/m²/hr, and*

*(b) power required to run the circulating water pump if it is 70 percent efficient.*

**Solution :** Absolute condenser pressure
$= 78 - 70 = 6$ cm of mercury

$$= \frac{1.01325}{76} \times 6 = 0.08 \text{ bar}$$

Corresponding to condenser pressure of 0.08 bar, it is noted from steam tables that

$h_f = 173.38$ kJ/kg

and $h_{fg} = 2403.4$ kJ/kg

Also at condensate temperature of 35°C,

$h_w = 146.66$ kJ/kg

From heat energy balance, heat lost by steam
= heat gained by water

Now : heat lost by steam

$= m_s \left[(h_f + h_{fg}) - h_w\right]$

$= 20 \times 10^3\ [(173.38 + 0.87 \times 2403.34) - 146.66]$

$= 42.35 \times 10^6$ kJ/hr

heat gained by water $= m_w\ c_{pw}\ \Delta T$

$= m_w \times 4.18 \times 22$

$= 91.96\ m_w$ kJ/hr

$\therefore$ $91.96\ m_w = 42.35 \times 10^6$

Solution gives : cooling water flow rate

$m_w = 4.605 \times 10^5$ kg/hour

Volume flow of water

$$= (4.604 \times 10^5) \times \frac{1}{1000}$$

$= 460.5\ m^3/hr$

where density of water has been taken as 1000 kg/m³

(*a*) Flow area required

$$= \frac{\text{volume flow of water}}{\text{flow velocity of water}}$$

$$= \frac{460.5}{60} \times \frac{1}{65} = 0.1181\ m^2$$

(*b*) Cooling surface area required

$$= \frac{\text{heat lost by steam}}{\text{average heat transfer rate}}$$

$$= \frac{42.35 \times 10^6}{15 \times 10^5} = 28.23\ m^2$$

(*c*) Velocity head $= \frac{V^2}{2g}$

$$= \frac{65^2}{2 \times (9.81 \times 3600)} = 0.0598 \text{ m}$$

Total head $h = 8 + 0.0598 = 8.059$ m

Power required to run the circulating water pump

$$= (m_w\ gh) \times \frac{1}{\eta}$$

$$= \frac{4.605 \times 10^5}{3600} \times 9.81 \times 8.059 \times \frac{1}{0.7}$$

$= 14447$ Nm/s $= 14.447$ kW

**EXAMPLE 20.11.**

*A barometric jet condenser handles 4500 kg of steam per hour which is 0.96 dry. The condenser maintains 65 cm mercury vecuum when the barometer indicates 76 cm mercury. The cooling water enters the condenser at 15°C temperature and the mixture of condensate and cooling water leaves at 40°C. Assuming no undercooling, make calculations for*



*(a) the minimum height of the tail pipe above the level of hot well, and*

*(b) the amount of cooling water required.*

**Solution :** Absolute condenser pressure

$= (76 - 65) = 11$ cm of marcury

$$= \frac{11}{100} \times 13.6 = 1.496 \text{ m of water}$$

Barometric height

$$= \frac{76}{100} \times 13.6 = 10.336 \text{ m of water}$$

$\therefore$ Length of tail pipe

$= 10.336 - 1.498 = 8.84$ m

(*b*) As there is no undercooling of condensate, the saturation temperature of steam entering the condenser will be the same as the temperature of the mixture of condensate and cooling water. Corresponding to this 40°C temperature.

$h_w = h_f = 167.54$ kJ/kg

and $h_{fg} = 2406.7$ kJ/kg

From energy balance, heat gained by cooling water = heat lost by steam

$m_w\ c_{pw}\ (t_2 - t_1) = m_s \left[(h_f + x h_{fg}) - h_w\right]$

or $m_w \times 4.18\ (40 - 15)$

$= 4500\ [(167.54 + 0.96 \times 2406.7) - 167.54]$

$\therefore$ Amount of cooling water required

$$m_w = \frac{4500 \times (0.96 \times 2406.7)}{4.18\ (40 - 15)}$$

$= 99492$ kg/hr

**EXAMPLE 20.12.**

*The steam at exit from a steam turbine enters a jet condenser at the rate of 325 kg/minute and the volume of cooling required for condensation of steam is 0.02 m³/kg of steam. The cooling water enters the condenser at 20°C and the condensate (mixture of condensed steam and water) leaves at 30°C. The condenser maintains a vacuum of 680 mm of Hg whilst the borometer reads 760 mm Hg. The air leakage into the condenser is 0.05% of the steam mass that enters the condenser. If the volume of air dissolved in the water injected is 5% of the volume at atmospheric pressure, calculate the volume handling capacity of air pump for removing air and condensate. Take volumetric efficiency of air pump as 88%.*

**Solution :** Absolate condenser pressure

$= (760 - 680) \times 0.001333$

$= 0.1066$ bar

From steam tables : At 30°C partial pressure of steam

$p_s = 0.00426$ bar

$\therefore$ partial pressure of air

$p_a = 0.1066 - 0.0426 = 0.064$ bar

Volume of cooling water required

$= 325 \times 0.02 = 6.5\ m^3/min$

Volume of air entering the condenser with cooling water $= 6.5 \times 0.05 = 0.325\ m^3/min$

This air enters the condenser at atmospheric pressure ($p = 1.0132$ bar) and 20°C temperature.

Then, mass of air entering the condenser with cooling water

$$= \frac{pv}{RT} = \frac{(1.0132 \times 10^5) \times 0.325}{287 \times (273 + 20)}$$

$= 0.3916$ kg/min

Mass of air leakage into the condenser

$$= \frac{0.05}{100} \times 325 = 0.1625 \text{ kg/min}$$

As such, the total mass of air inside the condenser

$= 0.3916 + 0.1625$

$= 0.5541$ kg/min

The pressure and temperature corresponding to this mass of air are 0.064 bar (partial pressure of air) and 30°C (temperature of mixture of steam and water)

$$\text{Volume of air} = \frac{mRT}{p}$$

$$= \frac{0.5541 \times 287 \times (273 + 30)}{0.064 \times 10^5}$$

$= 7.529\ m^3/min$

Volume of steam condensed

$= \text{mass of steam} \times V_f$ at 30°C

$= 325 \times 0.001004 = 0.3263\ m^3/min$

Then, total capacity of wet air pump

= volume of condensate
+ volume of air
+ volume of cooling water

$= 0.3263 + 7.529 + 6.5$

$= 14.325 \text{ m}^3/\text{min}$

$\therefore$ Actual capacity of air pump

$$= \frac{14.355}{0.88} = 16.31 \text{ m}^3/\text{min}$$

## 20.11. EDWARD AIR PUMP

The pressure of air and other non-condensable gases, which enter the condenser along with steam or infiltrate through the joints which are not absolutely tight, adversely affect the performance of the condenser by making the condenser air bound, *i.e.*, by increasing the condenser pressure. An increase in the condenser pressure results into a decrease in pressure drop through the turbine and that adversely affects the turbine performance. Further, the non-condensable gases are highly corrosive and their removal from the condenser is essential.

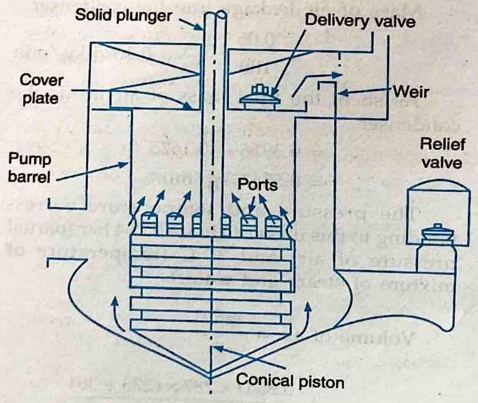


Fig. 20.10. Edrward air pump

Refer Fig. 20.10 for the construction and operation of Edward air. It is a wet air pump that is provided to maintain a desired vacuum in the condenser by extracting a mixture of condensate (in the form of water) and non-condensable gases (particularly air) from the condenser.

*Construction:* The unit consists of a conical piston that reciprocates inside a barrel. At the top of the barrel is a cover plate that carries a number of delivery valves for discharging the charge (moist air and water in fluid state) to the hot well. The bottom of the barrel follows the contour of the conical end of the piston. The upper surface of the piston has circumferential grooves which hold water and form a labyrinth packing. There are ports at the lower end of the barrel and these connect the pump with the condenser through flange.

*Working:* Consider the piston to be at the top and begining to execute the downward stroke. A partial vacuum is created in the barrel above the piston and that closes the delivery valve. When the piston uncovers the ports during its downward movement, the air and water vapour from the condenser rush into the space above the piston. With further downward movement, the condensate entrapped in the conical bottom of the pump is forced above the piston through ports. At this instant, pump is said to be fully charged of with water, air and steam above the piston.

During upward motion of the piston, the charge is compressed to a pressure higher than the atmospheric pressure. The delivery valve opens and the charge flows over the weir to hotwell. The weir maintains a sufficient constant head of water above the cover plate and that seals the delivery valve against air leakage (preventing the air to enter the barrel.)

There is a water sealed relief valve at the barrel of cylinder. If pressure below the piston exceeds the atmospheric pressure, the relief valve opens to atmosphere and that allows the condensate and air to escape through it.

**Note :** (1) This reciprocating pump is quite reliable and satisfactory as long as relatively low vacuum is required as is the case with land and marine condensing steam engines. For large power plants operating with high vacuum and dealing with large quantities of steam, such reciprocating pumps have limitations of size, speed and the resistances affered by passage to flow of air. Further, in central power stations the practice of removing both water and gases together is considered objectionable. The presence of oxygen in feed water is likely to cause damage to high pressure boilers. This has led to the development of steam jet ejector which is a dry air pump, a pump that removes only the air.



## 20.12. COOLING TOWERS AND COOLING PONDS

The cooling towers and cooling ponds are employed for cooling the hot water coming out of the condenser so that the resulting cooled water can be used again for circulation in the condenser. The cooling is affected by exposing the hot water to an atmosphere of air. The cooling tower becomes a necessity when the supply of cooling water is scarce and the same water has to be used over and over again for cooling purposes.

The cooling process in the cooling tower is normally affected by the following factors :

- temperature and humility of air,
- velocity with which the air enters the cooling tower,
- reach of air to different sections of the cooling tower,
- arrangement of plates/baffles inside the tower, and
- size and height of the tower.

The cooling towers operate on draught which may be natural, induced or forced.

*Natural draught cooling tower* : Water from the condenser is supplied to troughs and nozzles with the help of a pump installed at a height of about 8-10 m from the bottom. Water falls in the form of spray and meets the air which enters from the bottom. Water loses its heat to air and gets cooled. The cooled water is collected at the bottom and is returned to condenser for circulation. There are no moving parts in the natural draught cooling tower. Air flow through the shell is created by the density difference between the atmospheric air and the air inside the tower which has been warmed by the hot circulating water.

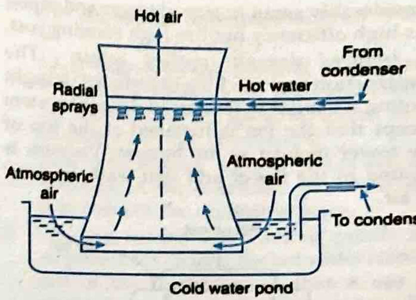


Fig. 20.11. Natural draught cooling tower

Some of the water during its fall is evaporated and is carried along by the heated air . The loss of water due to evaporation is made up by fresh water called *make up water*. The reduction in temperature of water is called *range*.

The performance of natural drought tower is affected by variation of atmospheric conditions and wind velocity.

*Forced draught cooling tower* : Hot water coming from the condenser enters the unit from the top and gets sprayed through nozzles. Water droplets falling downwards meet the air going upward. A good supply of air is made by a fan installed at the bottom of the tower. Spray eliminators are provided at the top and these entrap the water which is likely to go along with hot air leaving from the top.

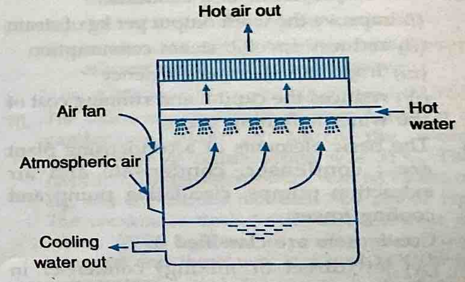


Fig. 20.12. Forced draught cooling tower

Compared to natural draught cooling tower, the forced draught system is considerably small in size of tower and pipes, has high efficiency but has high running cost.

*Induced draught cooling tower* : The construction and working of induced draught cooling is similar to the forced draught system except that the fan is installed at the top of the tower instead at the bottom. Vacuum is created in the tower and that leads to entry of air.

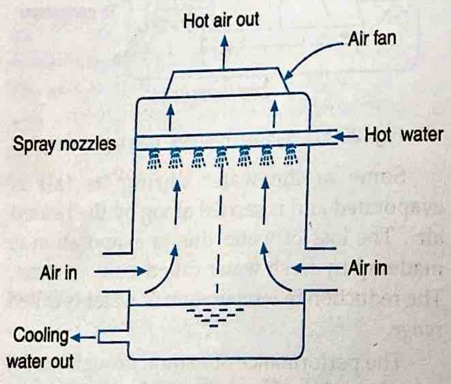


Fig. 20.13. Induced draught cooling tower

Cooling towers are generally of hyperbolic shape and are made of timber, steel or concrete. The timber towers have a short life, a limited cooling capacity, high maintenance cost and rot easily due to exposure to sun, wind, water etc. In contrast, the advantages of concrete cooling towers are: low maintenance, large capacity, increased stability, and improved draught and air circulation.

*Cooling pond* : The hot water coming from the condenser is sprayed into the cooling pond through nozzles. These nozzles are kept at a distance from each other so that there is no interference from between two sprays. Further, for effective cooling, the nozzles are fitted at a height of 1 to 2 m above the ground level. The evaporation from the water surface produces the cooling effect.

For the same duty, the surface area required in a cooling pond is about 30 times the size of cooling tower. Further, a good amount of water is lost by evaporation and by the wind blowing across the cooling pond.

## SALIENT POINTS

1. A steam condenser is a closed vessel into which the exhaust steam from an engine or a turbine is exhausted and condensed by means of cooling water at a pressure below atmospheric.

   The condensed steam is called condensate.
2. The employment of a condenser
   (*i*) improve the work output per kg of steam
   (*ii*) reduces specific steam consumption
   (*iii*) improves thermal efficiency
   (*iv*) reduces the capital and running cost of the water softening plant
3. The basic elements of a condensing plant are : condenser, condensate and air extraction pumps, circulating pump and cooling tower.
4. Condensers are classified as :

   (A) jet (direct or mixing) condenser in which there is a direct contact between the exhaust steam and the cooling water. The jet condensers are further classified as counter or parallel flow, high or low level and ejector condensers.

   (*B*) surface condensers in which water flows through a network of tubes over which passes the exhaust steam. The surface condensers are further classified according to direction of flow of condensate (central flow, down flow), number of passes of water (single pass or multi pass) and shape of shell (circular, oval or V-shaped).
5. The vacuum gange reading corrected to standard atmosphere is

$$= 760 - (\text{barometer reading} - \text{vacuum gauge reading})$$

The vacuum efficiency is prescribed as

$$\eta_v = \frac{\text{actual vacuum}}{\text{ideal vaccum}}$$

$$= \frac{\text{barometric pressure} - \text{actual pressure}}{\text{barometric pressure} - \text{ideal pressure}}$$

The condenser efficiency is

$$\eta_{con} = \frac{t_2 - t_1}{t_3 - t_1}$$

where $t_3$ is saturation temperature corresponding to condenser pressure, $t_2$ and $t_1$ are the outlet and inlet temperature of cooling water.

6. In a condenser, there is a mixture of steam and air leaking into the condenser, and according to Dalton's law :

$$p = p_s + p_a$$

where $p$ is the total pressure of the mixture $p_s$ and $p_a$ are the partial pressures of steam and air respectively.

7. The air leakage into the condenser is due to :

   (*i*) leakage through packing glands and microscopic holes in the shell and joints.
   (*ii*) dissolved air in spent up steam or in cooling water (jet condensers)

   Due to air leakage, there is loss of vaccum (increase in the condenser pressure) and that reduces the expansion work.
8. Cooling towers and cooling ponds are employed for cooling the hot water coming out of the condenser so that it can be recirculated for cooling purposes. The cooling towers may be of natural draught, induced draught or forced draught type.



## REVIEW QUESTIONS

**A. Conceptual and conventional questions.**

1. Enumerate the advantages of incorporating a condenser in a steam power plant.
2. Make a neat sketch of barometric jet condenser and explain its working.
3. (*a*) State the functions of condenser in a steam power plant.
   (*b*) Describe, with a neat sketch, the construction and working of a surface condenser.
4. State the relative advantages and disadvantages of surface condenser and a jet condenser.
5. (*a*) State Dalton's law of partial pressure. How does it apply to the condenser of a steam plant.
   (*b*) Define and explain the terms vacuum efficiency and condenser efficiency of a condensing plant.
6. (*a*) State briefly the sources and effects of air leakage into a condenser.
   (*b*) How do you proceed to check the leakage of air into a condenser ?
7. (*a*) What purpose is served by a cooling tower in a condensing plant ?
   (*b*) Name the different types of cooling towers and give a brief description of each one of them.
8. The vacuum in a surface condenser is noted to be 711 mm of mercury and the barometer stands at 760 mm of mercury. The hot well temperature is 32°C, and the inlet and outlet temperatures of cooling water are 12°C and 28°C respectively. Calculate the condenser efficiency and the vacuum efficiency.

   [Ans. 80% ; 98.15%]
9. The following observations pertain to a condenser test :

   Vacuum = 700 mm of Hg and barometer reading = 760 mm of Hg

   Inlet and outlet temperature of cooling water = 16.5° C and 31°C.

   Mean temperature of condensation = 29°C

   Mass of condensate = 1200 kg/hr, and

   Mass of cooling water = 45500 kg/hr

   Determine :

   (*a*) Mass of air present per unit condenser volume
   (*b*) state of steam at entry to condenser
   (*c*) condensate under cooling, condenser efficiency and vacuum efficiency.

   [Ans. 0.026 kg ; 094 ; 6°C, 78.5% and 97.5%]
10. The following observations were made on a condensing plant in which the mean temperature of condensation was 35°C. The recorded condenser vacuum was 68.73 cm of Hg and the barometer read 74.0 cm of Hg. The condenser dealt with 1170 kg/hour of steam. 45000 kg/hour of cooling water was supplied and there was a rise of 14°C in its temperature.

    Determine : (*a*) corrected vacuum corresponding to barometric reading of 76cm of Hg
    (*b*) vacuum efficiency (*c*) dryness fraction of

steam as it enters the condenser (*d*) mass of air present per cubic meter of condenser volume.

[Ans. 70.73cm of Hg, 98.5%, 0.923, 0.01505 kg/m$^3$]

11. The vacuum in a surface condenser is 70.5 cm of mercury when the barometer reads 75 cm. At inlet to air cooling section, the temperature is 33°C and at the outlet 29°C. If air leakage is estimated to be 2.3 kg/hr, make calculations for the mass of steam condensed in the cooling section.

[Ans. 4.6 kg/hr]

12. The following observations refer to a test on a steam condensate plant :

Recorded condenser vacuum = 710 mm of mercury

Barometer reading = 760 mm of mercury

Mean condenser temperature = 29°C

Condensate collected = 22700 kg/hr

The air infiltration into the condenser is prescribed by

$$\left(\frac{\text{steam condensed in kg/hr}}{2000}+2\right)$$

Compute the discharging capacity of wet air pump which removes both air and condensate. Take volumetric efficiency of air equal to 80 percent. If the air pump is single acting and runs at 60 rev/min and the stroke equals 1.5 times the piston diameter, compute the dimensions of the pump.

[Ans. 574 m$^3$/hr, 51.4 cm and 77.1 cm]

**B. Fill in the blanks with appropriate word/words.**

1. Condenser is a .................. vessel where condensation of steam takes place at constant pressure which is generally .................. than atmospheric pressure.
2. .................. pump is used to extract out the air and dissolved gases from the condenser.
3. High level jet condenser is also called as .................. condenser.
4. In a high level jet condenser, the tail pipe that discharges condensate from the condenser into the hot well is about .................. long.
5. The absolute pressure inside a condenser is the sum of partial pressures of steam and air inside it. This statement is referred to as ..................
6. In an .................. condenser, the mixture of condensate and water is discharged automatically in the hotwell and there is no need of extraction pump.
7. In an ideal condenser, the temperature of condensate equals the .................. corresponding to condenser pressure.
8. The vacuum obtainable in a condenser is dependent on the .................. of circulating water.
9. The thermal efficiency of a steam power plant is increased with the employment of surface condenser because the .................. of heat refection is reduced.

**Answers**

1. closed, less 2. air extraction 3. barometric 4. 10.36 m 5. Dalton's law of partial pressure 6. ejector 7. saturation temperature 8. temperature 9. average temperature

**C. Multiple choice questions.**

1. Which aspect may or may not be true regarding the advantage gained by using a condenser in a steam power plant ?
   (*a*) expansion of steam to a lower back pressure
   (*b*) increase in available enthalpy drop and consequently more work output
   (*c*) reduction in steam consumption per kW-hour
   (*d*) fuel economy and safety from thermal stresses in the boiler shell
2. Which one of the followings is a wrong statement?
   (*a*) the employment of condenser permits the expansion of steam to a pressure lower than atmospheric pressure
   (*b*) the thermodynamic efficiency of a condensing steam turbine unit is higher than that of a non-condensing one
   (*c*) the vacuum in a condenser is a function of both absolute pressure and the barometric pressure
   (*d*) the ratio of actual vacuum in the condenser to ideal vacuum is called condenser efficiency
3. The inlet and outlet temperatures of cooling water being supplied to a condenser are $t_1$ and $t_2$ respectively. If $t_s$ is the saturation



temperature corresponding to condenser pressure, the condenser efficiency will be
   (*a*) $\frac{t_2-t_1}{t_s-t_2}$ (*b*) $\frac{t_2-t_1}{t_s-t_1}$
   (*c*) $\frac{t_2-t_1}{t_s}$ (*d*) $\frac{t_1+t_2}{2t_s}$
4. In a steam condenser, the barometric reading and the recorded condenser vacuum are stated to be 76.5 cm and 71 cm of mercury. The vacuum corrected to standard barometer of 76 cm of mercury would be
   (*a*) 70.5 cm of Hg (*b*) 71.5 cm of Hg
   (*c*) 69.5 cm of Hg (*d*) 72.5 cm of Hg
5. The length of tail pipe fitted to the barometric or high level jet condenser is about
   (*a*) 6.2 m (*b*) 7.6 m
   (*c*) 8.5 m (*d*) 10.4 m
6. Which aspect is not true in the context of a jet condenser ?
   (*a*) more intimate mixing of steam and cooling water
   (*b*) air extraction pump requires high power
   (*c*) requires considerable space for erection
   (*d*) equipment simple and low in cost
7. The maximum limit of vacuum attained in a jet condenser is about
   (*a*) 55 cm of Hg (*b*) 60 cm of Hg
   (*c*) 65 cm of Hg (*d*) 70 cm of Hg
8. All of the following statements about surface condenser are correct, except
   (*a*) high vacuum can be attained and greater plant efficiency is achieved
   (*b*) condensate can be salvaged and used for boiler feed
   (*c*) high capital and maintenance cost
   (*d*) maximum chances of losing vacuum
   (*e*) suitable for high capacity units
9. In a surface condenser used in steam power station, under cooling of condensate is undersirable as this would
   (*a*) not absorb the gases in steam
   (*b*) reduce efficiency of the plant
   (*c*) increase the cooling water requirements
   (*d*) increase the thermal stresses in the condenser.
10. With the removal of air from a surface condenser, there is
   (*a*) no change in absolute pressure
   (*b*) rise in absolute pressure
   (*c*) fall in absolute pressure
   (*d*) rise in temperature of condensed steam
11. Edwards air pump removes .................. from the condenser
   (*a*) only air
   (*b*) only un-condensed vapour
   (*c*) moist air (air plus vapour)
   (*d*) moist air and also the condensate
12. Consider the following statements
   1. In a shell and tube condenser surface condenser, cooling water passes through the tubes and steam surrounds them
   2. Evaporative type of condenser has steam in pipes surrounded by water
   3. The cooling section in a surface condenser allows greater quantity of water to be extracted along with air

   Which of the statements made above are correct?
   (*a*) 1 and 2 (*b*) 2 and 3
   (*c*) 1 and 3 (*d*) 1, 2 and 3

**Answers**

| | | | | |
|---|---|---|---|---|
| 1. (*d*) | 2. (*d*) | 3. (*b*) | 4. (*a*) | 5. (*d*) |
| 6. (*c*) | 7. (*c*) | 8. (*d*) | 9. (*c*) | 10. (*c*) |
| 11. (*d*) | 12. (*a*) | | | |

□□□