17

Simple Steam Engines

1. Introduction. 2. Classification of Steam Engines. 3. Important Parts of a Steam Engine. 4. Working of a Single Cylinder Double Acting Horizontal Reciprocating Steam Engine. 5. Important Terms used in Steam Engines. 6. Indicator Diagram of a Simple Steam Engine. 7. Theoretical or Hypothetical Indicator Diagram. 8. Theoretical or Hypothetical Mean Effective Pressure. 9. Actual Indicator Diagram. 10. Diagram Factor. 11. Power Developed by a Simple Steam Engine. 12. Indicated Power. 13. Effect of Piston Rod in Double Acting Steam Engine. 14. Brake Power. 15. Measurement of Brake Power. 16. Prony Brake Dynamometer. 17. Rope Brake Dynamometer.

17.1. Introduction

In all steam engines, the steam is used as the working substance. These engines operate on the principle of first law of thermodynamics, *i.e.* heat and work are mutually convertible. In a reciprocating steam engine, as the heat energy in the steam is converted into mechanical work by the reciprocating (to and fro) motion of the piston, it is also called *reciprocating steam engine*. Moreover, as the combustion of the fuel takes place outside the engine cylinder, it is also called an *external combustion engine*.

17.2. Classification of Steam Engines

The steam engines have been classified by various scientists on different basis. But the following classifications are important from the subject point of view.

1. According to number of working strokes

(*a*) Single acting steam engine, and (*b*) Double acting steam engine.

When steam is admitted on one side of the piston, and one working stroke is produced during each revolution of the crankshaft, it is said to be a *single acting steam engine*. But when the steam is admitted, in turn, on both sides of the piston and two working strokes are produced during each revolution of the crankshaft, it is said to be a *double acting steam engine*. A double acting steam engine produces double the power than that produced by a single acting steam engine.

2. According to the position of the cylinder

(*a*) Horizontal steam engine, and (*b*) Vertical steam engine.

When the axis of the cylinder is horizontal, it is said to be a *horizontal steam engine*. But when the axis of the cylinder is vertical, it is called a *vertical steam engine*. A vertical steam engine requires less floor area than the horizontal steam engine.

3. According to the speed of the crankshaft

(*a*) Slow speed steam engine, (*b*) Medium speed steam engine, and (*c*) High speed steam engine.

When the speed of the crankshaft is less than 100 revolutions per minute (r.p.m.), it is called a *slow speed steam engine*. But when the speed of the crankshaft is between 100 r.p.m. and 250 r.p.m., it is called a *medium speed steam engine*. Similarly, when the speed of the crankshaft is above 250 r.p.m., it is known as a *high speed steam engine.*

4. *According to the type of exhaust*

(*a*) Condensing steam engine, and (*b*) Non-condensing steam engine.

When steam after doing work in the cylinder passes into a condenser, which condenses the steam into water at a pressure less than the atmospheric pressure, it is said to be a *condensing steam engine*. But when the steam after doing work in the cylinder is exhausted into the atmosphere, it is said to be a *non-condensing steam engine*. The steam pressure in the cylinder is, therefore, not allowed to fall below the atmospheric pressure.

5. *According to the expansion of the steam in the engine cylinder*

(*a*) Simple steam engine, and (*b*) Compound steam engine.

When the expansion of the steam is carried out in a single cylinder and then exhausted into the atmosphere or a condenser, it is said to be a *simple steam engine*. But when the expansion of the steam is completed in two or more cylinders, the engine is called a *compound steam engine*. The compound steam engines are generally condensing engines. But some of them may be non-condensing also.

6. *According to the method of governing employed*

(*a*) Throttling steam engine, and (*b*) Automatic cut-off steam engine.

When the engine speed is controlled by means of a throttle valve in the steam pipe, which regulates the pressure of steam to the engine, it is called a *throttling steam engine*. But when the speed is controlled by controlling the steam pressure with an automatic cut-off governor, it is called an *automatic cut-off steam engine*.

17.3. Important Parts of a Steam Engine

All the parts of a steam engine may be broadly divided into two groups *i.e.* stationary parts and moving parts. Though a steam engine consists of innumerable parts, both stationary and moving, yet the following are important from the subject point of view :

1. *Frame*. It is a heavy cast iron part, which supports all the stationary as well as moving parts and holds them in proper position. It generally, rests on engine foundations.

2. *Cylinder*. It is also a cast iron cylindrical hollow vessel, in which the piston moves to and fro under the steam pressure. Both ends of the cylinder are closed and made steam tight. In small steam engines, the cylinder is made an integral part of the frame.

3. *Steam chest*. It is casted as an integral part of the cylinder. It supplies steam to the cylinder with the movement of D-slide valve.

4. *D-slide valve*. It moves in the steam chest with simple harmonic motion. Its function is to exhaust steam from the cylinder at proper movement.

5. *Inlet and exhaust ports*. These are holes provided in the body of the cylinder for the movement of steam. The steam is admitted from the steam chest alternately to either sides of the cylinder through the inlet ports. The steam, after doing its work in the cylinder, is exhausted through the exhaust port.

6. *Piston*. It is a cylindrical disc, moving to and fro, in the cylinder because of the steam pressure. Its function is to convert heat energy of the steam into mechanical work. Piston rings, made from cast iron, are fitted in the grooves in the piston. Their purpose is to prevent the leakage of steam.

7. *Piston rod*. It is a circular rod, which is connected to the piston on one side and cross head to the other. Its main function is to transfer motion from the piston to the cross-head.

8. *Cross-head.* It is a link between the piston rod and connecting rod. Its function is to guide motion of the piston rod and to prevent it from bending.

9. *Connecting rod.* It is made of forged steel, whose one end is connected to the cross head and the other to the crank. Its function is to convert reciprocating motion of the piston (or cross head) into rotary motion of the crank.

10. *Crank shaft.* It is the main shaft of the engine having a crank. The crank works on the lever principle and produces rotary motion of the shaft. The crank shaft is supported on main bearing of the engine.

11. *Eccentric.* It is generally made of cast iron, and is fitted to the crank shaft. Its function is to provide reciprocating motion to the slide valve.

12. *Eccentric rod and valve rod.* The eccentric rod is made of forged steel, whose one end is fixed to the eccentric and other to the valve rod. Its function is to convert rotary motion of the crankshaft into to and fro motion of the valve rod. The valve rod connects the eccentric and the D-slide valve. Its function is to provide simple harmonic motion to the D-slide valve.

13. *Flywheel.* It is a heavy cast iron wheel, mounted on the crank shaft. Its function is to prevent the fluctuation of engine. It also prevents the jerks to the crankshaft.

14. *Governor.* It is a device to keep the engine speed, more or less, uniform at all load conditions. It is done either by controlling the quantity or pressure of the steam supplied to the engine.

17.4. Working of a Single Cylinder Double Acting Horizontal Reciprocating Steam Engine

The principal parts of a single cylinder, double acting horizontal reciprocating steam engine are shown in Fig. 17.1.

The superheated steam at a high pressure (about 20 atmospheres) from the boiler is led into the steam chest. After that the steam makes its way into the cylinder through any of the ports '*a*' or '*b*' depending upon the position of the D-slide valve. When port '*a*' is open, the steam rushes to the left side of the piston and forces it to the right. At this stage, the slide valve covers the exhaust port and the other steam port '*b*' as shown in Fig. 17.1. Since the pressure of steam is greater on the left side than that on right side, the piston moves to the right.

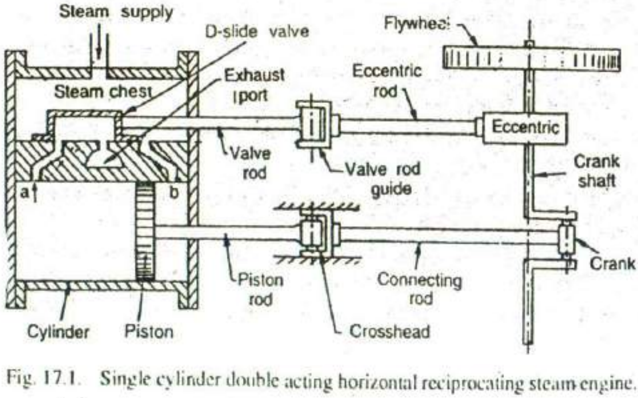


Fig. 17.1. Single cylinder double acting horizontal reciprocating steam engine.

When the piston reaches near the end of the cylinder, it closes the steam port '*a*' and exhaust port. The steam port '*b*' is now open, and the steam rushes to the right side of the piston. This forces the piston to the left and at the same time the exhaust steam goes out through the exhaust pipe, and thus completes the cycle of operation. The same process is repeated in other cycles of operation, and as such the engine works.

Note : At the end of each stroke, the piston changes its direction of motion and is momentarily stopped. The crank comes in line with the piston rod. The extreme left and right positions of the crank, where the piston rod exerts no turning tendency on the main shaft, are called dead centres of the crank.

17.5. Important Terms used in Steam Engines

The *theoretical indicator diagram for a simple steam engine is shown in Fig. 17.2. The following are some important terms used in steam engines.

1. *Bore.* The internal diameter of the cylinder of the engine is known as *bore.*

2. *Dead centres.* The extreme positions of the piston inside the cylinder during its motion are known as dead centres. There are two dead centres, *i.e.*

**(*a*) Inner dead centre (I.D.C.), and (*b*) Outer dead centre (O.D.C.).

In a horizontal engine, the inner most position of the piston (towards the cylinder cover end) is known as *inner dead centre*, whereas the outer most position of the piston towards the crank end is called *outer dead centre*, as shown in Fig. 17.2.

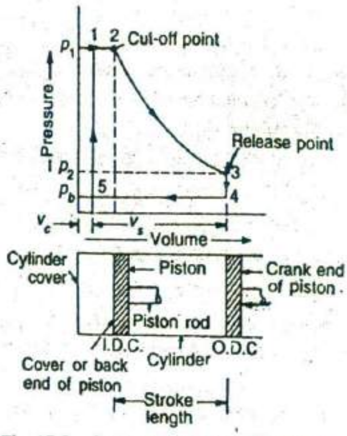


Fig. 17.2. Important terms used in steam engines.

3. *Clearance volume.* The volume of space between the cylinder cover and the piston, when the piston is at I.D.C. position is called clearance volume (v_c). It is usually represented as a percentage of stroke volume.

4. *Stroke volume or swept volume.* The volume swept by the piston when it moves from I.D.C. to O.D.C., is known as stroke volume or swept volume (v_s). It is also known as *piston displacement*. Mathematically, stroke volume or swept volume,

$$v_s = \frac{\pi}{4} \times D^2 \times L$$

where D = Bore or internal diameter of the cylinder, and

L = Length of the stroke.

5. *Cut-off volume.* Theoretically, the steam from the boiler enters the clearance space and pushes the piston outward doing external work. At some point during outward movement of the piston, the supply of steam is stopped. The point or the volume where the cut-off of steam takes place is called the point of cut-off or cut-off volume.

6. *Average piston speed.* The distance travelled by the piston per unit time is known as average piston speed. Mathematically,

Average piston speed $= LN$ m/min, for single acting steam engine

$= 2\,LN$ m/min, for double acting steam engine

where L = Length of the stroke in metres, and

N = Speed in R.P.M.

* For further details, see Art. 17.7.

** In a vertical engine, these centres are known as bottom dead centre (B.D.C.) and top dead centre (T.D.C.).

7. *Mean effective pressure.* The average pressure on the piston during the working stroke is called mean effective pressure. It is given by the mean depth of the p-v diagram. Mathematically, mean effective pressure,

$$p_m = \frac{\text{Workdone per cycle}}{\text{Stroke volume}}$$

17.6. Indicator Diagram of a Simple Steam Engine

It is a graphical representation of the variation in pressure and volume of steam inside the cylinder or p-v diagram. As a matter of fact, the theoretical or hypothetical indicator diagram of a simple steam engine has been developed from that of a modified Rankine cycle. It is based on the following assumptions :

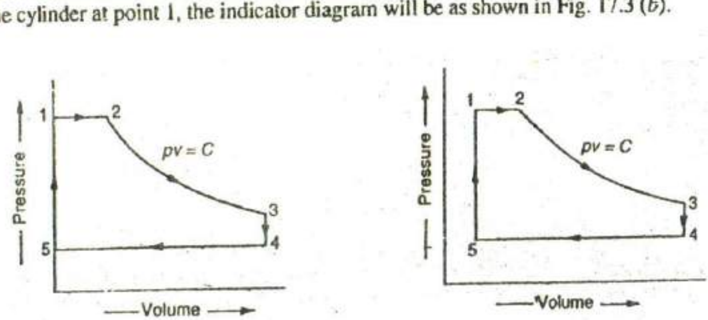
1. The opening and closing of steam ports is instantaneous.
2. There is no pressure drop due to condensation.
3. There is no wire drawing due to restricted valve opening.
4. The steam is admitted at boiler pressure and exhausted at condenser pressure.
5. The expansion (or compression) of the steam is hyperbolic (*i.e.* $p\,v = C$)

It may be noted that the above assumptions are not correct from the practical point of view. As a result of this, it has lead to the change in the indicator diagram from the basic modified Rankine cycle.

Note : In the succeeding articles, we shall discuss work done and mean effective pressure (also known as theoretical work done and theoretical mean effective pressure) from the theoretical indicator diagrams.

17.7. Theoretical or Hypothetical Indicator Diagram

The theoretical or hypothetical indicator diagram without clearance and with clearance is shown in Fig. 17.3. In other words, if there is no steam in the cylinder (or there is zero volume of steam at point 1), the indicator diagram will be as shown in Fig. 17.3 (*a*). Similarly, if there is some steam in the cylinder at point 1, the indicator diagram will be as shown in Fig. 17.3 (*b*).



(*a*) Without clearance. (*b*) With clearance.

Fig. 17.3. Theoretical or hypothetical indicator diagram.

The sequence of processes is given below :

1. *Process 1-2.* At point 1, the steam is admitted into the cylinder through the inlet port. As the piston moves towards right, therefore the steam is admitted at constant pressure. Since the supply of steam is cut off at point 2, therefore this point is known as *cut-off point.*

2. *Process 2-3.* At point 2, expansion of steam, in the cylinder, starts with movement of the piston till it reaches the dead end. This expansion takes place hyperbolically (*i.e.* $pv = C$) and pressure falls considerably as shown in Fig. 17.3.

3. *Process 3-4.* At point 3, the exhaust port opens and steam is released from the cylinder to

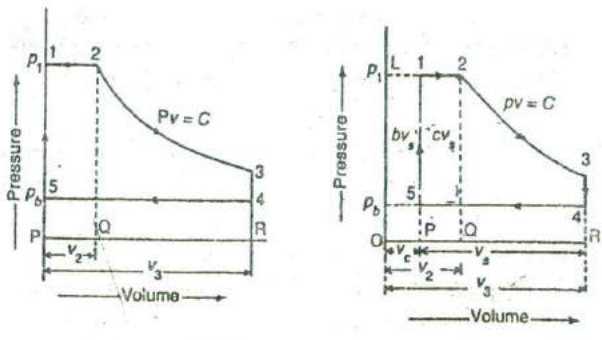
the exhaust. As a result of steam exhaust, pressure in the cylinder falls suddenly (without change in volume) as shown in Fig. 17.3. The point 3 is known as *release point.*

4. *Process 4-5.* At point 4, return journey of the piston starts. Now the used steam is exhausted at constant pressure, till the exhaust port is closed, and the inlet port is open. The steam pressure at point 4 is called *back pressure.*

5. *Process 5-1.* At point 5, the inlet port is opened and some steam suddenly enters into the cylinder, which increases the pressure of steam (without change in volume). This process continues till the original position is restored.

17.8. Theoretical or Hypothetical Mean Effective Pressure

The theoretical or hypothetical mean effective pressure may be determined as discussed below, by considering the theoretical indicator diagram without clearance and with clearance as shown in Fig. 17.4 (*a*) and (*b*) respectively.



(*a*) Without clearance. (*b*) With clearance.

Fig. 17.4. Theoretical or hypothetical indicator diagram

1. *Considering the theoretical or hypothetical indicator diagram without clearance*

The theoretical or hypothetical indicator diagram without clearance is shown in Fig. 17.4 (*a*).

Let p_1 = Initial or admission pressure of steam (*i.e.* pressure at point 1) or boiler pressure,

p_b = Back pressure (*i.e.* pressure at point 4 or 5) or condenser pressure,

v_2 = Volume of steam in the cylinder at the point of cut-off (*i.e.* volume at point 2), and

v_3 = Stroke volume or swept volume or piston displacement volume. It is the volume of steam in the cylinder at the end of stroke (*i.e.* volume at point 3 or 4).

We know that theoretical or hypothetical workdone per cycle

$$= \text{Area of figure } 123451$$

$$= \text{Area } 12\,QP + \text{Area } 23\,RQ - \text{Area } 45PR$$

$$= p_1 v_2 + 2.3\, p_1 v_2 \log (v_3 / v_2) - p_b v_3$$

and theoretical or hypothetical mean effective pressure,

$$p_m = \frac{\text{Workdone per cycle}}{\text{Stroke volume}} = \frac{p_1 v_2 + 2.3\, p_1 v_2 \log (v_3 / v_2) - p_b v_3}{v_3}$$

$$= p_1 \times \frac{v_2}{v_3} + 2.3\, p_1 \times \frac{v_2}{v_3} \log (v_3 / v_2) - p_b = \frac{p_1}{r} (1 + 2.3 \log r) - p_b$$

where $r = v_3 / v_2$ = Expansion ratio

$$= \frac{\text{Volume of steam in the cylinder at the end of stroke}}{\text{Volume of steam at the point of cut-off}}$$

Notes : 1. The volume of steam in the cylinder at the end of stroke, neglecting clearance, is equal to stroke volur.

2. The ratio v_2 / v_3 (*i.e.* reciprocal of expansion ratio)is termed as cut-off ratio. It is defined as the ratio of volume between the points of admission and cut-off of steam and the stroke volume.

3. The steam consumption in kg per cycle may be obtained as follows :

Steam consumption per cycle

$$= \frac{\text{Volume of steam supplied per cycle in m}^3}{\text{Specific volume of dry steam at admission pressure } (p_1) \text{ in m}^3/\text{kg, from steam tables}}$$

$$= \frac{\text{Volume of steam in the cylinder at the point of cut-off } (v_2)}{v_g}.$$

If the steam is initially wet having dryness fraction x, then steam consumption per cycle

$$= \frac{v_2}{x\, v_g} \text{ kg}$$

2. *Considering the theoretical or hypothetical indicator diagram with clearance*

The theoretical or hypothetical indicator diagram with clearance is shown in Fig. 17.4 (*b*).

Let p_1 = Initial or admission pressure of steam (*i.e.* pressure at point 1) or boiler pressure,

p_b = Back pressure (*i.e.* pressure at point 4 or 5) or condenser pressure,

v_c = Clearance volume of the cylinder,

v_2 = Volume of steam at the point of cut-off (*i.e.* volume at point 2)

v_s = Stroke volume or swept volume or piston displacement volume.

v_3 = Total volume of steam in the cylinder = $v_c + v_s$,

b = Ratio of clearance volume to stroke volume = v_c / v_s,

c = Ratio of volume between the points of admission and cut-off of steam to the stroke volume = $(v_2 - v_c) / v_s$

We know that theoretical or hypothetical workdone per cycle.

$$= \text{Area of figure } 123451 = \text{Area } 12\,QP + \text{Area } 23\,RQ - \text{Area } 45\,PR$$

$$= p_1 (v_2 - v_c) + 2.3\, p_1\, v_2 \log (v_3 / v_2) - p_b\, v_s \qquad \ldots (i)$$

and theoretical or hypothetical mean effective pressure,

$$p_m = \frac{\text{Workdone per cycle}}{\text{Stroke volume}} = \frac{p_1 (v_2 - v_c) + 2.3\, p_1\, v_2 \log (v_3 / v_2) - p_b\, v_s}{v_s}$$

$$= p_1 \left(\frac{v_2 - v_c}{v_s} \right) + 2.3 \frac{p_1\, v_2}{v_s} \log \left(\frac{v_3}{v_2} \right) - p_b$$

$$= p_1\, c + 2.3 p_1 \left(\frac{b\,v_s + c\,v_s}{v_s}\right) \log\left(\frac{v_c + v_s}{b\,v_s + c\,v_s}\right) - p_b$$

$$= p_1\, c + 2.3\, p_1\, (b + c) \log\left(\frac{b+1}{b+c}\right) - p_b \qquad \ldots (ii)$$

Notes : 1. We know that cut-off ratio

$$= \frac{\text{Volume of steam in the cylinder at the point of cut-off}}{\text{Volume of steam in the cylinder at the end of stroke}}$$

$$= \frac{v_2}{v_3} = \frac{b\,v_s + c\,v_s}{v_c + v_s} = \frac{b+c}{b+1}$$

and expansion ratio, $r = \dfrac{v_3}{v_2} = \dfrac{b+1}{b+c}$

2. When clearance is neglected, then $b = 0$ or $r = 1/c$. Now from equation (*ii*),

$$p_m = \frac{p_1}{r}(1 + 2.3 \log r) - p_b \qquad \ldots \text{(same as before)}$$

Example 17.1. *A steam engine cylinder receives steam at a pressure of 11.5 bar and cut-off takes place at half of the stroke. Find the theoretical mean effective pressure, if the back pressure of the steam is 0.15 bar. Neglect clearance.*

Solution. Given : $p_1 = 11.5$ bar ; $v_2 = 0.5\, v_3$; $p_b = 0.15$ bar

We know that expansion ratio,

$$r = v_3 / v_2 = v_3 / 0.5\, v_3 = 2$$

We know that theoretical mean effective pressure,

$$p_m = \frac{p_1}{r}(1 + 2.3 \log r) - p_b$$

$$= \frac{11.5}{2}(1 + 2.3 \log 2) - 0.15 = 9.58 \text{ bar} \quad \textbf{Ans.}$$

Example 17.2. *The cylinder of a non-condensing steam engine is supplied with steam at a pressure of 12 bar. The clearance volume is 1/10 of the stroke volume and the cut-off takes place at 0.25 of the stroke. The back pressure is 1.1 bar. Find the mean effective pressure of the steam on the piston. Assume hyperbolic expansion.*

Solution. Given : $p_1 = 12$ bar ; $b = v_c / v_s = 1/10 = 0.1$; $c = (v_2 - v_c)/v_s = 0.25$; $p_b = 1.1$ bar

We know that mean effective pressure,

$$p_m = p_1\, c + 2.3\, p_1\, (b + c) \log\left(\frac{b+1}{b+c}\right) - p_b$$

$$= 12 \times 0.25 + 2.3 \times 12\, (0.1 + 0.25) \log\left(\frac{0.1 + 1}{0.1 + 0.25}\right) - 1.1$$

$$= 3 + 9.66 \log 3.143 - 1.1 = 6.7 \text{ bar} \quad \textbf{Ans.}$$

17.9. Actual Indicator Diagram

The comparison of actual indicator diagram (drawn with firm line) and theoretical indicator diagram (drawn with dotted line) is shown in Fig. 17.5. The following points, regarding actual indicator diagram, are important from the subject point of view :

1. The pressure of steam in the engine cylinder at the beginning of the stroke is less than the boiler pressure. This happens because of the fact that a certain pressure drop is necessary to produce a flow of steam from boiler to the engine cylinder.

2. During the forward stroke of the piston, there is always a slight fall in pressure (shown by line AB) due to wire drawing through the steam ports.

3. As the inlet port can not close instantaneously, the point of cut-off will not be sharp as 2, but rounded off as at B. The rounding of the cut-off point depends upon the type of valve and valve-mechanism employed.

4. The exhaust port opens before the end of the forward stroke (as shown by point C) due to wire drawing through exhaust ports. This causes the rounding off the toe of the diagram.

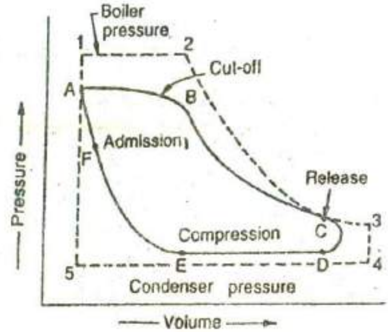


Fig. 17.5. Actual indicator diagram.

5. During the exhaust stroke, the pressure in the cylinder is higher than that of condenser pressure in case of condensing steam engines and higher than atmospheric pressure in case of non-condensing steam engines.

6. The exhaust valve closes at some point E, and the remaining steam in the cylinder is compressed along the curve EF before the end of the exhaust stroke. This reduces the wire drawing when the inlet valve opens at F and also reduces initial condensation. This also serves the purpose of cushioning, which gradually brings the piston, to rest, and thus prevents the shock on the connecting rod bearings, which would otherwise be produced.

7. Due to wire drawing effects, the steam is admitted just before the end of exhaust stroke at F. The pressure produced by compression upto this point is raised to admission pressure by the time piston has reached at the end of exhaust stroke.

Notes : 1. The effect of wire drawing is to decrease the area of indicator diagram. In other words, work done by the engine is reduced. This is, however, compensated by the fact that the wire drawing or throttling dries the steam slightly.

2. The effect of clearance, at the first sight, appears to increase the steam consumption from E to F. But the clearance increases the mean effective pressure and thus increases the work done. However, the net effect of the clearance is to decrease the efficiency. The increase in steam consumption may be reduced by making the point of compression earlier, and thus increasing the pressure obtained at F when the fresh steam is admitted to the cylinder. Earlier compression, however, decreases the area of indicator diagram, *i.e.* work done is reduced.

17.10. Diagram Factor

The diagram factor (usually denoted by K) is the ratio of the area of actual indicator diagram to the area of theoretical or hypothetical indicator diagram. Mathematically, diagram factor,

$$K = \frac{\text{Area of actual indicator diagram}}{\text{Area of theoretical indicator diagram}}$$

We know that the area of the indicator diagram represents the work done per stroke. Therefore, diagram factor may also be expressed mathematically,

$$K = \frac{\text{Actual work done per stroke}}{\text{Theoretical work done per stroke}}$$

We also know that work done per stroke

$$= \text{Mean effective pressure} \times \text{Swept volume}$$

$\therefore$ Actual work done per stroke

$$= \text{Actual m.e.p} \times \text{Swept volume}$$

and theoretical work done per stroke

$$= \text{Theoretical m.e.p} \times \text{Swept volume}$$

$$\therefore \qquad K = \frac{\text{Actual mean effective pressure}\,(p_a)}{\text{Theoretical mean effective pressure}\,(p_m)}$$

The diagram factor may, therefore, be defined as the ratio of actual mean effective pressure to the theoretical mean effective pressure.

Notes : 1. The value of the diagram factor, to be used in any particular case, depends upon a number of factors such as initial condition of steam, initial pressure of steam, back pressure, speed of the engine, type of the engine, type of the valves, etc.

2. An average value of K lies between 0.65 and 0.9.

3. Actual mean effective pressure,

$$p_a = \text{Theoretical m.e.p} \times \text{Diagram factor} = p_m \times K$$

Example 17.3. *The steam is supplied at a pressure of 8.4 bar and cut-off occurs at 0.35 of the stroke. The back pressure is 1.25 bar. If the diagram factor is 0.75, determine the actual mean effective pressure. Neglect clearance.*

Solution. Given : $p_1 = 8.4$ bar ; $v_2 = 0.35\, v_3$; $p_b = 1.25$ bar ; $K = 0.75$

We know that expansion ratio,

$$r = v_3 / v_2 = v_3 / 0.35\, v_3 = 2.86$$

We know that theoretical mean effective pressure,

$$p_m = \frac{p_1}{r}(1 + 2.3 \log r) - p_b$$

$$= \frac{8.4}{2.86}(1 + 2.3 \log 2.86) - 1.25 = 4.77 \text{ bar}$$

$\therefore$ Actual mean effective pressure,

$$p_a = p_m \times K = 4.77 \times 0.75 = 3.58 \text{ bar} \quad \textbf{Ans.}$$

17.11. Power Developed by a Simple Steam Engine

The term 'power' may be defined as the rate of doing work. It is thus the measure of performance of a steam engine, *e.g.* an engine doing a certain amount of work in one second will be twice as powerful as an engine doing the same amount of work in two seconds. Mathematically, power developed by an engine,

$$P = \frac{\text{Work done}}{\text{Time taken}}$$

In S.I. system of units, the unit of power is watt (briefly written as W) which is equal to 1 N-m/s or 1 J/s. Generally, a bigger unit of power known as kilowatt (briefly written as kW) is used which is equal to 1000 W.

In case of steam engines, the following two terms are commonly used for the power developed :

1. Indicated power, and 2. Brake power.

The indicated power and brake power are discussed, in detail, in the following pages.

17.12. Indicated Power

The actual power generated in the engine cylinder is called *power input* or *indicated power* (briefly written as I.P.). Since the instrument used to draw the p-v diagram (from which work done during the stroke is obtained), is known as indicator, that is why this power is called indicated power. Now consider a simple steam engine, whose indicated power is required to be found out.

Let p_a = Actual mean effectivepressure in N/m^2,

A = Area of the cylinder or piston in m^2,

L = Length of the stroke in metres, and

N = Speed of the crankshaft in revolution per minute (r.p.m.).

We know that force on the piston

$$= \text{Pressure} \times \text{Area} = p_a \times A \text{ (in N)}$$

and work done per stroke $= p_a \times A \times L$ N-m

$\therefore$ Work done per minute

$$= p_a \times A \times L \times N \qquad \text{... (For single acting)}$$

$$= p_a \times A \times L \times 2N \qquad \text{... (For double acting)}$$

and indicated power, $$\text{I.P.} = \frac{p_a L A N}{60} \text{ W} \qquad \text{... (For single acting)}$$

[$\because$ 1 N-m/s or 1 J/s = 1 W]

$$= \frac{2 p_a L A N}{60} \text{ W} \qquad \text{... (For double acting)}$$

When the actual mean effective pressure is given in bar, then

$$\text{I.P.} = \frac{p_a \times 10^5 L A N}{60} \text{ W} \qquad \text{... (}\because \text{ 1 bar} = 10^5 \text{ N/m}^2$$

$$= \frac{p_a \times 10^5 \times L A N}{60 \times 1000} \text{ kW} = \frac{100\, p_a L A N}{60} \text{ kW} \qquad \text{... (For single acting)}$$

$$= \frac{200\, p_a L A N}{60} \text{ kW} \qquad \text{... (For double acting)}$$

Note : The value of actual mean effective pressure (p_a) may also be obtained from the following expression :

$$p_a = \frac{\text{Area of actual indicator diagram in m}^2 \times \text{Spring strength in bar/m}}{\text{Length of actual indicator diagram in m}} \text{ (in bar)}$$

Example 17.4. *A double acting single cylinder has 200 mm stroke, 160 mm diameter. It runs at 250 r.p.m. and the cut-off is 25% of the stroke. The pressure at cut-off is 15 bar and exhaust is at 0.3 bar for a diagram factor of 0.75. Estimate the indicated power in kW.*

Solution. Given : L = 200 mm = 0.2 m ; D = 160 mm = 0.16 m ; N = 250 r.p.m. ; v_2 = 25% of stroke = 0.25 v_3 ; p_1 = 15 bar ; p_b = 0.3 bar ; K = 0.75

We know that expansion ratio,

$$r = \frac{v_3}{v_2} = \frac{v_3}{0.25\, v_3} = 4$$

$\therefore$ Theoretical mean effective pressure,

$$p_m = \frac{p_1}{r}(1 + 2.3 \log r) - p_b = \frac{15}{4}(1 + 2.3 \log 4) - 0.3 = 8.64 \text{ bar}$$

and actual mean effective pressure,

$$p_a = p_m \times K = 8.64 \times 0.75 = 6.48 \text{ bar}$$

Area of the cylinder,

$$A = \frac{\pi}{4} \times D^2 = \frac{\pi}{4}(0.16)^2 = 0.02 \text{ m}^2$$

We know that indicated power,

$$\text{I.P.} = \frac{200\, p_a\, L\, A\, N}{60} = \frac{200 \times 6.48 \times 0.2 \times 0.02 \times 250}{60} \text{ kW}$$

$$= 21.6 \text{ kW} \quad \textbf{Ans.}$$

Example 17.5. *Calculate the indicated power and steam consumption in kg/h of a double acting steam engine from the following data :*

Diameter of cylinder = 300 mm ; Stroke = 450 mm ; R.P.M. = 120 ; Steam pressure = 7 bar, and 0.9 dry ; Back pressure = 1.2 bar ; Cut-off takes place at 32 % of stroke for both ends.

Solution. Given : $D = 300$ mm $= 0.3$ m ; $L = 450$ mm $= 0.45$ m ; $N = 120$ r.p.m. ; $p_1 = 7$ bar $x = 0.9$; $p_b = 1.2$ bar

Since the cut-off takes place at 32% of the stroke for both ends, therefore

Volume at cut-off $= 0.32 \times$ Stroke volume

$$= 0.32 \times \frac{\pi}{4} \times D^2 \times L = 0.32 \times \frac{\pi}{4}(0.3)^2\, 0.45 = 0.01 \text{ m}^3$$

and expansion ratio, $$r = \frac{1}{0.32} = 3.125$$

Indicated power

We know that actual mean effective pressure,

$$p_a = K\left[\frac{p_1}{r}(1 + 2.3 \log r) - p_b\right]$$

$$= 1\left[\frac{7}{3.125}(1 + 2.3 \log 3.125) - 1.2\right] = 3.6 \text{ bar} \quad \text{...(Taking } K = 1)$$

Area of cylinder, $$A = \frac{\pi}{4} \times D^2 = \frac{\pi}{4}(0.3)^2 = 0.07 \text{ m}^2$$

$\therefore$ Indicated power,

$$\text{I.P.} = \frac{200\, p_a\, L\, A\, N}{60} = \frac{200 \times 3.6 \times 0.45 \times 0.07 \times 120}{60} = 45.4 \text{ kW} \quad \textbf{Ans.}$$

Steam consumption

From steam tables, corresponding to a pressure of 7 bar, we find that specific volume of dry steam,

$$v_g = 0.2727 \text{ m}^3/\text{kg}$$

We know that mass of steam used per stroke

$$= \frac{\text{Volume of steam at cut-off}}{x\, v_g} = \frac{0.01}{0.9 \times 0.2727} = 0.0407 \text{ kg}$$

and mass of steam used per minute $= 0.0407 \times 2\,N = 0.0407 \times 2 \times 120 = 9.77$ kg/min

$\therefore$ Steam consumption per hour

$$= 9.77 \times 60 = 586.2 \text{ kg/h } \textbf{Ans.}$$

Example 17.6. *Determine the stroke and diameter of a double acting steam engine cylinder developing 180 kW under the following conditions :*

Initial steam pressure 7 bar ; back pressure 1.12 bar ; crank speed 100 r.p.m. ; average piston speed 135 m/min ; diagram factor 0.8 ; cut-off at 0.4 of the stroke.

Solution. Given : I.P. = 180 kW ; $p_1 = 7$ bar ; $p_b = 1.12$ bar ; $N = 100$ r.p.m. ; Average piston speed = 135 m/min ; $K = 0.8$; $v_2 = 0.4\,v_3$

Stroke of the cylinder

Let L = Length of the stroke in metres.

We know that average piston speed,

$$2\,L\,N = 135 \text{ or } L = 135/2N = 135/2 \times 100 = 0.675 \text{ m } \textbf{Ans.}$$

Diameter of the cylinder

Let D = Diameter of the cylinderin metres.

$$\therefore \qquad \text{Area, } A = \frac{\pi}{4} \times D^2 = 0.7854\,D^2 \text{ m}^2$$

First of all, let us find the actual mean effective pressure (p_a).

We know that expansion ratio,

$$r = v_3/v_2 = v_3/0.4\,v_3 = 2.5$$

$\therefore$ Theoretical mean effective pressure,

$$p_m = \frac{p_1}{r}(1 + 2.3 \log r) - p_b$$

$$= \frac{7}{2.5}(1 + 2.3 \log 2.5) - 1.12 = 4 \text{ bar}$$

and actual mean effective pressure,

$$p_a = p_m \times K = 4 \times 0.8 = 3.2 \text{ bar}$$

We know that indicated power (I.P.),

$$180 = \frac{200\,p_a\,L\,A\,N}{60} = \frac{200 \times 3.2 \times 0.675 \times 0.7854\,D^2 \times 100}{60}$$

$$= 565.5\,D^2$$

$$\therefore \qquad D^2 = 0.318 \text{ or } D = 0.564 \text{ m } \textbf{Ans.}$$

Example 17.7. *A single cylinder double acting steam engine is supplied with steam at 11.5 bar and exhaust occurs at 1.1 bar. The cut-off occurs at 40% of the stroke. If the stroke is equal to 1.25 times the cylinder bore and the engine develops an indicated power of 60 kW at 90 r.p.m., determine the bore and stroke of the engine.*

Assume hyperbolic expansion and a diagram factor of 0.79. Also estimate the theoretical steam consumption in m³/min.

Solution. Given : $p_1 = 11.5$ bar ; $p_b = 1.1$ bar ; $v_2 = 40\%\,v_3 = 0.4\,v_3$; $L = 1.25\,D$; I.P. = 60 kW ; $N = 90$ r.p.m. ; $K = 0.79$

Bore and stroke of the engine

Let D = Bore of the cylinder in metres,

L = Length of the stroke in metres = 1.25 D ... (Given)

We know that expansion ratio,

$$r = v_3 / v_2 = v_3 / 0.4\, v_3 = 2.5$$

and actual mean effective pressure,

$$p_a = K\left[\frac{p_1}{r}(1 + 2.3 \log r) - p_b\right]$$

$$= 0.79\left[\frac{11.5}{2.5}(1 + 2.3 \log 2.5) - 1.1\right] = 6.1 \text{ bar}$$

Area of the cylinder,

$$A = \frac{\pi}{4} \times D^2 = 0.7854 D^2 \text{ m}^2$$

We know that indicated power (I.P.)

$$60 = \frac{200\, p_a\, L\, A\, N}{60} = \frac{200 \times 6.1 \times 1.25\, D \times 0.7854\, D^2 \times 90}{60}$$

$$= 1797\, D^3$$

$\therefore$ $D^3 = 0.0334$ or $D = 0.322$ m **Ans.**

and $L = 1.25\, D = 1.25 \times 0.322 = 0.403$ m **Ans.**

Theoretical steam consumption

We know that stroke volume,

$$v_3 = A\, L = 0.7854\, D^2 L = 0.7854\,(0.322)^2\, 0.403 = 0.0328 \text{ m}^3$$

Since the cut-off occurs at 40% of the stroke, therefore volume of steam at the point of cut-off per stroke,

$$v_2 = 0.4 \times 0.0328 = 0.013\,12 \text{ m}^3$$

$\therefore$ Theoretical steam consumption per min.

$$= v_2 \times 2\, N = 0.013\,12 \times 2 \times 90 = 2.36 \text{ m}^3\text{/min} \quad \textbf{Ans.}$$

17.13. Effect of Piston Rod in Double Acting Steam Engines

In a double acting steam engine, the piston rod reduces the effective area of the piston during return stroke as shown in Fig. 17.6.

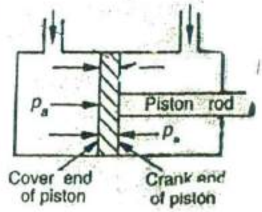


Fig. 17.6. Double acting steam engine

Let p_a = Actual mean effective pressure on each side of piston in bar,

A = Area of the piston on the cover end or back end side in m^2, and

a = Area of the piston rod in m^2.

$\therefore$ Effective area of the piston on the crank end side,

$$A_1 = (A - a) \text{ m}^2$$

We know that indicated power for the cover end or back end side of the piston,

$$(\text{I.P.})_1 = \frac{100\, p_a\, L\, A\, N}{60} \text{ kW}$$

and indicated power for the crank end side of the piston,

$$(\text{I.P.})_2 = \frac{100\, p_a\, L\, A_1\, N}{60} = \frac{100\, p_a\, L\,(A-a)\, N}{60} \text{ kW}$$

$\therefore$ Total indicated power of a double acting steam engine,

$$\text{I.P.} = (\text{I.P.})_1 + (\text{I.P.})_2$$

$$= \frac{100\, p_a\, L\, A\, N}{60} + \frac{100\, p_a\, L\,(A-a)\, N}{60} \text{ kW}$$

$$= \frac{100\, p_a\, L\, N\,(A + A - a)}{60} = \frac{100\, p_a\, L\, N\,(2A - a)}{60} \text{ kW}$$

Note : If the actual mean effective pressure on both sides of the piston is different, then total indicated power,

$$\text{I.P.} = \frac{100\, p_{a1}\, L\, A\, N}{60} + \frac{100\, p_{a2}\, L\,(A-a)\, N}{60} \text{ kW}$$

$$= \frac{100\, L\, N}{60}\,[p_{a1}\, A + p_{a2}\,(A-a)] \text{ kW}$$

Example 17.8. *Following data refer to a double acting steam engine :*

Bore = 300 mm ; Stroke = 550 mm ; Piston rod diameter = 30 mm ; Speed = 97 r.p.m. ; Base of both indicator diagrams = 100 mm ; Area of cover and crank side diagrams = 765 and 741 mm² respectively ; Spring number = 500 bar/m ; Steam pressure at inlet = 7.5 bar ; Steam pressure at exhaust = 0.3 bar ; Cut-off = 2/5 of stroke.

Neglect clearance volume of the cylinder and calculate the indicated power of the engine.

Solution : Given : $D = 300$ mm $= 0.3$ m ; $L = 550$ mm $= 0.55$ m ; $d = 30$ mm $= 0.03$ m ; $N = 97$ r.p.m. ; $b = 100$ mm $= 0.1$ m ; $a_1 = 765$ mm$^2 = 765 \times 10^{-6}$ m^2 ; $a_2 = 741$ mm$^2 = 741 \times 10^{-6}$ m^2 ; $s = 500$ bar/m ; *$p_1 = 7.5$ bar ; $p_b = 0.3$ bar ; Cut-off $= 2/5$ of stroke

We know that actual mean effective pressure for the cover side,

$$p_{a1} = \frac{\text{Area of indicator diagram for cover side} \times \text{Spring number}}{\text{Length or base of indicator diagram}}$$

$$= \frac{a_1 \times s}{b} = \frac{765 \times 10^{-6} \times 500}{0.1} = 3.825 \text{ bar}$$

Similarly, actual mean effective pressure for the crank side,

$$p_{a2} = \frac{a_2\, s}{b} = \frac{741 \times 10^{-6} \times 500}{0.1} = 3.705 \text{ bar}$$

Area of the cylinder, $A = \frac{\pi}{4} \times D^2 = \frac{\pi}{4}(0.3)^2 = 0.0707 \text{ m}^2$

and area of the piston rod, $a = \frac{\pi}{4} \times d^2 = \frac{\pi}{4}(0.03)^2 = 0.707 \times 10^{-3} \text{ m}^2$

---

* Superfluous data

We know that indicated power for the cover side,

$$(\text{I.P.})_1 = \frac{100\, p_{a1}\, L\, A\, N}{60} = \frac{100 \times 3.825 \times 0.55 \times 0.0707 \times 97}{60} = 24.04 \text{ kW}$$

and indicated power for the crank side,

$$(\text{I.P.})_2 = \frac{100\, p_{a2}\, L\,(A - a)\, N}{60} = \frac{100 \times 3.705 \times 0.55\,(0.0707 - 0.707 \times 10^{-3})\, 97}{60}$$

$$= 23.06 \text{ kW}$$

∴ Total indicated power,

$$\text{I.P.} = (\text{I.P.})_1 + (\text{I.P.})_2 = 24.04 + 23.06 = 47.1 \text{ kW } \textbf{Ans.}$$

17.14. Brake Power

The power available at the crankshaft of an engine is called *power output* or ***brake power*** (briefly written as B.P.). It has been observed that all the power generated by the engine cylinder is not available at the crankshaft for doing useful work. This happens because some of the power is utilised in overcoming the internal friction of the moving parts of the engine. This power lost in friction is known as *frictional power* (briefly written as F.P.). Thus

$$\text{B.P.} = \text{I.P} - \text{F.P.}$$

17.15. Measurement of Brake Power

The brake power of an engine is measured by an apparatus known as dynamometer. Though there are many types of dynamometers for measuring the brake power of an engine, yet the absorption type dynamometers are important from the subject point of view. These dynamometers are of the following two types :

1. Prony brake dynamometer, and 2. Rope brake dynamometer.

These dynamometers are discussed, in detail, in the following pages.

17.16. Prony Brake Dynamometer

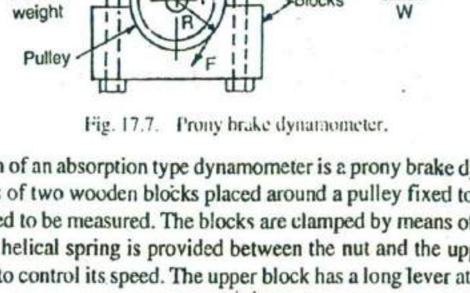


Fig. 17.7. Prony brake dynamometer.

A simplest form of an absorption type dynamometer is a prony brake dynamometer, as shown in Fig. 17.7. It consists of two wooden blocks placed around a pulley fixed to the shaft of an engine whose power is required to be measured. The blocks are clamped by means of two bolts and nuts, as shown in Fig. 17.7. A helical spring is provided between the nut and the upper block to adjust the pressure on the pulley to control its speed. The upper block has a long lever attached to it and carries a weight W at its outer end. A counter weight is placed at the other end of the lever which balances the brake when unloaded. Two stops S, S are provided to limit the motion of the lever.

When the brake is to be put in operation, the long end of the lever is loaded with suitable weights W and the nuts are tightened until the engine shaft runs at a constant speed and the lever is in horizontal position. Under these conditions, the moment due to the weight W must balance the moment of the frictional resistance between the blocks and the pulley.

Let W = Weight at the outer end of the lever in newtons.

l = Horizontal distance of the weight W from the centre of the pulley in metres,

F = Frictional resistance between the blocks and the pulley in newtons,

R = Radius of the pulley in metres, and

N = Speed of the shaft in r.p.m.

We know that the moment of the frictional resistance or torque on the shaft,

$$T = W\,l = F\,R \text{ N-m}$$

Workdone in one revolution

$$= \text{Torque} \times \text{Angle turned in radians} = T \times 2\pi \text{ N-m}$$

$\therefore$ Workdone per minute $= T \times 2\pi N$ N-m

We know that brake power of the engine,

$$\text{B.P.} = \frac{\text{Workdone per min}}{60} = \frac{T \times 2\pi N}{60} = \frac{W\,l \times 2\pi N}{60} \text{ watts}$$

Notes: 1. From the above expression, we see that while determining the brake power of an engine with the help of a prony brake dynamometer, it is not necessary to know the radius of the pulley, the coefficient of friction between the wooden blocks and the pulley and the pressure exerted by tightening of the nuts.

2. When the driving torque on the shaft is not uniform, this dynamometer is subjected to severe oscillations.

Example 17.9. *Following observations were recorded during the trial of a prony brake dynamometer.*

Weight hung from the lever = 100 N ; Distance between weight and pulley = 1.2 m ; Shaft speed = 150 r.p.m. Find the brake power of the engine.

Solution. Given : $W = 100$ N ; $l = 1.2$ m ; $N = 150$ r.p.m.

We know that brake power of the engine,

$$B.P. = \frac{W.l \times 2\pi N}{60} = \frac{100 \times 1.2 \times 2\pi \times 150}{60} \text{ W}$$

$$= 1885 \text{ W} = 1.885 \text{ kW} \textbf{ Ans.}$$

17.17. Rope Brake Dynamometer

It is another form of absorption type dynamometer which is most commonly used for measuring the brake power of the engine. It consists of one, two or more ropes wound around the flywheel or rim of a pulley fixed rigidly to the shaft of an engine. The upper end of the ropes is attached to a spring balance while the lower end of the ropes is kept in position by applying a dead weight, as shown in Fig. 17.8. In order to prevent the slipping of the rope over the flywheel, wooden blocks are placed at intervals around the circumference of the flywheel.

In the operation of the brake, the engine is made to run at a constant speed. The frictional torque, due to the rope, must be equal to the torque being transmitted by the engine.

Let W = Dead load in newtons,

S = Spring balance reading in newtons,

D = Diameter of the wheel in metres,

d = Diameter of rope in metres, and

N = Speed of the engine shaft in r.p.m.

∴ Net load on the brake

$$= (W - S) \text{ N}$$

We know that distance moved in one revolution

$$= \pi (D + d) \text{ m}$$

∴ Workdone per revolution

$$= (W - S) \pi (D + d) \text{ N-m}$$

and workdone per minute

$$= (W - S) \pi (D + d) N \text{ N-m}$$

∴ Brake power of the engine,

$$\text{B.P.} = \frac{\text{Workdone per min.}}{60}$$

$$= \frac{(W - S) \pi (D + d) N}{60} \text{ watts}$$

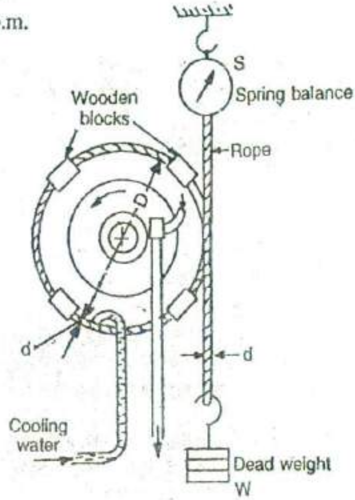


Fig. 17.8. Rope brake dynamometer.

If the diameter of the rope (d) is neglected, then brake power of the engine,

$$\text{B.P.} = \frac{(W - S) \pi DN}{60} \text{ watts}$$

Note : Since the energy produced by the engine is absorbed by the frictional resistances of the brake and is transformed into heat, therefore it is necessary to keep the flywheel of the engine cool with soapy water. The flywheels have their rims made of a channel section so as to receive a stream of water which is being whirled round by the wheel. The water is kept continually flowing into the rim and is drained away by a sharp edged scoop on the other side, as shown in Fig. 17.8.

Example 17.10. *The following data were recorded in laboratory experiment with the rope brake :*

Diameter of the flywheel = 1.2 m ; Diameter of the rope = 12.5 mm ; Engine speed = 200 r.p.m. ; Dead load on the brake = 600 N ; Spring balance reading = 150 N.

Calculate brake power of the engine.

Solution. Given : $D = 1.2$ m ; $d = 12.5$ mm $= 0.0125$ m ; $N = 200$ r.p.m. ; $W = 600$ N ; $S = 150$ N

We know that brake power of the engine,

$$\text{B.P.} = \frac{(W - S) \pi (D + d) N}{60} = \frac{(600 - 150) \pi (1.2 + 0.0125) 200}{60} \text{ W}$$

$$= 5715 \text{ W} = 5.715 \text{ kW} \textbf{ Ans.}$$

EXERCISES

1. In a reciprocating simple steam engine, the steam is supplied at 7 bar and cut-off occurs at 1/4 of the stroke. Determine the mean effective pressure, if the back pressure of the steam is 1 bar [**Ans.** 3.17 bar]

2. Steam at a pressure of 10 bar is admitted into a steam engine with expansion ratio of 4.5. If the mean effective pressure required is 3.5 bar, determine the back pressure. Neglect compression and clearance of the engine. [**Ans.** 2.06 bar]

3. Find the actual mean effective pressure of a double acting single cylinder steam engine with admission pressure of 10 bar and cut-off at 1/3 of the stroke. The exhaust pressure is 657 mm vacuum with 760 mm barometric pressure. Tak · diagram factor as 0.68. [Ans. 4.66 bar]

[Hint : p_b = ˙ .0 − 657 = 103 mm of mercury = $103 \times 133.3 = 13\,730$ N/m^2 = 0.1373 bar]

4. Calculate ina ated power of a double acting steam engine from the following data :

Diameter of th ylinder = 600 mm ; Stroke = 900 mm ; R.P.M. = 88 ; Admission pressure = 8 bar ; Back pressure = 1.8 bar.

The cut-off takes . ce at 20% of the stroke for both ends. Assume the diagram factor to be 0.8. Neglect the effect of clearance. [Ans. 141.6 kW]

5. The stroke length to diameter ratio in a double acting steam engine cylinder is 1.3 and its expansion ratio is 2.5. The engine is supplied with dry saturated steam at 9.8 bar which exhausts at 1.05 bar. The engine develops an indicated power of 185 kW at a speed of 200 r.p.m. Assuming a diagram factor of 0.8; determine the dimensions of the cylinder.

Assume hyperbolic expansion and neglect clearance. [Ans. 374.5 mm ; 486.8 mm]

6. Find out the diameter and stroke of the engine to develop 35 kW at a speed of 120 r.p.m. The piston speed is 72 m/min. The steam enters at a pressure of 11 bar for 3 / 8 of the stroke. The steam engine is to be of the condensing type and the back pressure is 0.3 bar. Assume the diagram factor to be 0.82 and no clearance. [Ans. 240 mm ; 300 mm]

7. A single cylinder double acting steam engine has piston diameter 250 mm, stroke 400 mm and diameter of the piston rod 50 mm. The mean effective pressure on both sides of the piston is 2.5 bar. Determine the indicated power when the engine runs at 200 r.p.m. [Ans. 32.08 kW]

8. A double acting steam engine with a bore of 300 mm and stroke of 400 mm runs at 300 r.p.m. The inlet is at 8 bar and the back pressure is 1.2 bar. The cut-off occurs at 30% of the stroke. Determine the power developed, taking the diagram factor as 0.8. If the steam is dry saturated at 8 bar at the point of cut-off, determine the steam consumption in kg/h. [Ans. 91.6 kW ; 1260 kg/h]

9. Dry saturated steam is supplied to a single cylinder double acting steam engine at a pressure of 9 bar and is exhausted at 1.4 bar. The cut-off takes place at 0.4 stroke and the engine develops 25 kW. The stroke-bore ratio of the engine is 1.25 and the speed 250 r.p.m. Assume diagram factor of 0.75, Calculate the cylinder bore and piston stroke of the engine. Neglect clearance and assume hyperbolic expansion.

If the actual steam consumption is 1.3 times the theoretical quantity, find specific steam consumption in kg/kWh. [Ans. 0.195 m. 0.244 m ; 21.17 kg/kW h]

10. The following observations were recorded during a test on a single acting, non condensing, single cylinder steam engine : Effective brake diameter = 2.75 m ; Net load on the brake = 1650 N ; Speed = 100 r.p.m.

Find the brake power of the engine. [Ans. 23.76 kW]

11. In a single cylinder double acting steam engine, steam is admitted at a pressure of 12 bar and is exhausted at 1.3 bar. The diameter of the cylinder is 250 mm and the stroke length is 450 mm. The cut-off of steam occurs when the piston has moved 150 mm from its I.D.C. position. The r.p.m. of the engine is 260. Neglecting clearance and assuming a diagram factor of 0.9, determine the indicated power of the engine. If the power lost in friction amounts to 7.5 kW, what will be the brake power of the engine. [Ans. 114.86 kW]

QUESTIONS

1. State the classifications of steam engine.

2. Describe, with a neat sketch, the working of a single cylinder, double acting reciprocating steam engine.

3. Explain the following terms as applied to steam engines : (*a*) clearance volume ; (*b*) stroke volume ; (*c*) cut-off volume ; (*d*) release and back pressure ; (*e*) mean effective pressure.

4. What do you understand by hypothetical indicator diagram ? Derive an expression to determine hypothetical mean effective pressure of a steam engine having clearance.

5. How and why does the hypothetical indicator diagram differ from actual indicator diagram ?

6. Discuss, with the help of pressure-volume diagram, the effect of clearance and compression on the workdone per stroke in a steam engine.

7. What is diagram factor ? State the reasons why its value is less than unity.

8. Differentiate between indicated power and brake power of a steam engine.

9. Describe the effect of piston rod in a double acting steam engine.

10. Explain the method of measuring brake power of a steam engine.

OBJECTIVE TYPE QUESTIONS

1. All steam engines work on
   (*a*) Zeroth law of thermodynamics (*b*) first law of thermodynamics
   (*c*) second law of thermodynamics (*d*) none of these
2. A single acting steam engine produces ..... power than that of double acting steam engine.
   (*a*) half (*b*) double (*c*) four times
3. The function of a D-slide valve in a steam engine is
   (*a*) to guide motion of the piston rod and prevent it from bending
   (*b*) to transfer motion from the piston to the crosshead
   (*c*) to convert heat energy of the steam into mechanical work
   (*d*) to exhaust steam from the cylinder at proper moment
4. The ratio of clearance volume to the swept volume is called
   (*a*) cut-off ratio (*b*) expansion ratio
   (*c*) clearance ratio (*d*) none of these
5. In case of condensing steam engines, the pressure of steam in the cylinder during exhaust stroke is ..... condenser pressure.
   (*a*) equal to (*b*) lower than (*c*) higher than
6. The clearance in the engine cylinder
   (*a*) increases the mean effective pressure (*b*) increases the workdone
   (*c*) decreases the efficiency of the engine (*d*) all of these
7. The diagram factor is the ratio of the
   (*a*) area of the actual indicator diagram to the area of theoretical indicator diagram
   (*b*) actual workdone per stroke to the theoretical workdone per stroke
   (*c*) actual mean effective pressure to the theoretical mean effective pressure
   (*d*) all of the above
8. The average value of diagram factor lies between
   (*a*) 0.2 to 0.5 (*b*) 0.6 to 0.65 (*c*) 0.65 to 0.9 (*d*) 0.9 to 1.2
9. For the same length of stroke and speed of crankshaft, the piston speed for a double acting steam engine is ..... the piston speed of single acting steam engine.
   (*a*) equal to (*b*) twice (*c*) four times
10. The actual power supplied by the engine crankshaft is called
   (*a*) indicated power (*b*) brake power (*c*) frictional power

ANSWERS

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

18

Compound Steam Engines

1. Introduction. 2. Arrangement of Cylinders in Compound Steam Engines. 3. Advantages of Compounding of Steam Engines. 4. Classification of Compound Steam Engines. 5. Tandem Type Compound Steam Engine. 6. Woolf Type Compound Steam Engine. 7. Receiver Type Compound Steam Engine. 8. Combined Indicator Diagram of a Compound Steam Engine. 9. Work Done and Power Developed by a Compound Steam Engine. 10. Determination of Cylinder Dimensions for a Compound Steam Engine. 11. Two Cylinder Compound Steam Engine with Complete Continuous Expansion in both the H.P. and L.P. Cylinders. 12. Two Cylinder Compound Steam Engine with Complete Expansion in H.P. Cylinder and Incomplete Expansion in L.P. Cylinder. 13. Two Cylinder Compound Steam Engine with Incomplete Expansion in both the H.P. and L.P. Cylinders. 14. Three Cylinder Compound Steam Engines.

18.1. Introduction

In the previous chapter, we have discussed the working of Simple Steam Engines. The scientists and engineers, working for the improvement of simple steam engines, thought of various methods. One of the method was to use a high pressure steam. They faced the following difficulties in expanding a high pressure steam in a single cylinder :

1. The steam, when admitted into a cylinder, comes in contact with a relatively cold cylinder walls which cause initial condensation.
2. When the steam is expanded down to the condenser pressure, there is a greater range of pressure difference. This causes a larger temperature range in the cylinder.
3. Due to the greater range of pressure difference, the ratio of expansion is large.
4. The stroke of the piston is large due to the large ratio of expansion.

To overcome these difficulties, the expansion of steam is divided into stages, each stage taking place in a different cylinder. In this method, the high pressure steam is first partly expanded in the high pressure cylinder and then exhausted into the low pressure cylinder, where the expansion is completed. This method reduces the ratio of expansion in the cylinder, as a result of which the length of stroke is also reduced. Moreover, it reduces the temperature range in each cylinder. This lead to the development of compound steam engines.

18.2. Arrangement of Cylinders in Compound Steam Engines

We have already discussed that a steam engine, in which the expansion of steam takes place, in more than one cylinder, is known as a compound steam engine. The cylinder, which receives the high pressure steam, is known as high pressure (H.P.) cylinder. The steam after expanding in the high pressure cylinder, exhausts into a larger cylinder known as low pressure (L.P.) cylinder. In this cylinder, the last stage of expansion is performed. The L.P. cylinder generally exhausts into a condenser. That is why, the compound steam engines are generally condensing type, but they may be non-condensing also. If the expansion of steam takes place in three cylinders, the engine is called triple expansion engine. Similarly, if the expansion is carried out in four cylinders, it is known quadruple expansion engine.

In case of triple expansion engines, the first stage of expansion is performed in H.P. cylinder, intermediate expansion takes place in intermediate pressure (I.P.) cylinder, and the last expansion is completed in L.P. cylinder. In quadruple expansion engine, the intermediate expansion is carried out in two I.P. cylinders.

Note : The high pressure cylinder is, generally, of smaller size than the low pressure cylinder.

18.3. Advantages of Compounding of Steam Engines

Following are the advantages of compounding the expansion of steam in two or more cylinders :

1. There is a considerable economy in steam for high pressure operations.
2. The temperature range per cylinder is reduced, with a corresponding reduction in the condensation.
3. The ratio of expansion is reduced, thus reducing the length of stroke.
4. The leakage past the valves and piston is reduced, because of the reduced pressure difference across these parts.
5. The steam can be reheated after expansion in one cylinder, and before entering the next.
6. The mechanical balance can be made more nearly perfect, and therefore high speeds are possible.
7. In case of a breakdown, the engine can be modified to continue working on reduced load.
8. More uniform turning moment is exerted on the crank shaft, by spacing the cranks at 90° in the case of a two cylinder engines or at 120° in triple expansion engines. Thus a lighter flywheel is required.
9. The forces in the working parts are reduced, as the forces are distributed over more parts.
10. The cost of the engine, for the same power and economy, is less than that of a simple steam engine.

18.4. Classification of Compound Steam Engines

The compound steam engines may be classified according to the arrangement of cranks, and the angles between them. Two-cylinder compound engines are generally classified as:

1. Tandem type compound steam engines,
2. Woolf type compound steam engines, and
3. Receiver type compound steam engines.

These compound engines are discussed, in detail, in the following pages :

18.5. Tandem Type Compound Steam Engine

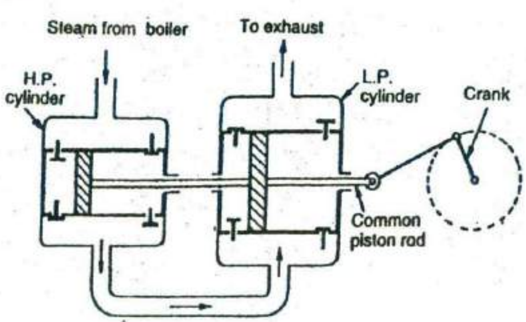


Fig. 18.1 Tandem type compound steam engine.

In this engine, the cylinders (H.P. and L.P.) have a common piston rod working on the same crank as shown in Fig. 18.1.

In a *Tandem type compound steam engine,* the steam from the boiler is admitted to one side of the high pressure cylinder. The exhaust steam, from this cylinder, passes directly into the low pressure cylinder. Since both the pistons are at the end of their strokes, these cylinders may therefore be regarded as having cranks at 0° to each other.

Fig. 18.2 shows the graph of turning moment of the crankshaft *versus* crank angle. It may be noticed from the graph that their cycles are in phase, therefore maximum and minimum turning moments on the crankshaft due to each cylinder will act at the same time (*i.e.* at the same crank angle). This is the disadvantage of this type of compound engine, as a larger flywheel is required to overcome these fluctuations in turning moment.

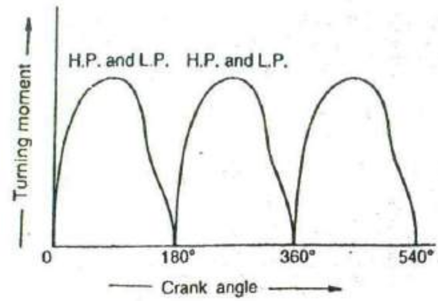


Fig. 18.2. Turning moment-crank angle graph for Tandem type compound steam engine.

18.6. Woolf Type Compound Steam Engine

In this engine, the two cylinders (H.P. and L.P.) have different piston rods attached to two differen cranks set of 180° to each other. These cranks are cast in the same crank shaft as shown in Fig. 18.3.

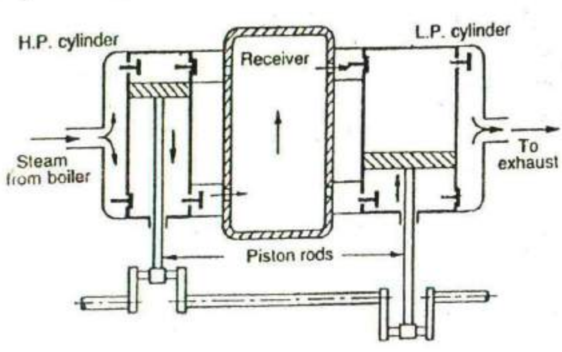


Fig. 18.3. Woolf type compound steam engine.

In a *Woolf type compound steam engine*, the steam from the boiler is admitted to one side of the high pressure cylinder. The exhaust steam, from this cylinder, passes directly into the low pressure cylinder.

Fig. 18.4 shows the graph of turning moment on the crankshaft *versus* crank angle. As the cranks are 180° apart, the two cycles are in phase and there is a large variation in the turning moment on the crankshaft, which requires a large flywheel. Thus the Woolf type compound steam engine has the same disadvantage as the Tandem type compound steam engine.

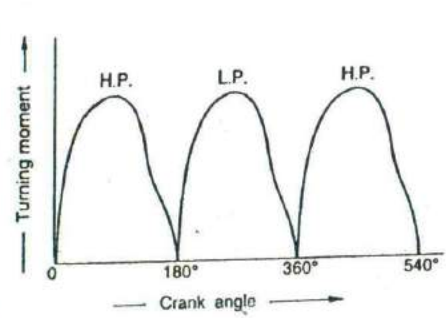


Fig. 18.4 Turning moment-crank angle graph for Woolf type compound steam engine.

18.7. Receiver Type Compound Steam Engine

In this engine, the two cylinders (H.P. and L.P.) have different piston rods attached to two different cranks set at 90° to each other. These cranks are cast in the same crank shaft as shown in Fig. 18.5.

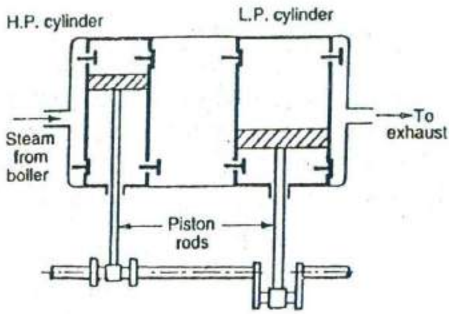


Fig. 18.5. Receiver type compound steam engine.

In a *receiver type compound steam engine*, the steam from the boiler is admitted to one side of the high pressure cylinder. Since the two cranks are set at 90°, therefore the two cylinders are out of phase. As a result of this, the steam can not directly pass from the high pressure cylinder to the low pressure cylinder. It is, therefore, essential to introduce an intermediate vessel known as receiver between the high pressure cylinder and low pressure cylinder, as shown in Fig. 18.5. The steam from the high pressure cylinder enters the receiver, from which it enters the low pressure cylinder.

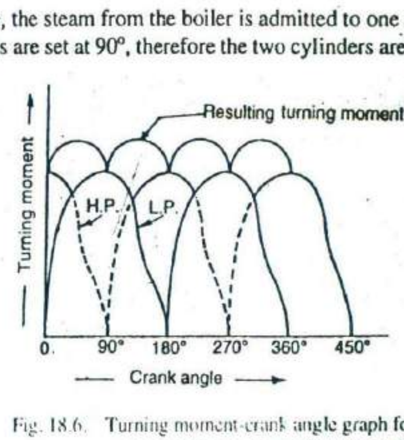


Fig. 18.6. Turning moment-crank angle graph for receiver type compound steam engine.

Fig. 18.6 shows the graph of turning moment *versus* crank angle of a receiver type compound engine. As the cranks are 90° apart, the two cycles are out of phase by 90°. The resulting turning moment diagram is also shown in Fig. 18.6. It will be noticed that variation of turning moment is considerably reduced by placing the cranks at 90°, hence a lighter flywheel is required. This is the chief advantage of the receiver type compound engine.

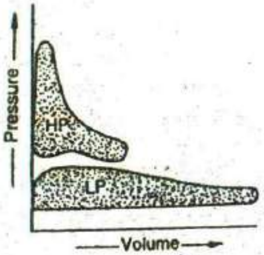
Note : Woolf type and Receiver type compound steam engines are *cross compound steam engines*. In cross compounding, the cylinders are arranged side by side and each cylinder has separate piston, connecting rod and crank.

18.8. Combined Indicator Diagram of a Compound Steam Engine

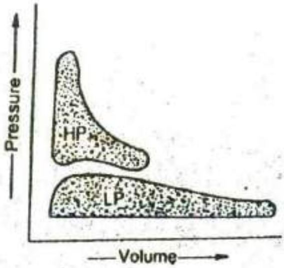
We have already discussed in the last chapter, that an indicator diagram is a graphical representation of the variation in pressure and volume of steam inside the cylinder on *p-v* diagram. In a compound steam engine, since there are two (or more) cylinders, therefore separate indicator diagrams are first drawn for H.P. and L.P. cylinders. These two diagrams are then combined together into one diagram, as shown in Fig. 18.7, as discussed below :

1. First of all draw the average indicator diagrams for both sides of the H.P. and L.P. cylinders.

2. Replot both the diagrams to the same *scale of pressure and volume.
3. Now plot the two diagrams together to give a combined indicator diagram, as shown in Fig. 18.7.



(a) Without cushion steam.



(b) With cushion steam.

Fig. 18.7. Combined indicator diagram of a compound steam engine.

Note : If cushion steam is to be shown (or in other words, the diagram is to be drawn with clearance), then the same may be done as shown in Fig. 18.7 (b).

18.9. Work Done and Power Developed by a Compound Steam Engine

We have already discussed, in the last chapter, work done and power developed by a simple steam engine. The same equation, in a slightly modified form, is used for the work done by a compound engine. The following two conditions are generally regarded as the guiding factors for the work done by a compound steam engine :

1. Total work done by a compound steam engine is shared equally by both the cylinders. Or in other words, both the H.P. and L.P. cylinders do the same amount of work.
2. The initial thrust or load on the piston of both the H.P. and L.P. cylinders is the same.

As a matter of fact, it is difficult to satisfy both the conditions from the practical point of view. But these are taken only as guiding factors in order to derive the equations for the work done.

The power developed by a double cylinder compound steam engine is given by the relation :

$$\text{I.P.} = \frac{200\, p_a\, L\, A\, N}{60}\ \text{kW}$$

where p_a = Actual mean effective pressure referred to L.P. cylinder in bar,

L = Stroke length in metres,

A = Area of the L.P. cylinder or piston in m^2, and

N = Engine speed in r.p.m.

Note : The actual mean effective pressure referred to L.P. cylinder may be obtained as discussed in Art. 18.11.

18.10. Determination of Cylinder Dimensions for a Compound Steam Engine

The estimation of cylinder dimensions is one of the most important step in the calculation and design of the compound steam engines. The common cylinder dimensions are diameter of H.P. cylinder, diameter of L.P. cylinder, stroke length etc.

In the following pages, we shall discuss following types of compound steam engines :

1. Two cylinder compound steam engine with complete continuous expansion in both the cylinders.

* It is a common practice to use different scales of pressure and volume for the indicator diagrams of H.P. and L.P. cylinders.

2. Two cylinder compound steam engine with complete expansion in H.P. cylinder but incomplete expansion in L.P. cylinder.
3. Two cylinder compound steam engine with incomplete expansion in both the cylinders.

18.11. Two Cylinder Compound Steam Engine with Complete Continuous Expansion in both the H.P. and L.P. Cylinders

Consider a two cylinder compound steam engine having complete continuous expansion of steam in both the H.P. and L.P. cylinders. Let the theoretical indicator diagram (neglecting cushion steam or clearance volume) of such an engine be drawn, as shown in Fig. 18.8.

Let p_1 = Pressure of steam admitted into the H.P. cylinder,

p_2 = Pressure of steam at the release point of H.P. cylinder (or admission point of L.P. cylinder),

p_b = Back pressure of the L.P. cylinder,

v_1 = Volume of steam admitted into the H.P. cylinder,

v_2 = Volume of steam at the release point of H.P. cylinder (or admission point of L.P. cylinder),

v_3 = Volume of steam at the release point of L.P. cylinder,

A_H = Area of H.P. cylinder,

A_L = Area of L.P. cylinder,

L = Length of stroke,

N = Speed of the engine in r.p.m, and

K = Overall diagram factor for the combined indicator diagram.

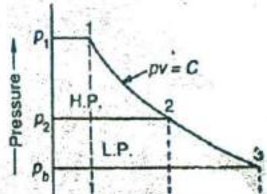


Fig. 18.8. Compound steam engine with complete continuous expansion.

We know that volume of H.P. cylinder (neglecting clearance volume),

$$v_2 = A_H \times L$$

and volume of L.P. cylinder (neglecting clearance volume),

$$v_3 = A_L \times L$$

We also know that expansion ratio in H.P. cylinder,

$$r_H = \frac{v_2}{v_1}$$

Similarly, expansion ratio in L.P. cylinder,

$$r_L = \frac{v_3}{v_2}$$

We know that the actual mean effective pressure in H.P. cylinder,

$$p_{aH} = K\left[\frac{p_1}{r_H}(1 + 2.3 \log r_H) - p_2\right]$$

and work done,

$$W_H = p_{aH} \times v_2 = K\left[\frac{p_1}{r_H}(1 + 2.3 \log r_H) - p_2\right]v_2$$

$$= K\left[\frac{p_1}{v_2/v_1}(1 + 2.3 \log r_H) - p_2\right]v_2 \qquad \ldots\left(\because r_H = \frac{v_2}{v_1}\right)$$

$$= K[p_1v_1(1+2.3\log r_H) - p_2v_2]$$

$$= K[p_1v_1 + 2.3\,p_1v_1\log r_H - p_2v_2]$$

Since expansion of steam follows the law $p\,v = C$, therefore

$$p_1v_1 = p_2v_2 = p_bv_3$$

$$\therefore \quad W_H = K \times 2.3\,p_1v_1\log r_H \qquad \ldots(i)$$

Similarly, mean effective pressure in L.P. cylinder,

$$p_{aL} = K\left[\frac{p_2}{r_L}(1+2.3\log r_L) - p_b\right]$$

and work done,

$$W_L = p_{aL} \times v_3 = K\left[\frac{p_2}{v_3/v_2}(1+2.3\log r_L) - p_b\right]v_3$$

$$= K[p_2v_2(1+2.3\log r_L) - p_bv_3]$$

$$= K[p_2v_2 + 2.3\,p_2v_2\log r_L - p_bv_3]$$

$$= K \times 2.3\,p_2v_2\log r_L \qquad \ldots(ii)$$

Now let us consider the following two conditions :

1. When the workdone is both the cylinders is equal, and
2. When the initial load or thrust on both the pistons is equal.

1. *When the workdone in both the cylinders is equal*

Considering that the workdone in both the cylinders (*i.e.* in H.P. and L.P. cylinders) is equal. Therefore equating equations (*i*) and (*ii*),

$$K \times 2.3\,p_1v_1\log r_H = K \times 2.3\,p_2v_2\log r_L$$

$$\therefore \quad r_H = r_L \qquad \ldots(\because p_1v_1 = p_2v_2)$$

or

$$\frac{v_2}{v_1} = \frac{v_3}{v_2} \qquad \ldots\left(\because r_H = \frac{v_2}{v_1} \text{ and } r_L = \frac{v_3}{v_2}\right)$$

and

$$\frac{p_1}{p_2} = \frac{p_2}{p_b} \qquad \ldots\left(\because \frac{v_2}{v_1} = \frac{p_1}{p_2} \text{ and } \frac{v_3}{v_2} = \frac{p_2}{p_b}\right)$$

or

$$p_2 = \sqrt{p_1\,p_b}$$

2. *When the initial load or thrust on both the pistons is equal*

Considering that the initial load or thrust on both the pistons (*i.e.* the pistons of both the H.P. and L.P. cylinders) is equal. Therefore

$$(p_1 - p_2)A_H = (p_2 - p_b)A_L$$

or

$$\frac{(p_1-p_2)}{(p_2-p_b)} = \frac{A_L}{A_H} = \frac{v_3}{v_2} = \frac{p_2}{p_b} \qquad \ldots\left[\because v_3 = A_L \times L;\ v_2 = A_H \times L \text{ and } \frac{v_3}{v_2} = \frac{p_2}{p_b}\right]$$

$$\therefore \quad p_1\,p_b - p_2\,p_b = p_2^2 - p_2\,p_b$$

or

$$p_2 = \sqrt{p_1\,p_b} \qquad \ldots\text{(same as above)}$$

Notes : 1. It is only a theoretical case, for double cylinder compound steam engine. However, its relations are used in a triple cylinder compound steam engine.

2. The actual mean effective presssure for the combined indicator diagram is generally known as actual mean effective pressure referred to L.P. cylinder and is given by

$$p_a = K\left[\frac{p_1}{R}(1 + 2.3 \log R) - p_b\right]$$

where $$R = \text{Total expansion ratio} = \frac{v_3}{v_1} = \frac{v_2}{v_1} \times \frac{v_3}{v_2} = r_H \times r_L$$

and total workdone, $$W = p_a \times v_3 = K\left[\frac{p_1}{R}(1 + 2.3 \log R) - p_b\right] v_3$$

18.12. Two Cylinder Compound Steam Engine with Complete Expansion in H.P. Cylinder and Incomplete Expansion in L.P. Cylinder

Consider a two cylinder compound steam engine having complete expansion in H.P. cylinder and incomplete expansion in L.P. cylinder. Let the theoretical indicator diagram (neglecting cushion steam or clearance volume) of such an engine be drawn as shown in Fig. 18.9.

Let p_1 = Pressure of steam admitted into the H.P. cylinder,

p_2 = Pressure of steam at the release point of H.P. cylinder (or admission point of L.P. cylinder),

p_3 = Pressure of steam at the release point of L.P. cylinder,

p_b = Back pressure of steam in L.P. cylinder,

v_1 = Volume of steam admitted into the H.P. cylinder,

v_2 = Volume of steam at the release point of H.P. cylinder (or admission point of L.P. cylinder),

v_3 = Volume of steam at the release point of L.P. cylinder,

A_H = Area of H.P. cylinder,

A_L = Area of L.P. cylinder,

L = Length of stroke,

N = Speed of the engine in r.p.m., and

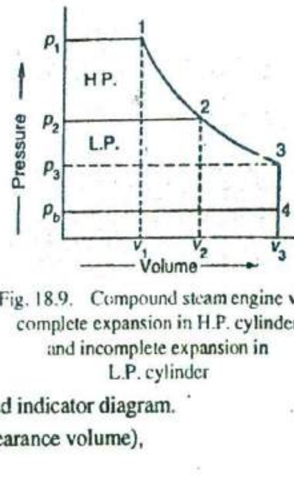


Fig. 18.9. Compound steam engine with complete expansion in H.P. cylinder and incomplete expansion in L.P. cylinder

K = Overall diagram factor for the combined indicator diagram.

We know that volume of H.P. cylinder (neglecting clearance volume),

$$v_2 = A_H \times L$$

and volume of L.P. cylinder (neglecting clearance volume),

$$v_3 = A_L \times L$$

We also know that expansion ratio in H.P. cylinder,

$$r_H = \frac{v_2}{v_1}$$

Similarly, expansion ratio in L.P. cylinder,

$$r_L = \frac{v_3}{v_2}$$

We know that actual mean effective pressure in H.P. cylinder,

$$p_{aH} = K\left[\frac{p_1}{r_H}(1 + 2.3 \log r_H) - p_2\right]$$

and work done.

$$W_H = p_{aH} \times v_2 = K\left[\frac{p_1}{r_H}(1 + 2.3 \log r_H) - p_2\right]v_2$$

$$= K\left[\frac{p_1}{v_2/v_1}(1 + 2.3 \log r_H) - p_2\right]v_2$$

$$= K\,[p_1 v_1 (1 + 2.3 \log r_H) - p_2 v_2]$$

$$= K\,[p_1 v_1 + 2.3\, p_1 v_1 \log r_H - p_2 v_2]$$

Since expansion of steam follows the law $p\,v = C$, therefore

$$p_1 v_1 = p_2 v_2 = p_3 v_3$$

$\therefore$

$$W_H = K \times 2.3\, p_1 v_1 \log r_H \qquad \ldots (i)$$

Similarly, mean effective pressure in L.P. cylinder,

$$p_{aL} = K\left[\frac{p_2}{r_L}(1 + 2.3 \log r_L) - p_b\right]$$

and work done,

$$W_L = p_{aL} \times v_3 = K\left[\frac{p_2}{r_1}(1 + 2.3 \log r_L) - p_b\right]v_3$$

$$= K\left[\frac{p_2}{v_3/v_2}(1 + 2.3 \log r_L) - p_b\right]v_3$$

$$= K\,[p_2 v_2 (1 + 2.3 \log r_L) - p_b v_3] \qquad \ldots (ii)$$

Now let us consider the following two conditions :

1. When the workdone in both the cylinders is equal, and
2. When the initial load or thrust on both the pistons is equal.

1. *When the workdone in both the cylinders is equal*

Considering that the workdone in both the cylinders (*i.e.* in H.P. and L.P. cylinders) is equal. Therefore equating equations (*i*) and (*ii*),

$$K \times 2.3\, p_1 v_1 \log r_H = K\,[p_2 v_2 (1 + 2.3 \log r_L) - p_b v_3]$$

$$2.3\, p_1 v_1 \log r_H = p_2 v_2 (1 + 2.3 \log r_L) - p_b v_3$$

$$2.3 \log r_H = \frac{p_2 v_2}{p_1 v_1}(1 + 2.3 \log r_L) - \frac{p_b v_3}{p_1 v_1}$$

$$= 1 + 2.3 \log r_L - \frac{p_b v_3}{p_1 v_1} \qquad \ldots (\because p_1 v_1 = p_2 v_2)$$

$$2.3 \log\left(\frac{r_H}{r_L}\right) = 1 - \frac{p_b v_3}{p_1 v_1}$$

$$2.3 \log\left(\frac{v_2}{v_1} \times \frac{v_2}{v_3}\right) = 1 - \frac{p_b v_3}{p_1 v_1}$$

$$2.3 \log\left(\frac{v_2^2}{v_1 v_3}\right) = 1 - \frac{p_b v_3}{p_1 v_1}$$

2. *When the initial load or thrust on both the pistons is equal*

Considering that the initial load or thrust on both the pistons (*i.e.* the piston of both the H.P. and L.P. cylinders) is equal. Therefore

$$(p_1 - p_2) A_H = (p_2 - p_b) A_L$$

or
$$\frac{p_1 - p_2}{p_2 - p_b} = \frac{A_L}{A_H} = \frac{v_3}{v_2} \qquad \ldots (\because v_3 = A_L \times L \text{ and } v_2 = A_H \times L)$$

Example 18.1. *A two cylinder compound steam engine is to develop 90 kW at 110 r.p.m. The steam is supplied at 7.35 bar and the condenser pressure is 0.21 bar. The stroke of each piston is equal to L.P. cylinder diameter. The total expansion ratio is 15. Allow a diagram factor of 0.7. Assume hyperbolic expansion and neglect clearance and receiver loss. Determine the diameter of the cylinders so that they may develop equal power.*

Solution. Given : I.P. = 90 kW ; N = 110 r.p.m. ; p_1 = 7.35 bar ; p_b = 0.21 bar ; $L = D_L$; $R = v_3/v_1 = 15$; $K = 0.7$

Diameter of L.P. cylinder

Let D_L = Diameter of L.P. cylinder is metres,

A = Area of L.P. cylinder in m² $= \frac{\pi}{4}(D_L)^2$

$= 0.7854\,(D_L)^2$

L = Length of stroke in metres = D_L ... (Given)

We know that actual mean effective pressure referred to L.P. cylinder,

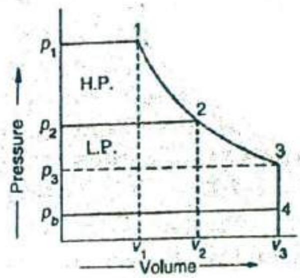


Fig. 18.10

$$p_a = K\left[\frac{p_1}{R}(1 + 2.3 \log R) - p_b\right]$$

$$= 0.7\left[\frac{7.35}{15}(1 + 2.3 \log 15) - 0.21\right] = 1.124 \text{ bar}$$

We also know that indicated power (I.P.),

$$90 = \frac{200\, p_a L A N}{60} = \frac{200 \times 1.124 \times D_L\,(0.7854\,(D_L)^2\, 110}{60} \text{ kW}$$

$$= 323.7\,(D_L)^2$$

$\therefore$ $(D_L)^2 = 0.278$ or $D_L = 0.527$ m **Ans.**

Diameter of H.P. cylinder

Let D_H = Diameter of H.P. cylinder in metres, and

A_H = Area of H.P. cylinder in m² $= \frac{\pi}{4}(D_H)^2 = 0.7854\,(D_H)^2$ m²

For equal powers in H.P. and L.P. cylinders, we know that

$$2.3 \log\left(\frac{v_2^2}{v_1 v_3}\right) = 1 - \frac{p_b v_3}{p_1 v_1}$$

$$2.3 \log\left(\frac{v_2^2}{v_1 \times 15 v_1}\right) = 1 - \frac{0.21}{7.35} \times 15 = 0.5714 \qquad \ldots (\because v_3/v_1 = 15)$$

$$\log\left(\frac{v_2^2}{15 v_1^2}\right) = \frac{0.5714}{2.3} = 0.2484$$

$$\frac{v_2^2}{15 v_1^2} = 1.772 \qquad \ldots \text{(Taking antilog of 0.2484)}$$

$$\therefore \qquad \frac{v_2}{v_1} = \sqrt{15 \times 1.772} = 5.155$$

We know that $$\frac{v_2}{v_3} = \frac{v_2}{v_1} \times \frac{v_1}{v_3} = \frac{5.155}{15} = 0.3437 \qquad \ldots (\because v_3 = 15 v_1)$$

or $$\frac{0.7854 (D_H)^2 L}{0.7854 (D_L)^2 L} = 0.3437$$

$$\therefore \qquad D_H = \sqrt{0.3437 (D_L)^2} = \sqrt{0.3437 (0.527)^2} = 0.309 \text{ m} \quad \textbf{Ans.}$$

Example 18.2. *The following data refer to a double acting compound steam engine :*

I.P. = 375 kW ; R.P.M = 420 ; Stroke = 600 mm ; Admission pressure = 10 bar ; Back pressure = 0.3 bar ; Expansion ratio = 10 ; Diagram factor = 0.8.

Assuming complete expansion in H.P. cylinder and equal initial load and expansion follows the law p v = constant and neglecting clearance, determine

1. The admission pressure for the low pressure cylinder, and 2. The diameter of each cylinder.

Solution : Given : I.P. = 375 kW ; N = 420 r.p.m. ; L = 600 mm = 0.6 m ; p_1 = 10 bar ; p_b = 0.3 bar ; $R = v_3/v_1 = 10$; $K = 0.8$

1. Admission pressure for the low pressure cylinder

Let p_2 = Admission pressure for the low pressure cylinder,

D_L = Diameter of L.P. cylinder,

A_L = Area of L.P. cylinder,

$= \frac{\pi}{4} (D_L)^2 = 0.7854 (D_L)^2$,

D_H = Diameter of H.P. cylinder, and

A_H = Area of H.P. cylinder,

$= \frac{\pi}{4} (D_H)^2 = 0.7854 (D_H)^2$

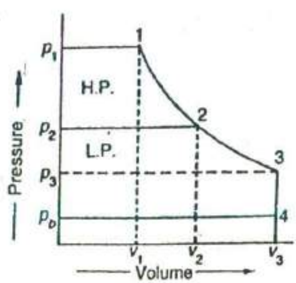


Fig. 18.11

Since the expansion follows the law $p v$ = constant, therefore

$$p_1 v_1 = p_2 v_2 = p_3 v_3$$

$$\therefore \qquad p_3 = \frac{p_1 v_1}{v_3} = \frac{10}{10} = 1 \text{ bar} \qquad \ldots (\because v_3/v_1 = 10)$$

and
$$p_2 = \frac{p_3 v_3}{v_2} = \frac{p_3 A_L L}{A_H L} = \frac{1 \times 0.7854\,(D_L)^2 L}{0.7854\,(D_H)^2 L} = \frac{(D_L)^2}{(D_H)^2} \qquad \ldots (i)$$

For equal initial load on the pistons of H.P. and L.P. cylinders, we know that

$$(p_1 - p_2) A_H = (p_2 - p_b) A_L$$

$$\frac{p_1 - p_2}{p_2 - p_b} = \frac{A_L}{A_H} = \frac{0.7854\,(D_L)^2}{0.7854\,(D_H)^2} = \frac{(D_L)^2}{(D_H)^2} \qquad \ldots (ii)$$

From equations (*i*) and (*ii*),

$$\frac{p_1 - p_2}{p_2 - p_b} = p_2 \quad \text{or} \quad p_1 - p_2 = p_2^2 - p_2 p_b$$

$$\therefore \qquad 10 - p_2 = p_2^2 - p_2 \times 0.3 \quad \text{or} \quad p_2^2 + 0.7\, p_2 - 10 = 0$$

$$\therefore \qquad p_2 = \frac{-0.7 \pm \sqrt{(0.7)^2 + 4 \times 10}}{2} = \frac{-0.7 \pm 6.36}{2}$$

$$= 2.83 \text{ bar } \textbf{Ans.} \qquad \ldots \text{(Taking +ve sign)}$$

2. *Diameter of each cylinder*

We know that actual mean effective pressure referred to L.P. cylinder,

$$p_a = K\left[\frac{p_1}{R}(1 + 2.3 \log R) - p_b\right]$$

$$= 0.8\left[\frac{10}{10}(1 + 2.3 \log 10) - 0.3\right] = 2.4 \text{ bar}$$

and indicated power (I.P.),
$$375 = \frac{200 \times p_a L A N}{60} = \frac{200 \times 2.4 \times 0.6 \times 0.7854\,(D_L)^2\, 420}{60} \text{ kW}$$

$$= 1583\,(D_L)^2$$

$$\therefore \qquad (D_L)^2 = 0.237 \quad \text{or} \quad D_L = 0.4867 \text{ m } \textbf{Ans.}$$

From equation (*i*),
$$(D_H)^2 = \frac{(D_L)^2}{p_2} = \frac{0.237}{2.83} = 0.0837$$

$$\therefore \qquad D_H = 0.2894 \text{ m } \textbf{Ans.}$$

Example 18.3. *A double acting compound steam engine with two cylinders is supplied with steam at 14 bar and 0.9 dry. The steam is exhausted into the condenser at 0.35 bar. Both the cylinders have stroke length of 350 mm and have equal loads on their pistons initially. The diameters of H.P. and L.P. cylinders are 200 mm and 300 mm respectively. If the engine runs at 300 r.p.m., find : 1. intermediate pressure ; 2. indicated power ; and 3. steam consumption of the engine in kg/hour. Assume diagram factor as 0.8 and complete expansion of steam in H.P. cylinder.*

Solution. Given : $p_1 = 14$ bar ; $x = 0.9$; $p_b = 0.35$ bar ; $L = 350$ mm $= 0.35$ m ; $D_H = 200$ mm $= 0.2$ m ; $D_L = 300$ mm $= 0.3$ m ; $N = 300$ r.p.m. ; $K = 0.8$

We know that volume of H.P. cylinder,

$$v_2 = \frac{\pi}{4}(D_H)^2 L = \frac{\pi}{4}(0.2)^2\, 0.35 = 0.011 \text{ m}^3$$

and volume of L.P. cylinder,

$$v_3 = \frac{\pi}{4}(D_L)^2 L = \frac{\pi}{4}(0.3)^2\, 0.35 = 0.0247 \text{ m}^3$$

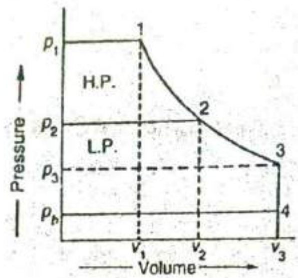


Fig. 18.12

1. *Intermediate pressure*

Let p_2 = Intermediate pressure.

We know that for equal initial load on the pistons,

$$\frac{(p_1 - p_2)}{(p_2 - p_b)} = \frac{v_3}{v_2}$$

$$\frac{(14 - p_2)}{(p_2 - 0.35)} = \frac{0.0247}{0.011} = 2.25$$

$$14 - p_2 = 2.25\, p_2 - 2.25 \times 0.35 \quad \text{or} \quad p_2 = 4.55 \text{ bar} \quad \textbf{Ans.}$$

2. *Indicated power*

Since expansion of the steam is hyperbolic, therefore

$$p_1 v_1 = p_2 v_2$$

or

$$v_1 = \frac{p_2 v_2}{p_1} = \frac{4.55 \times 0.011}{14} = 0.0036 \text{ m}^3$$

$\therefore$ Total expansion ratio, $R = \dfrac{v_3}{v_1} = \dfrac{0.0247}{0.0036} = 6.86$

We know that actual mean effective pressure referred to L.P. cylinder,

$$p_a = K\left[\frac{p_1}{R}(1 + 2.3 \log R) - p_b\right]$$

$$= 0.8\left[\frac{14}{6.86}(1 + 2.3 \log 6.86) - 0.35\right] = 4.5 \text{ bar}$$

Area of L.P. cylinder, $A = \dfrac{\pi}{4}(D_L)^2 = \dfrac{\pi}{4}(0.3)^2 = 0.0707 \text{ m}^2$

We know that indicated power,

$$\text{I.P.} = \frac{200\, p_a\, L\, A\, N}{60} = \frac{200 \times 4.5 \times 0.35 \times 0.0707 \times 300}{60} \text{ kW}$$

$$= 111.3 \text{ kW} \quad \textbf{Ans.}$$

3. *Steam consumption of the engine in kg/hour*

From the steam tables, we find that the specific volume of steam at 14 bar (*i.e.* admission pressure),

$$v_s = 0.1407 \text{ m}^3/\text{kg}$$

Since the steam admitted into the H.P. cylinder is 0.9 dry, therefore actual volume of steam (this is equal to the steam admitted in one stroke).

$$= x\,v_x = 0.9 \times 0.1407 = 0.1266 \text{ m}^3/\text{kg}$$

Total volume of steam admitted (or consumed) in one hour

$$= v_1 \times 2N \times 60 = 0.0036 \times 2 \times 300 \times 60 = 129.6 \text{ m}^3/\text{h}$$

$\therefore$ Steam consumption of the engine

$$= \frac{129.6}{0.1266} = 1023.7 \text{ kg/h } \textbf{Ans.}$$

18.13. Two Cylinder Compound Steam Engine with Incomplete Expansion in both the H.P. and L.P. Cylinders

Consider a two cylinder compound steam engine having incomplete expansion in both the H.P. and L.P. cylinders. Let the theoretical indicator diagram (neglecting cushion steam or clearance volume) of such an engine be drawn as shown in Fig. 18.13.

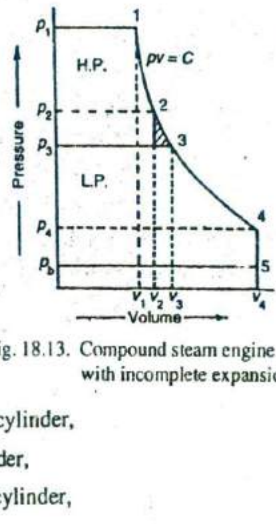


Fig. 18.13. Compound steam engine with incomplete expansion.

Let p_1 = Pressure of steam admitted into the H.P. cylinder,

p_2 = Pressure of steam at release point of H.P. cylinder,

p_3 = Pressure of steam admitted into the L.P. cylinder,

p_4 = Pressure of steam at the release point of L.P. cylinder,

p_b = Back pressure of steam in L.P. cylinder,

v_1 = Volume of steam admitted into the H.P. cylinder,

v_2 = Volume of steam at the release point of H.P. cylinder,

v_3 = Volume of steam admitted into the L.P. cylinder,

v_4 = Volume of steam at the release point of L.P. cylinder,

A_H = Area of H.P. cylinder,

A_L = Area of L.P. cylinder,

L = Length of stroke,

N = Speed of the engine in r.p.m, and

K = Overall diagram factor for the combined indicator diagram.

We know that volume of H.P. cylinder (neglecting clearance volume),

$$v_2 = A_H \times L$$

and volume of L.P. cylinder (neglecting clearance volume),

$$v_4 = A_L \times L$$

We also know that expansion ratio in H.P. cylinder,

$$r_H = \frac{v_2}{v_1}$$

Similarly, expansion ratio in L.P. cylinder,

$$r_L = \frac{v_4}{v_3}$$

We know that actual mean effective pressure in H.P. cylinder,

$$p_{aH} = K\left[\frac{p_1}{r_H}(1 + 2.3 \log r_H) - p_3\right]$$

and work done,
$$W_H = p_{aH} \times v_2 = K\left[\frac{p_1}{r_H}(1 + 2.3 \log r_H) - p_3\right]v_2 \qquad \ldots (i)$$

Similarly, actual mean effective pressure in L.P. cylinder,

$$p_{aL} = K\left[\frac{p_3}{r_L}(1 + 2.3 \log r_L) - p_b\right]$$

and work done,
$$W_L = p_{aL} \times v_4 = K\left[\frac{p_3}{r_L}(1 + 2.3 \log r_L) - p_b\right]v_4 \qquad \ldots (ii)$$

Now let us consider the following two conditions :

1. When the workdone in both the cylinders is equal, and
2. When the initial load or thrust on both the pistons is equal.

1. *When the workdone in both the cylinders is equal*

Considering that the workdone in both the cylinders (*i.e.* in H.P. and L.P. cylinders) is equal. Therefore equating equations (*i*) and (*ii*),

$$K\left[\frac{p_1}{r_H}(1 + 2.3 \log r_H) - p_3\right]v_2$$

$$= K\left[\frac{p_3}{r_L}(1 + 2.3 \log r_L) - p_b\right]v_4$$

or
$$\left[\frac{p_1}{r_H}(1 + 2.3 \log r_H) - p_3\right]$$

$$= \frac{v_4}{v_2}\left[\frac{p_3}{r_L}(1 + 2.3 \log r_L) - p_b\right] = \frac{R}{r_H}\left[\frac{p_3}{r_L}(1 + 2.3 \log r_L) - p_b\right]$$

$$\ldots \left(\because \frac{v_4}{v_2} = \frac{v_4}{v_1} \times \frac{v_1}{v_2} = R \times \frac{1}{r_H}\right)$$

2. *When the initial load or thrust on both the pistons is equal*

Considering that the initial load or thrust on both the pistons (*i.e.* pistons of both the H.P. and L.P. cylinders) is equal. Therefore

$$(p_1 - p_3)A_H = (p_3 - p_b)A_L$$

or
$$\frac{p_1 - p_3}{p_3 - p_b} = \frac{A_L}{A_H} = \frac{v_4}{v_2} \qquad \ldots \left(\because v_4 = A_L \times L \text{ and } v_2 = A_H \times L\right)$$

Example 18.4. *The following data refer to a double acting two cylinder compound steam engine :*

	H.P. cylinder	*L.P. cylinder*
Piston diameter	*250 mm*	*450 mm*
Stroke length	*600 mm*	*600 mm*
Cut-off (percentage of stroke length)	*25%*	*35%*
Expansion follows the law	*hyperbolic*	*hyperbolic*
Diagram factor	*0.75*	*0.65*
R.P.M.	*120*	*120*

If the steam is supplied to H.P. cylinder at 10 bar and exhaust takes place at 0.11 bar, determine : 1. Mean effective pressure of H.P. and L.P. cylinders ; and 2. Ratio of workdone in two cylinders. Assume no clearance and compression.

Solution. Given : $D_H = 250$ mm $= 0.25$ m ; $D_L = 450$ mm $= 0.45$ m ; $L = 600$ mm $= 0.6$ m ; $v_1 = 25\%\ v_2 = 0.25\ v_2$; $v_3 = 35\%\ v_4 = 0.35\ v_4$; $K_H = 0.75$; $K_L = = 0.65$; $N = 120$ r.p.m. ; $p_1 = 10$ bar ; $p_b = 0.11$ bar

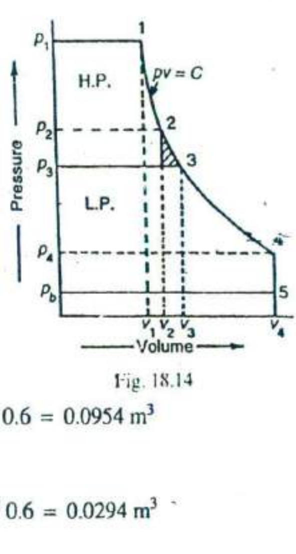


Fig. 18.14

1. *Mean effective pressure of H.P. and L.P. cylinders*

We know that expansion ratio in H.P. cylinder,

$$r_H = \frac{v_2}{v_1} = \frac{v_2}{0.25\, v_2} = 4$$

and expansion ratio in L.P. cylinder,

$$r_L = \frac{v_4}{v_3} = \frac{v_4}{0.35\, v_4} = 2.857$$

We also know that volume of L.P. cylinder,

$$v_4 = A_L \times L = \frac{\pi}{4}(D_L)^2 L = \frac{\pi}{4}(0.45)^2\, 0.6 = 0.0954 \text{ m}^3$$

and volume of H.P. cylinder,

$$v_2 = A_H \times L = \frac{\pi}{4}(D_H)^2 L = \frac{\pi}{4}(0.25)^2\, 0.6 = 0.0294 \text{ m}^3$$

$$\therefore \quad \frac{v_4}{v_2} = \frac{0.0954}{0.0294} = 3.245$$

Since the expansion is hyperbolic (*i.e.* $p\,v$ = constant), therefore

$$p_1 v_1 = p_2 v_2 = p_3 v_3 = p_4 v_4$$

$$\therefore \quad p_3 = \frac{p_1 v_1}{v_3} = p_1 \times \frac{v_1}{v_2} \times \frac{v_2}{v_4} \times \frac{v_4}{v_3} = 10 \times \frac{1}{4} \times \frac{1}{3.245} \times 2.857 = 2.2 \text{ bar}$$

We know that actual mean effective pressure of H.P. cylinder,

$$p_{aH} = K_H \left[\frac{p_1}{r_H}(1 + 2.3 \log r_H) - p_3 \right]$$

$$= 0.75 \left[\frac{10}{4}(1 + 2.3 \log 4) - 2.2 \right] = 2.82 \text{ bar} \quad \textbf{Ans.}$$

and actual mean effective pressure of L.P. cylinder,

$$p_{aL} = K_L\left[\frac{p_3}{r_L}(1 + 2.3 \log r_L) - p_b\right]$$

$$= 0.65\left[\frac{2.2}{2.857}(1 + 2.3 \log 2.857) - 0.11\right] = 0.954 \text{ bar} \quad \textbf{Ans.}$$

2. *Ratio of workdone in H.P. and ' .P. cylinders*

We know that workdone in H.P. cylinder per minute,

$$W_H = p_{aH} \times v_2 \times N = 2.82 \times 10^5 \times 0.0294 \times 120 = 9.92 \times 10^5 \text{ N-m}$$

... (p_{aH} is taken in N/m²)

and workdone in L.P. cylinder per minute

$$W_L = p_{aL} \times v_4 \times N = 0.954 \times 10^5 \times 0.0954 \times 120 = 10.92 \times 10^5 \text{ N-m}$$

... (p_{aL} is taken in N/m²)

$$\therefore \qquad \frac{W_H}{W_L} = \frac{9.95 \times 10^5}{10.92 \times 10^5} = 0.911 \quad \textbf{Ans.}$$

Example 18.5. *The steam is supplied at 7.5 bar to a double acting, two cylinder compound steam engine. The back pressure is 0.2 bar. If the cylinder volume ratio is 3.5, cut-off in H.P. cylinder is 40% of stroke, and cut-off in L.P. cylinder is 53% of stroke, determine the L.P. receiver pressure. Also compare the initial loads on the piston.*

Assume hyperbolic expansion and neglect clearance.

Solution. Given : $p_1 = 7.5$ bar ; $p_b = 0.2$ bar ; $v_4/v_2 = 3.5$; $v_1 = 40\%$ $v_2 = 0.4\, v_2$; $v_3 = 53\%\, v_4 = 0.53\, v_4$

We know that ratio of expansion in H.P. cylinder,

$$r_H = \frac{v_2}{v_1} = \frac{v_2}{0.4\, v_2} = 2.5$$

and ratio of expansion in L.P. cylinder,

$$r_L = \frac{v_4}{v_3} = \frac{v_4}{0.53\, v_4} = 1.89$$

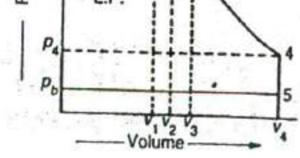


Fig. 18.15

L.P. receiver pressure

Let p_3 = L.P. receiver pressure.

Since the expansion is assumed hyperbolic, therefore

$$p_1 v_1 = p_3 v_3$$

or

$$p_1 \times \frac{v_1}{v_2} \times v_2 = p_3 \times \frac{v_3}{v_4} \times v_4$$

$$p_1 \times \frac{1}{r_H} = p_3 \times \frac{1}{r_L} \times \frac{v_4}{v_2}$$

$$p_3 = p_1 \times \frac{r_L}{r_H} \times \frac{v_2}{v_4} = 7.5 \times \frac{1.89}{2.5} \times \frac{1}{3.5} = 1.62 \text{ bar} \quad \textbf{Ans.}$$

Comparison of initial loads on the piston

We know that

$$\frac{\text{Initial load on H.P. piston}}{\text{Initial load on L.P. piston}} = \frac{(p_1 - p_3)A_H}{(p_3 - p_b)A_L} = \frac{(p_1 - p_3)v_2}{(p_3 - p_b)v_4} \quad \ldots (\because A_H = v_2/L \text{ and } A_L = v_4/L)$$

$$= \frac{7.5 - 1.62}{1.62 - 0.2} \times \frac{1}{3.5} = 1.183 \text{ Ans.}$$

Example 18.6. *A compound steam engine is to develop 260 kW when taking steam at 8.75 bar and exhausting at 0.15 bar. The engine speed is 140 r.p.m. and the piston speed is 150 m/min. The cut-off in the H.P. cylinder is to be 0.4 and the cylinder volume ratio is 3.7. Allow a diagram factor of 0.83 for the combined cards and determine suitable dimensions of the cylinders. If the diagram factor for the H.P. cylinder alone is 0.85, determine the separate powers developed in the two cylinders when the L.P. cut-off is arranged to give equal initial loads on the pistons. Assume hyperbolic expansion and neglect clearance effects.*

Solution. Given : I.P. = 260 kW ; $p_1 = 8.75$ bar ; $p_b = 0.15$ bar ; $N = 140$ r.p.m. ; Piston speed = 150 m/min ; $v_1 = 0.4\, v_2$; $v_4/v_2 = 3.7$; $K = 0.83$; $K_H = 0.85$

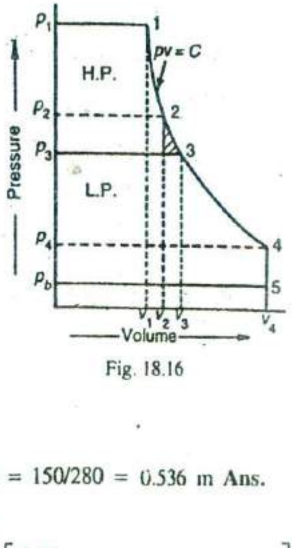


Fig. 18.16

We know that expansion ratio in H.P. cylinder,

$$r_H = \frac{v_2}{v_1} = \frac{v_2}{0.4\, v_2} = 2.5$$

and total expansion ratio,

$$R = \frac{v_4}{v_1} = \frac{v_4}{v_2} \times \frac{v_2}{v_1} = 3.7 \times 2.5 = 9.25$$

Suitable dimensions of the cylinders

Let L = Length of stroke in metres,

D_L = Diameter of L.P. cylinder in metres, and

D_H = Diameter of H.P. cylinder in metres.

We know that piston speed,

$$150 = 2\,L\,N = 2\,L \times 140 = 280\,L \quad \text{or} \quad L = 150/280 = 0.536 \text{ m Ans.}$$

Actual mean effective pressure referred to L.P. cylinder,

$$p_a = K\left[\frac{p_1}{R}(1 + 2.3 \log R) - p_b\right] = 0.83\left[\frac{8.75}{9.25}(1 + 2.3 \log 9.25) - 0.15\right]$$

$$= 2.4 \text{ bar}$$

Area of L.P. cylinder,

$$A = \frac{\pi}{4}(D_L)^2 = 0.7854\,(D_L)^2 \text{ m}^2$$

We know that indicated power (I.P.),

$$260 = \frac{200\, p_a\, L\, A\, N}{60} = \frac{200 \times 2.4 \times 0.536 \times 0.7854\,(D_L)^2\, 140}{60} = 470\,(D_L)^2$$

$$\therefore \quad (D_L)^2 = 0.553 \quad \text{or} \quad D_L = 0.744 \text{ m Ans.}$$

We know that ratio of L.P. cylinder volume to H.P. cylinder volume,

$$3.7 = \frac{v_4}{v_2} = \frac{0.7854\,(D_L)^2\,L}{0.7854\,(D_H)^2\,L} = \frac{(D_L)^2}{(D_H)^2}$$

$$\therefore \quad (D_H)^2 = \frac{(D_L)^2}{3.7} = \frac{0.553}{3.7} = 0.1495 \text{ or } D_H = 0.386 \text{ m} \textbf{ Ans.}$$

Power developed in H.P. cylinder

First of all, let us find the pressure of steam at release point of H.P. cylinder (*i.e.* p_3).

We know that for equal initial loads on both the pistons,

$$\frac{p_1 - p_3}{p_3 - p_b} = \frac{v_4}{v_2} = 3.7$$

$$\frac{8.75 - p_3}{p_3 - 0.15} = 3.7 \text{ or } 8.75 - p_3 = 3.7\,p_3 - 3.7 \times 0.15$$

$$\therefore \quad p_3 = 1.98 \text{ bar}$$

Actual mean effective pressure in H.P. cylinder,

$$p_{aH} = K_H \left[\frac{p_1}{r_H}(1 + 2.3 \log r_H) - p_3 \right]$$

$$= 0.85 \left[\frac{8.75}{2.5}(1 + 2.3 \log 2.5) - 1.98 \right] = 4 \text{ bar}$$

Area of H.P. cylinder, $A_H = \frac{\pi}{4}(D_H)^2 = \frac{\pi}{4}(0.386)^2 = 0.1174 \text{ m}^2$

$\therefore$ Power developed in H.P. cylinder

$$= \frac{200\, p_{aH}\, L\, A_H\, N}{60} = \frac{200 \times 4 \times 0.536 \times 0.1174 \times 140}{60} \text{ kW}$$

$$= 117.4 \text{ kW} \textbf{ Ans.}$$

and power developed in L.P. cylinder

$$= 260 - 117.4 = 142.6 \text{ kW} \textbf{ Ans.}$$

Example 18.7. *A compound steam engine develops 200 kW at 150 r.p.m. when the steam enters the H.P. cylinder at 10 bar and leaves the L.P. cylinder at 0.2 bar. Cut-off takes place at 0.45 stroke in H.P. cylinder and ratio of L.P. cylinder volume to H.P. cylinder volume is 3.2. Calculate the cylinder diameters and stroke length assuming a diagram factor of 0.75 and mean piston speed as 180 m/min.*

Also find the fraction of stroke at which cut-off occurs in L.P. cylinder for approximately equal initial force on both the pistons.

Solution. Given : I.P. = 200 kW ; N = 150 r.p.m. ; p_1 = 10 bar ; p_b = 0.2 bar ; $v_1 = 0.45\,v_2$; $v_4/v_2 = 3.2$; $K = 0.75$; Mean piston speed = 180 m/min

We know that expansion ratio in H.P. cylinder,

$$r_H = \frac{v_2}{v_1} = \frac{v_2}{0.45\,v_2} = 2.22$$

and total expansion ratio, $R = \frac{v_4}{v_1} = \frac{v_4}{v_2} \times \frac{v_2}{v_1} = 3.2 \times 2.22 = 7.1$

Stroke length

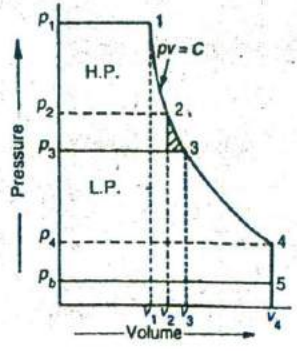
Let L = Stroke length in metres

We know that mean piston speed,

$$180 = 2\,L\,N = 2\,L \times 150 = 300\,L$$

or $$L = 180/300 = 0.6 \text{ m} \quad \textbf{Ans.}$$

Diameter of L.P. cylinder



18.17

Let D_L = Diameter of L. P. cylinder in metres.

$\therefore$ Area, $A = \frac{\pi}{4}(D_L)^2 = 0.7854\,(D_L)^2$

We know that actual mean effective pressure referred to L.P. cylinder,

$$p_a = K\left[\frac{p_1}{R}(1 + 2.3 \log R) - p_b\right]$$

$$= 0.75\left[\frac{10}{7.1}(1 + 2.3 \log 7.1) - 0.2\right] = 2.97 \text{ bar}$$

and indicated power (I.P.),

$$200 = \frac{200\,p_a\,L\,A\,N}{60} = \frac{200 \times 2.97 \times 0.6 \times 0.7854\,(D_L)^2\,150}{60} = 700\,(D_L)^2$$

$$\therefore \quad (D_L)^2 = 200/700 = 0.286 \text{ or } D_L = 0.534 \text{ m} \quad \textbf{Ans.}$$

Diameter of H.P. cylinder

Let D_H = Diameter of H.P. cylinder in metres.

We know that ratio of L.P. cylinder volume to H.P. cylinder volume,

$$3.2 = \frac{v_4}{v_2} = \frac{0.7854\,(D_L)^2\,L}{0.7854\,(D_H)^2\,L} = \frac{(D_L)^2}{(D_H)^2}$$

$$\therefore \quad D_H = \sqrt{\frac{(D_L)^2}{3.2}} = \sqrt{\frac{(0.534)^2}{3.2}} = 0.298 \text{ m} \quad \textbf{Ans.}$$

Fraction of stroke at which cut-off occurs in L.P. cylinder

First of all, let us find the pressure of steam at release point of H.P. cylinder (*i.e.* p_3). We know that for equal initial loads on both the pistons,

$$\frac{p_1 - p_3}{p_3 - p_b} = \frac{v_4}{v_2} = 3.2$$

$$\frac{10 - p_3}{p_3 - 0.2} = 3.2 \quad \text{or} \quad 10 - p_3 = 3.2\,p_3 - 3.2 \times 0.2$$

$$\therefore \quad p_3 = 2.53 \text{ bar}$$

We know that volume of H.P. cylinder,

$$v_2 = \frac{\pi}{4}(D_H)^2 L = \frac{\pi}{4}(0.298)^2\, 0.6 = 0.042 \text{ m}^3$$

$$\therefore \quad v_1 = 0.45 \times 0.042 = 0.019 \text{ m}^3 \qquad \ldots (\because v_1 = 0.45\, v_2)$$

and volume of L.P. cylinder,

$$v_4 = \frac{\pi}{4}(D_L)^2 L = \frac{\pi}{4}(0.534)^2\, 0.6 = 0.1344 \text{ m}^3$$

Since expansion of the steam follows the law $pv = C$, therefore

$$p_1 v_1 = p_3 v_3$$

or

$$v_3 = \frac{p_1 v_1}{p_3} = \frac{10 \times 0.019}{2.53} = 0.075 \text{ m}^3$$

$\therefore$ Cut-off in L.P. cylinder

$$= \frac{v_3}{v_4} = \frac{0.075}{0.1344} = 0.56 \textbf{ Ans.}$$

Note : The cut-off in L.P. cylinder may also be obtained as follows :

We know that $p_1 v_1 = p_3 v_3$

$$\therefore \quad \frac{v_3}{v_1} = \frac{p_1}{p_3} = \frac{10}{2.53} = 3.95$$

Now

$$\frac{v_3}{v_4} = \frac{v_3}{v_1} \times \frac{v_1}{v_2} \times \frac{v_2}{v_4} = 3.95 \times \frac{1}{2.22} \times \frac{1}{3.2} = 0.56 \textbf{ Ans.}$$

18.14. Three Cylinder Compound Steam Engines

In the previous articles, we have been discussing double cylinder, (or in other words double expansion) compound steam engines. In such engines, we have always been referring to H.P. and L.P. cylinders.

But sometimes, we use three cylinder compound steam engines instead of two cylinder compound steam engines. As the name indicates, a three cylinder (or triple expansion) compound steam engine consists of three cylinders namely H.P. (high pressure) cylinder, I.P. (intermediate pressure) cylinder and L.P. (low pressure) cylinder. In a triple expansion compound steam engine, the steam is first admitted into the H.P. cylinder. After expansion in the H.P. cylinder, it is admitted into the I.P. cylinder. Similarly, after further expansion in the I.P. cylinder, the steam is admitted into the L.P. cylinder. Again after further expansion in the L.P. cylinder, the steam is exhausted.

All the relations of double cylinder compound steam engines also hold good for triple cylinder compound steam engines. The following examples will illustrate the theory of triple cylinder compound steam engines.

Example 18.8. *A triple expansion engine is supplied with steam at 13 bar and the condenser pressure is 0.2 bar. The overall expansion ratio is 13. Neglecting clearance effects, assuming no pressure drop at release in the high pressure and intermediate pressure cylinders, and assuming hyperbolic expansion, determine the ratio of cylinder volumes. Take the high pressure cylinder volume as unity, in order that equal powers may be developed in the three cylinders.*

Solution. Given : $p_1 = 13$ bar ; $p_b = 0.2$ bar ; $R = v_4/v_1 = 13$; $v_2 = 1$

Let v_3 = Volume of I.P. cylinder, and

v_4 = Volume of L.P. cylinder.

We have discussed in Art 18.11 that work done in H.P. cylinder,

$$W_H = K \times 2.3\, p_1 v_1 \log r_H \qquad \ldots (i)$$

Similarly, work done in I.P. cylinder,

$$W_I = K \times 2.3\, p_2 v_2 \log r_I \qquad \ldots (ii)$$

and work done in L.P. cylinder,

$$W_L = K\,[\, p_3 v_3 (1 + 2.3 \log r_L) - p_b v_4] \qquad \ldots (iii)$$

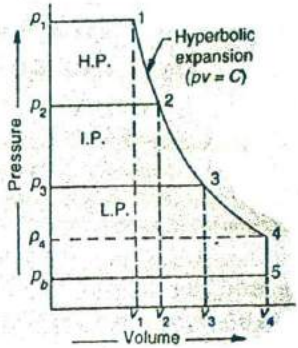


Fig. 18.18

Since power developed in each cylinder is the same, therefore

$$W_H = W_I = W_L$$

$$K \times 2.3\, p_1 v_1 \log r_H = K \times 2.3\, p_2 v_2 \log r_I$$

... (Equating W_H and W_I)

$$\therefore \qquad r_H = r_I \qquad \ldots (\because p_1 v_1 = p_2 v_2)$$

or

$$\frac{v_2}{v_1} = \frac{v_3}{v_2} \qquad \ldots \left(\because r_H = \frac{v_2}{v_1} \text{ and } r_I = \frac{v_3}{v_2} \right)$$

$$\therefore \qquad v_3 = \frac{v_2 \times v_2}{v_1} = \frac{1}{v_1} \qquad \ldots (\text{Given } v_2 = 1)$$

Again $\quad K \times 2.3\, p_1 v_1 \log r_H = K\,[\, p_3 v_3 (1 + 2.3 \log r_L) - p_b v_4] \qquad \ldots (\text{Equating } W_H = W_L)$

$$\therefore \qquad 2.3 \log r_H = \frac{p_3 v_3}{p_1 v_1}\left[1 + 2.3 \log\left(\frac{v_4}{v_3}\right)\right] - \frac{p_b v_4}{p_1 v_1} \qquad \ldots \left(\because r_L = \frac{v_4}{V_3} \right)$$

$$2.3 \log\left(\frac{v_2}{v_1}\right) = 1 + 2.3 \log\left(\frac{v_4}{v_3}\right) - \frac{p_b v_4}{p_1 v_1} \qquad \ldots (\because p_1 v_1 = p_2 v_2 = p_3 v_3)$$

$$2.3 \log\left(\frac{v_2}{v_1}\right) - 2.3 \log\left(\frac{v_4}{v_3}\right) = 1 - \frac{p_b}{p_1} \times \frac{v_4}{v_1} = 1 - \frac{0.2}{13} \times 13 = 0.8 \qquad \ldots (\because v_4/v_1 = 13)$$

$$\log\left(\frac{v_2}{v_1} \times \frac{v_3}{v_4}\right) = \frac{0.8}{2.3} = 0.3478$$

or

$$\frac{v_2}{v_1} \times \frac{v_3}{v_4} = 2.227 \qquad \ldots (\text{Taking antilog of } 0.3478)$$

Substituting the values of $v_2 = 1$, $v_3 = 1/v_1$ and $v_4 = 13\, v_1$, in the above equation,

$$\frac{1}{v_1} \times \frac{1}{v_1 \times 13\, v_1} = 2.227 \text{ or } \frac{1}{(v_1)^3} = 13 \times 2.227 = 28.95$$

$$\therefore \qquad \frac{1}{v_1} = 3.07 \text{ or } v_1 = 0.326 \text{ m}^3$$

We know that $v_3 = \frac{1}{v_1} = 3.07$

and $v_4 = 13\,v_1 = 13 \times 0.326 = 4.24$

$\therefore$ Ratio of cylinder volumes,

$$v_2 : v_3 : v_4 = 1 : 3.07 : 4.24 \text{ Ans.}$$

Example 18.9. *A triple expansion engine is required to develop 3000 kW I.P. at 100 r.p.m. The steam is supplied at 10.5 bar and back pressure is 0.07 bar. The piston speed is 180 m/min, and diagram factor is 0.6. If the ratio of cylinder volumes is 1 : 3 : 7.5 and cut-off in H.P. cylinder is 0.7, determine the size of cylinders.*

Solution. Given : I.P. = 3000 kW ; N = 100 r.p.m. ; p_1 = 10.5 bar ; p_b = 0.07 bar ; Piston speed = 180 m/min ; $K = 0.6$; $v_4 / v_2 = 3$; $v_6 / v_2 = 7.5$; $v_1 / v_2 = 0.7$

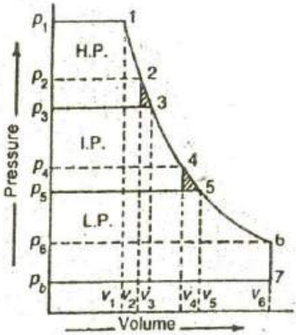


Fig. 18.19

We know that total expansion ratio,

$$R = \frac{v_6}{v_1} = \frac{v_6}{v_2} \times \frac{v_2}{v_1}$$

$$= \frac{7.5}{0.7} = 10.7$$

and actual m.e.p. referred to L.P. cylinder,

$$p_a = K\left[\frac{p_1}{R}(1 + 2.3 \log R) - p_b\right]$$

$$= 0.6\left[\frac{10.5}{10.7}(1 + 2.3 \log 10.7) - 0.07\right] = 1.94 \text{ bar}$$

Diameter of L.P. cylinder

Let D_L = Diameter of L.P. cylinder in metres.

$\therefore$ Area, $A = \frac{\pi}{4}(D_L)^2 = 0.7854\,(D_L)^2 \text{ m}^2$

We know that indicated power (I.P.),

$$3000 = \frac{200\, p_a\, L\, A\, N}{60} = \frac{100\, p_a\, A \times 2L\, N}{60}$$

$$= \frac{100 \times 1.94 \times 0.7854\,(D_L)^2\, 180}{60} = 457\,(D_L)^2 \quad \ldots (\because\ 2\,L\,N = 180 \text{ m/min})$$

$\therefore$ $(D_L)^2 = 5.56$ or $D_L = 2.56$ m **Ans.**

Diameter of H.P. cylinder

Let D_H = Diameter of H.P. cylinder in metres.

We know that volume of L.P. cylinder,

$$v_6 = \frac{\pi}{4}(D_L)^2 L = 0.7854\,(2.56)^2 L = 5.147\,L$$

and volume of H.P. cylinder,

$$v_2 = \frac{\pi}{4}(D_H)^2 L = 0.7854\,(D_H)^2 L$$

We are given that $\frac{v_6}{v_2} = 7.5$ or $\frac{5.147\,L}{0.7854\,(D_H)^2\,L} = 7.5$

$\therefore$ $(D_H)^2 = 0.8738$ or $D_H = 0.935$ m **Ans.**

Diameter of I.P. cylinder

Let D_I = Diameter of I.P. cylinder in metres,

We know that volume of I.P. cylinder,

$$v_4 = \frac{\pi}{4}(D_I)^2 L = 0.7854 (D_I)^2 L$$

We are given that $\frac{v_4}{v_2} = 3$ or $\frac{0.7854\,(D_I)^2\,L}{0.7854\,(0.935)^2\,L} = 3$

$\therefore$ $(D_I)^2 = 2.623$ or $D_I = 1.62$ m **Ans.**

Example 18.10. *In a triple cylinder compound steam engine, the steam is supplied at 12.5 bar and the back pressure is 0.2 bar. All the cylinders have restricted expansions with volumes in the ratio of 1 : 2.5 : 6 and overall expansion ratio is 15. The cut-off in I.P. cylinder is 50% of the stroke and in L.P. cylinder is 65% of the stroke. Assuming hyperbolic expansion and neglecting clearance, find : 1. the theoretical mean effective pressure of the three cylinders referred to L.P. cylinder ; 2. the theoretical mean effective pressure referred to L.P. cylinder when there is a complete expansion in H.P. and I.P. cylinders. The overall expansion ratio is 12.5 ; and 3. the percentage loss of power due to restricted expansion in H.P. and I.P. cylinders.*

Solution. Given : $p_1 = 12.5$ bar ; $p_b = 0.2$ bar ; $v_4 / v_2 = 2.5$; $v_6 / v_2 = 6$; $R = v_6 / v_2 = 15$; $v_3 / v_4 = 50\%$ of stroke or $v_3 = 0.5\,v_4$; $v_5 / v_6 = 65\%$ of stroke or $v_5 = 0.65\,v_6$

We know that cut-off in H.P. cylinder,

$$\frac{v_1}{v_2} = \frac{v_1}{v_6} \times \frac{v_6}{v_2} = \frac{1}{15} \times 6 = 0.4$$

1. *Theoretical mean effective pressure of the three cylinders referred to L.P. cylinder*

Let p_m = Theoretical mean effective pressure.

From the geometry of the cylinders, we find that expansion ratio in H.P. cylinder,

$$r_H = \frac{v_2}{v_1} = \frac{1}{0.4} = 2.5$$

Similarly, $$r_I = \frac{v_4}{v_3} = \frac{1}{0.5} = 2$$

and $$r_L = \frac{v_6}{v_5} = \frac{1}{0.65} = 1.54$$

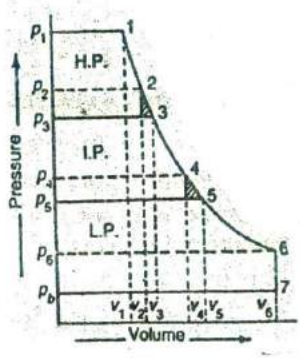


Fig. 18.20

Since expansion of steam follows the law $p\,v = C$, therefore

$$p_1 v_1 = p_3 v_3 = p_5 v_5$$

$\therefore$
$$p_3 = p_1 \times \frac{v_1}{v_3} = p_1 \times \frac{v_1}{v_2} \times \frac{v_2}{v_4} \times \frac{v_4}{v_3}$$
$$= 12.5 \times 0.4 \times \frac{1}{2.5} \times 2 = 4 \text{ bar}$$

Similarly
$$p_5 = p_3 \times \frac{v_3}{v_5} = p_3 \times \frac{v_3}{v_4} \times \frac{v_4}{v_6} \times \frac{v_6}{v_5}$$
$$= 4 \times 0.5 \times \frac{2.5}{6} \times 1.54 = 1.28 \text{ bar} \qquad \ldots \left(\frac{v_4}{v_6} = \frac{v_4}{v_2} \times \frac{v_2}{v_6} = 2.5 \times \frac{1}{6} \right)$$

We know that theoretical mean effective pressure of H.P. cylinder,
$$p_{mH} = \frac{p_1}{r_H}(1 + 2.3 \log r_H) - p_3$$
$$= \frac{12.5}{2.5}(1 + 2.3 \log 2.5) - 4 = 5.58 \text{ bar}$$

and theoretical mean effective pressure of H.P. cylinder referred to L.P. cylinder
$$= p_{mH} \times \frac{v_2}{v_6} = 5.58 \times \frac{1}{6} = 0.93 \text{ bar} \qquad \ldots \left(\because \frac{v_2}{v_6} = \frac{1}{6} \right)$$

Similarly, theoretical mean effective pressure of I.P. cylinder,
$$p_{mI} = \frac{p_3}{r_I}(1 + 2.3 \log r_I) - p_5$$
$$= \frac{4}{2}(1 + 2.3 \log 2) - 1.28 = 2.1 \text{ bar}$$

and theoretical mean effective pressure of I.P. cylinder referred to L.P. cylinder
$$= p_{mI} \times \frac{v_4}{v_6} = 2.1 \times \frac{2.5}{6} = 0.88 \text{ bar} \qquad \ldots \left(\because \frac{v_4}{v_6} = \frac{2.5}{6} \right)$$

Similarly, theoretical mean effective pressure of L.P. cylinder,
$$p_{mL} = \frac{p_5}{r_L}(1 + 2.3 \log r_L) - p_b$$
$$p_{mL} = \frac{1.28}{1.54}(1 + 2.3 \log 1.54) - 0.2 = 0.99 \text{ bar}$$

$\therefore$ Total theoretical mean effective pressure referred to L.P. cylinder,
$$p_{m1} = 0.93 + 0.88 + 0.99 = 2.80 \text{ bar } \textbf{Ans.} \qquad \ldots (i)$$

2. *Theoretical mean effective pressure referred to L.P. cylinder for complete expansion*

We know that theoretical mean effective pressure referred to L.P. cylinder when there is complete expansion in H.P. and I.P. cylinders (or in other words there is a complete expansion from v_1 to v_6),
$$p_{m2} = \frac{p_1}{R}(1 + 2.3 \log R) - p_b$$
$$= \frac{12.5}{12.5}(1 + 2.3 \log 12.5) - 0.2 = 3.3 \text{ bar } \textbf{Ans.} \qquad \ldots (\text{Given } R = 12.5)$$

3. *Percentage loss of power due to restricted expansion in H.P. and I.P. cylinders*

We know that percentage loss of power due to restricted expansion in H.P. and I.P. cylinders

$$= \frac{p_{m2} - p_{m1}}{p_{m2}} \times 100 = \frac{3.3 - 2.8}{3.3} = 0.1515 \text{ or } 15.15\ \% \text{ Ans.}$$

EXERCISES

1. A horizontal, double acting compound steam engine is to develop 360 kW under the following conditions :

Pressure at steam chest = 12 bar ; Back pressure = 0.2 bar ; Average piston speed = 180 m/min ; R.P.M. = 180 ; Ratio of L.P. to H.P. cylinder areas = 4 ; Total expansion ratio = 15 ; Overall diagram factor = 0.8.

Determine the main dimensions of the engine. Assume complete expansion in H.P. cylinder and hyperbolic expansion. Neglect clearance. [**Ans.** $L = 0.5$ m ; $D_H = 0.415$ m ; $D_L = 0.831$ m]

2. A compound steam engine in required to develop 110 kW at 150 r.p.m. The steam is supplied at 7.5 bar and back pressure is 0.2 bar. Assuming complete hyperbolic expansion ratio of 15, calculate the cylinder diameters, so that power developed by each cylinder is equal. Take stroke length equal to diameter of L.P. cylinder and diagram factor as 0.7. Neglect clearance. [**Ans.** $D_H = 0.367$ m ; $D_L = 0.623$ m]

3. The following data refer to a double acting compound steam engine :

Indicated power = 300 kW ; Speed = 120 r.p.m ; Stroke = 900 mm ; Initial pressure = 12 bar ; Back pressure = 0.14 bar ; Expansion ratio = 15 ; Diagram factor = 0.85.

Assuming complete expansion in H.P. cylinder, equal initial load, hyperbolic expansion and neglecting clearance, determine : 1. the receiver pressure for L.P. cylinder ; and 2. the diameters of H.P. and L.P. cylinders. [**Ans.** 2.7 bar ; $D_L = 0.669$ m ; $D_H = 0.364$ m]

4. A double acting, two cylinder compound steam engine develops 185 kW at 120 r.p.m., when supplied with steam at a pressure of 7.5 bar. The condenser pressure is 0.2 bar ; allowable piston speed is 150 m/s; the ratio of cylinder volumes is 3.5; diagram factor is 0.8; and cut-off in H.P. cylinder is at 0.4 stroke.

Assuming hyperbolic expansion and neglecting clearance, determine the cylinder dimensions.

[**Ans.** $D_L = 0.684$ m ; $D_H = 0.366$ m]

5. A double acting compound engine working between pressures 10.5 bar and 0.3 bar has a total ratio of expansion as 10. The cut-off in the H.P. cylinder takes place at 0.4 of stroke. Assuming hyperbolic expansion, find : 1. the cut-off in the L.P. cylinder if the initial loads on two pistons are equal, and 2. the ratio of workdone in the two cylinders. [**Ans.** 0.45 of stroke ; 1.117]

6. A double acting compound steam engine is required to develop 400 kW at 125 r.p.m. The steam is supplied at 13 bar with cut-off at 0.5 stroke in the H.P. cylinder. The condenser pressure is 0.14 bar, the mean piston speed 235 metres/min; overall expansion ratio 14; overall diagram factor 0.7; the stroke of H.P. and L.P. cylinders are same. Neglecting clearance and assuming hyperbolic expansion, calculate the L.P. and H.P. cylinder diameters and the stroke. Also find the cut-off in L.P. cylinder for equal load on both the pistons.

[**Ans.** $L = 0.94$ m ; $D_L = 0.758$ m ; $D_H = 0.286$ m ; 0.53]

7. A double acting compound steam engine has cylinder diameter H.P. = 300 mm ; L.P. = 600 mm and stroke of both cylinders is 400 mm. The engine develops 32 kW while running at 160 r.p.m. The steam is supplied at 13.8 bar and back pressure is 0.28 bar. If cut-off in the H.P. cylinder is at one-third stroke, find the actual and theoretical mean effective pressure referred to L.P. cylinder and hence find the overall diagram factor. Neglect clearance. [**Ans.** 2.64 bar ; 3.73 bar ; 0.82]

8. A two cylinder compound steam engine receives steam at a pressure of 8 bar and discharges at 0.2 bar. The cylinder volume ratio is 4. The cut-off in H.P. and L.P. cylinder is 0.4 and 0.5 of the stroke respectively. Determine : 1. the actual mean effective pressure in H.P. and L.P. cylinders, if the diagram factor for the H.P. and L.P. cylinders is 0.75 and 0.65 respectively, and 2. the ratio of workdone in H.P. and L.P. cylinders.

[**Ans.** 3.4 bar, 0.75 bar ; 0.985]

9. A compound steam engine develops 330 kW indicated power when taking steam at 10.5 bar and exhausting at 0.21 bar. The rotational speed is 180 r.p.m. and the piston speed is 183 m/min. The cut-off in H.P. cylinder is to be 0.4 of the stroke; cut-off in L.P. cylinder is 0.5 of stroke, and the cylinder volume ratio is

4. Assuming hyperbolic expansion and neglecting clearance, determine suitable cylinder dimensions. Take diagram factor 0.84. Find also the L.P. receiver pressure and compare the initial loads on the piston.

[**Ans.** $L = 0.508$ m ; $D_L = 0.71$ m, $D_H = 0.355$ m ; 2.1 bar ; 1.11]

10. A double acting compound steam engine working between pressures of 10 bar and 0.21 bar is to develop 150 kW indicated power at 90 r.p.m. The H.P. cylinder cut-off is at 0.35 of stroke and ratio of L.P. cylinder to H.P. cylinder volume is 3.2. Determine the cylinder diameters and stroke assuming a diagram factor of 0.7 and mean piston speed of 135 metres/min.

Find also the fraction of stroke at which the cut-off should occur in L.P. cylinder for equal initial load on both the pistons. [**Ans.** $L = 0.75$ m ; $D_L = 0.603$ m ; $D_H = 0.337$ m ; 0.43 of stroke]

11. A double acting compound steam engine is developing 45 kW at 200 r.p.m. The engine is supplied with steam at 10 bar. The back pressure is 650 mm of Hg and the barometer reads 750 mm of Hg. The cut-off in H.P. cylinder takes place at 50% of the stroke and the overall expansion ratio is 12. The ratio of stroke to L.P. cylinder diameter is 1.2 and the overall diagram factor is 0.7. Find the cylinder diameters, stroke and L.P. cut-off for equal initial piston loads. [**Ans.** $D_L = 0.333$ m, $D_H = 0.136$ m, $L = 0.4$ m ; 0.54 of stroke]

12. A double acting compound steam engine is supplied with steam at 8 bar and exhausts into a condenser at a pressure of 0.3 bar. The ratio of cylinder volumes is 4 and cut-off in H.P. cylinder takes place at 50% of the stroke. The length of stroke of both the cylinders is same. Assuming hyperbolic expansion and neglecting clearance, find :

1. the cut-off in L.P. cylinder for equal initial load on the pistons,
2. the mean effective pressure of each cylinder referred to L.P. cylinder, and
3. the percentage loss of work due to incomplete expansion in H.P. cylinder.

[**Ans.** 0.543 ; 1.23 bar, 1.31 bar ; 8.63 %]

13. A triple expansion engine is to develop 2200 kW indicated power at a piston speed of 400 m/min. The volumes of H.P., I.P. and L.P. cylinders are in the ratio of 1 : 2.5 : 7.5. The initial pressure of steam is 17 bar and the exhaust pressure is 0.2 bar. The cut-off in H.P. cylinder is at 0.5 of the stroke and overall diagram factor is 0.65.

Assuming complete expansion in H.P. and I.P. cylinders and hyperbolic expansion, determine the diameters of the cylinders. Neglect clearance. [**Ans.** $D_L = 1.272$ m ; $D_I = 0.734$ m ; $D_H = 0.464$ m]

14. In a triple expansion steam engine, the steam is supplied at 15 bar and back pressure is 0.14 bar. All the cylinders have restricted expansion. The cylinder volumes are in the ratio of 1 : 2.5 : 7.5 and overall expansion ratio is 15. If the cut-off in I.P. cylinder occurs at 50% of stroke and cut-off in L.P. cylinder is 60% of stroke, calculate the total mean effective pressure of all the cylinders referred to L.P. cylinder. Assume hyperbolic expansion and neglect clearance [**Ans.** 3.402 bar]

15. A triple expansion engine is required to develop 2950 kW at a piston speed of 210 m/min, under the following conditions :

Pressure in the steam chest = 15 bar ; Back pressure = 0.15 bar ; Cylinder volume ratios = 1 : 2.4 : 7.2; Total expansion ratio = 18; Overall diagram factor = 0.62.

Assuming equal initial load on each piston, find : 1. cylinder diameters ; 2. receiver pressures ; and 3. cut-off points in each cylinder.

Neglect clearance and assume hyperbolic expansion.

[**Ans.** $D_L = 2.367$ m, $D_I = 1.367$ m, $D_H = 0.882$ m ; 5.46 bar, 1.475 bar ; 0.4, 0.458, 0.565]

QUESTIONS

1. Explain what is meant by compounding of steam engines.
2. State the advantages of compounding a steam engine.
3. Explain, with the help of line diagrams, the various methods of compounding a steam engine.
4. What are the main factors in deciding the sizes of the cylinder in a compound steam engine ?

OBJECTIVE TYPE QUESTIONS

1. In a compound steam engine, the last stage of expansion is carried out in a
   (*a*) low pressure cylinder (*b*) high pressure cylinder
   (*c*) Intermediate pressure cylinder (*d*) none of these
2. By compounding the expansion of steam in two or more cylinders, the ratio of expansion
   (*a*) increases (*b*) decreases (*c*) does not change
3. By compounding the expansion of steam in two or more cylinders, the length of stroke
   (*a*) increases (*b*) decreases (*c*) does not change
4. A compound steam engine requires ..... flywheel than simple steam engine.
   (*a*) lighter (*b*) heavier
5. In a compound steam engine, the diameter of high pressure cylinder is ..... the low pressure cylinder.
   (*a*) equal to (*b*) less than (*c*) greater than
6. In a Tandem type compound steam engine, the high pressure and low pressure cylinders
   (*a*) have common piston rod (*b*) are set at 90°
   (*c*) have separate piston rods (*d*) are set in V-arrangement
7. The high pressure and low pressure cylinders in a Woolf type compound steam engine are regarded as having cranks
   (*a*) 180° to each other (*b*) 90° to each other
   (*c*) 0° to each other (*d*) none of these
8. In a receiver type compound steam engine, the high pressure and low pressure cylinders
   (*a*) have common piston rod (*b*) are set at 90°
   (*c*) have separate piston rods (*d*) are set in V-arrangement
9. A compound steam engine in which the piston rods of high and low pressure cylinders are attached to two different cranks set at 180° to each other, is called
   (*a*) receiver type compound engine (*b*) Woolf type compound engine
   (*c*) Tandem type compound engine (*d*) both (*a*) and (*b*)
10. The high pressure and low pressure cylinders in a receiver type compound steam engine are regarded as having cranks
   (*a*) 180° to each other (*b*) 90° to each other
   (*c*) 0° to each other (*d*) none of these

ANSWERS

1. (*a*)	2. (*b*)	3. (*b*)	4. (*a*)	5. (*b*)
6. (*a*)	7. (*a*)	8. (*c*)	9. (*b*)	10. (*b*)

19

Performance of Steam Engines

1. Introduction. 2. Efficiencies of a Steam Engine. 3. Mass of Steam in Engine Cylinder. 4. Missing Quantity. 5. Methods of Reducing Missing Quantity or Cylinder Condensation. 6. Heat Balance Sheet. 7. Governing of Steam Engines. 8. Governing of Simple Steam Engines. 9. Throttle Governing of Simple Steam Engines. 10. Cut-off Governing of Simple Steam Engines. 11. Governing of Compound Steam Engines. 12. Throttle Governing of Compound Steam Engines. 13. Cut-off Governing of Compound Steam Engines. 14. Steam Consumption (Willian's Law).

19.1. Introduction

In the last two chapters, we have discussed simple and compound steam engines. In these chapters, we have discussed power generated and cylinder dimensions of the steam engines. But in this chapter, we shall discuss their performance *i.e.* efficiencies, governing etc.

19.2. Efficiencies of a Steam Engine

The efficiency of an engine is defined as the ratio of work done to the energy supplied to an engine. The following efficiencies of a steam engine are important from the subject point of view :

1. *Mechanical efficiency.* It is the ratio of the *brake power (B.P.) to the indicated power (I.P.). Mathematically, mechanical efficiency,

$$\eta_m = \frac{\text{B.P.}}{\text{I.P.}}$$

It may be observed that the mechanical efficiency is always less than unity (*i.e.* 100%) because some power is lost in overcoming the engine friction. In other words, the indicated power is always greater than brake power. This power which is lost in overcoming the engine friction is known as *frictional power*. Therefore, frictional power,

$$\text{F.P.} = \text{I.P.} - \text{B.P.}$$

2. *Overall efficiency.* It is the ratio of the work obtained at the crank shaft in a given time to the energy supplied by fuel during the same time.

Let m_f = Mass of fuel burnt in kg per hour, and

C = Calorific value of fuel in kJ/kg of fuel.

$\therefore$ Energy supplied by fuel/min

$$= \frac{m_f \times C}{60} \text{ kJ}$$

and work obtained at the crank shaft/min

$$= \text{B.P.} \times 60 \text{ kJ} \qquad \ldots (\because \text{B.P. is in kW and } 1 \text{ kW} = 1 \text{ kJ/s})$$

* The indicated power (I.P.) and brake power (B.P.) has already been discussed in chapter 17.

∴ Overall efficiency,

$$\eta_0 = \frac{\text{B.P.} \times 60 \times 60}{m_f \times C} = \frac{\text{B.P.} \times 3600}{m_f \times C}$$

3. *Indicated thermal efficiency.* It is the ratio of heat equivalent of indicated power to the energy in the steam supplied per minute.

Let m_s = Mass of steam used in kg/min,

h_1 = Enthalpy or total heat of steam supplied at admission pressure p_1 in kJ/kg (from steam tables), and

h_{fb} = Enthalpy or sensible heat of feed water at back pressure p_b in kJ/kg (from steam tables).

∴ Energy in steam supplied/min

$$= m_s (h_1 - h_{fb}) \text{ kJ/min}$$

and heat equivalent to I.P. $= \text{I.P.} \times 60$ kJ/min

∴ Indicated thermal efficiency

$$= \frac{\text{I.P.} \times 60}{m_s (h_1 - h_{fb})}$$

Note : The mass of steam used in kg per indicated power or brake nower per hour (*i.e.* in kg/kWh) is known as *specific steam consumption.*

We know that mass of steam used per hour

$$= m_s \times 60 \text{ kg/h}$$

∴ Specific steam consumption

$$= \frac{m_s \times 60}{\text{I.P. or B.P.}} \text{ kg/kWh}$$

4. *Brake thermal efficiency.* It is the ratio of the heat equivalent of brake power to the energy in the steam supplied per minute. Mathematically,

Brake thermal efficiency

$$= \frac{\text{B.P.} \times 60}{m_s (h_1 - h_{fb})}$$

Note : Whenever thermal efficiency is mentioned without qualifying the name, *i.e.* "indicated" or "brake", the indicated thermal efficiency should be calculated.

5. *Relative efficiency.* The relative efficiency is also known as *efficiency ratio.* It is the ratio of thermal efficiency to the Rankine efficiency. Mathematically, relative efficiency,

$$\eta = \frac{\text{Thermal efficiency}}{\text{Rankine efficiency}}$$

Example 19.1. *During a test on a single acting non-condensing, single cylinder steam engine, the following observations were recorded :*

Bore = 225 mm ; Stroke = 600 mm ; Speed = 100 r.p.m. ; Effective brake diameter = 2.75 m ; Net load on the brake = 1650 N ; Area of indicator diagram = 2500 mm² ; Length of indicator diagram = 100 mm ; Spring strength = 530 bar/m.

Determine: 1. Indicated power ; 2. Brake power ; and 3. Mechanical efficiency.

Solution. Given : $D = 225$ mm $= 0.225$ m ; $L = 600$ mm $= 0.6$ m ; $N = 100$ r.p.m. ; $D_1 = 2.75$ m ; $(W - S) = 1650$ N ; $a_1 = 2500 \text{ mm}^2 = 2500 \times 10^{-6} \text{ m}^2$; $b = 100$ mm $= 0.1$ m ; $s = 530$ bar/m

We know that actual mean effective pressure,

$$p_a = \frac{\text{Area of indicator diagram} \times \text{Spring strength}}{\text{Length of indicator diagram}}$$

$$= \frac{a_1 \times s}{b} = \frac{2500 \times 10^{-6} \times 530}{0.1} = 13.25 \text{ bar}$$

Area of cylinder, $A = \frac{\pi}{4} \times D^2 = \frac{\pi}{4} (0.225)^2 = 0.04 \text{ m}^2$

1. *Indicated power*

We know that indicated power,

$$\text{I.P.} = \frac{100\, p_a\, L\, A\, N}{60} = \frac{100 \times 13.25 \times 0.6 \times 0.04 \times 100}{60} = 53 \text{ kW} \quad \textbf{Ans.}$$

... ($\because$ Engine is single acting)

2. *Brake Power*

We know that brake power,

$$\text{B.P.} = \frac{(W - S)\, \pi\, D_1\, N}{60} = \frac{1650 \times \pi \times 2.75 \times 100}{60} = 23\,760 \text{ W}$$

$$= 23.76 \text{ kW} \quad \textbf{Ans.}$$

3. *Mechanical efficiency,*

We know that mechanical efficiency,

$$\eta_m = \frac{\text{B.P.}}{\text{I.P.}} = \frac{23.76}{53} = 0.448 \text{ or } 44.8\,\% \quad \textbf{Ans.}$$

Example 19.2. *Estimate the brake power of simple steam engine having 250 mm piston diameter, and 40 mm piston rod diameter with 250 mm stroke length operating at 300 r.p.m. The initial and back pressure of steam is 8.5 bar and 1.2 bar respectively. Assume 90% mechanical efficiency, cut-off at 25% of the forward stroke and 0.73 diagram factor. Neglect clearance and compression.*

Solution. Given : $D = 250$ mm $= 0.25$ m ; $d = 40$ mm $= 0.04$ m ; $L = 250$ mm $= 0.25$ m ; $N = 300$ r.p.m. ; $p_1 = 8.5$ bar ; $p_b = 1.2$ bar ; $\eta_m = 90\,\% = 0.9$; $v_1 = 0.25\, v_2$; $K = 0.73$

We know that expansion ratio,

$$r = \frac{v_2}{v_1} = \frac{v_2}{0.25\, v_2} = 4$$

and actual mean effective pressure,

$$p_a = K\left[\frac{p_1}{r}(1 + 2.3 \log r) - p_b\right] = 0.73\left[\frac{8.5}{4}(1 + 2.3 \log 4) - 1.2\right]$$

$$= 2.82 \text{ bar}$$

Area of piston, $A = \frac{\pi}{4} \times D^2 = \frac{\pi}{4} (0.25)^2 = 0.0491 \text{ m}^2$

and area of piston rod, $a = \frac{\pi}{4} \times d^2 = \frac{\pi}{4} (0.04)^2 = 0.001\,26 \text{ m}^2$

∴ Indicated power,

$$\text{I.P.} = \frac{100\,p_a\,L\,A\,N}{60} + \frac{100\,p_a\,L\,(A-a)\,N}{60} = \frac{100\,p_a\,L\,(2A-a)\,N}{60}$$

$$= \frac{100 \times 2.82 \times 0.25\,(2 \times 0.0491 - 0.001\,26)\,300}{60} = 34.2\text{ kW}$$

We know that mechanical efficiency (η_m),

$$0.9 = \frac{\text{B.P}}{\text{I.P.}} \quad \text{or} \quad \text{B.P.} = 0.9 \times \text{I.P.} = 0.9 \times 34.2 = 30.78\text{ kW} \quad \textbf{Ans.}$$

Example 19.3. *A double acting steam engine with cylinder 150 mm diameter and 200 mm stroke is to develop 18 kW at 300 r.p.m. with cut-off at 20% of the stroke. The back pressure is 0.3 bar. Determine the admission pressure if diagram factor is 0.7. Also calculate the indicated thermal efficiency of the engine if it receives 220 kg of dry steam per hour. Neglect clearance.*

Solution. Given : $D = 150$ mm $= 0.15$ m ; $L = 200$ mm $= 0.2$ m ; I.P. $= 18$ kW ; $N = 300$ r.p.m. ; $v_2 = 0.2\,v_3$; $p_b = 0.3$ bar ; $K = 0.7$; $m_s = 220$ kg/h $= 3.67$ kg/min

Admission pressure

Let p_1 = Admission pressure in bar, and

p_a = Actual mean effective pressure in bar.

We know that area of the cylinder,

$$A = \frac{\pi}{4} \times D^2 = \frac{\pi}{4}(0.15)^2 = 0.0177\text{ m}^2$$

and indicated power (I.P.),

$$18 = \frac{200\,p_a\,L\,A\,N}{60} = \frac{200 \times p_a \times 0.2 \times 0.0177 \times 300}{60} = 3.54\,p_a$$

$$\therefore \quad p_a = 5.085\text{ bar}$$

Expansion ratio, $r = v_3 / v_2 = v_3 / 0.2\,v_3 = 5$

We also know that actual mean effective pressure (p_a),

$$5.085 = K\left[\frac{p_1}{r}(1 + 2.3\log r) - p_b\right] = 0.7\left[\frac{p_1}{5}(1 + 2.3\log 5) - 0.3\right]$$

$$= 0.7\,(0.52\,p_1 - 0.3) = 0.364\,p_1 - 0.21$$

$$\therefore \quad p_1 = \frac{5.085 + 0.21}{0.364} = 14.5\text{ bar} \quad \textbf{Ans.}$$

Indicated thermal efficiency

From steam tables, corresponding to a pressure of 14.5 bar, we find that

$$h_1 = h_g = 2789\text{ kJ/kg} \qquad \text{. . . (For dry steam)}$$

and corresponding to a pressure of 0.3 bar,

$$h_{fb} = 289.3\text{ kJ/kg}$$

We know that indicated thermal efficiency.

$$= \frac{\text{I.P.} \times 60}{m_s\,(h_1 - h_{fb})} = \frac{18 \times 60}{3.67\,(2789 - 289.3)} = 0.118 \quad \text{or} \quad 11.8\% \quad \textbf{Ans.}$$

Example 19.4. *A double acting single cylinder steam engine runs at 250 r.p.m. and develops 30 kW. The pressure limits of operation are 10 bar and 1 bar. The cut-off is at 40% of the stroke. The stroke/bore ratio is 1.25 and the diagram factor is 0.75. Assume dry saturated steam at inlet, hyperbolic expansion and negligible effect of piston rod.*

Find : 1. the mean effective pressure ; 2. the cylinder dimensions, and 3. the indicated thermal efficiency.

Solution. Given : $N = 250$ r.p.m. ; I.P.= 30 kW ; $p_1 = 10$ bar ; $p_b = 1$ bar ; $v_2 = 0.4\, v_3$; $L/D = 1.25$; $K = 0.75$

1. *Mean effective pressure*

We know that expansion ratio,

$$r = v_3 / v_2 = v_3 / 0.4\, v_3 = 2.5$$

and actual mean effective pressure,

$$p_a = K\left[\frac{p_1}{r}(1 + 2.3 \log r) - p_b\right]$$

$$= 0.75\left[\frac{10}{2.5}(1 + 2.3 \log 2.5) - 1\right] = 5 \text{ bar} \quad \textbf{Ans.}$$

2. *Cylinder dimensions*

Let D = Diameter of the cylinder in metres, and

L = Length of the stroke in metres = $1.25\, D$ . . . (Given)

Area of the cylinder, $A = \frac{\pi}{4} \times D^2 = 0.7854\, D^2 \text{ m}^2$

We know that indicated power (I.P.),

$$30 = \frac{200\, p_a\, L\, A\, N}{60} = \frac{200 \times 5 \times 1.25\, D \times 0.7854\, D^2 \times 250}{60}$$

$$= 4090\, D^3$$

$\therefore$ $D^3 = 0.0733$ or $D = 0.194$ m **Ans.**

and $L = 1.25\, D = 1.25 \times 0.194 = 0.2425$ m **Ans.**

3. *Indicated thermal efficiency*

First of all, let us find the mass of steam (m_s) used per minute.

From steam tables, corresponding to a pressure of 10 bar, we find that for dry saturated steam,

$$v_g = 0.1943 \text{ m}^3\text{/kg, and } h_1 = 2776.2 \text{ kJ/kg}$$

and corresponding to a pressure of 1 bar,

$$h_{fb} = 417.5 \text{ kJ/kg}$$

We know that stroke volume,

$$v_3 = \frac{\pi}{4} \times D^2 \times L = \frac{\pi}{4}(0.194)^2\, 0.2425 = 0.007\,17 \text{ m}^3$$

and volume of steam in the cylinder at the point of cut-off,

$$v_2 = 0.4\, v_3 = 0.4 \times 0.007\,17 = 0.002\,87 \text{ m}^3$$

We know that mass of steam used per stroke

$$= \frac{v_2}{v_g} = \frac{0.002\,87}{0.1943} = 0.0147 \text{ kg}$$

and mass of steam used per minute,

$$m_s = 0.0147 \times 2\,N = 0.0147 \times 2 \times 250 = 7.35 \text{ kg/min}$$

∴ Indicated thermal efficiency

$$= \frac{\text{I.P.} \times 60}{m_s\,(h_1 - h_{fb})} = \frac{30 \times 60}{7.35\,(2776.2 - 417.5)} = 0.104 \quad \text{or} \quad 10.4\ \% \ \textbf{Ans.}$$

Example 19.5. *The following data were obtained during test on double acting steam engine :*

Indicated mean effective pressure = 2.5 bar ; R.P.M. = 104 ; Bore = 250 mm ; Stroke = 300 mm ; Net brake load = 1150 N ; Effective brake drum diameter = 1.65 m.

The steam is supplied at 7 bar and is dry and saturated. The condenser pressure = 0.07 bar ; condenser temperature = 22° C and condensate quantity = 3.3 kg/min.

Determine : 1. indicated power ; 2. brake power ; 3. mechanical efficiency ; and 4. brake thermal efficiency.

Solution. Given : $p_a = 2.5$ bar ; $N = 104$ r.p.m. ; $D = 250$ mm $= 0.25$ m ; $L = 300$ mm $= 0.3$ m ; $(W - S) = 1150$ N ; $D_1 = 1.65$ m ; $p_1 = 7$ bar ; $p_b = 0.07$ bar ; $t = 22°$ C ; $m_s = 3.3$ kg/min

1. *Indicated power*

We know that area of piston,

$$A = \frac{\pi}{4} \times D^2 = \frac{\pi}{4}(0.25)^2 = 0.0491 \text{ m}^2$$

∴ Indicated power,

$$\text{I.P.} = \frac{200\,p_a\,L\,A\,N}{60} = \frac{200 \times 2.5 \times 0.3 \times 0.0491 \times 104}{60} = 12.8 \text{ kW} \ \textbf{Ans.}$$

2. *Brake power*

We know that brake power,

$$\text{B.P.} = \frac{(W - S)\,\pi\,D_1\,N}{60} = \frac{1150 \times \pi \times 1.65 \times 104}{60} = 10.33 \times 10^3 \text{ W}$$

$$= 10.33 \text{ kW} \ \textbf{Ans.}$$

3. *Mechanical efficiency*

We know that mechanical efficiency,

$$\eta_m = \frac{\text{B.P.}}{\text{I.P.}} = \frac{10.33}{12.8} = 0.807 \quad \text{or} \quad 80.7\ \% \ \textbf{Ans.}$$

4. *Brake thermal efficiency*

From steam tables, corresponding to a pressure of 7 bar, we find that

$$h_1 = h_g = 2762 \text{ kJ/kg} \qquad \text{... (For dry saturated steam)}$$

and corresponding to a condenser pressure of 0.07 bar,

$$h_{fb} = 163.4 \text{ kJ/kg}$$

We know that brake thermal efficiency

$$= \frac{\text{B.P.} \times 60}{m_s (h_1 - h_{fb})} = \frac{10.33 \times 60}{3.3\,(2762 - 163.4)} = 0.0723 \text{ or } 7.23\% \text{ Ans.}$$

Example 19.6. *Steam is supplied at a pressure of 12 bar and 0.95 dry to a simple double acting non-condensing steam engine working with the following data :*

B.P. = 40 kW ; Cut-off = 0.5 of stroke ; Back pressure = 1.1 bar ; Clearance = 5% of stroke; Mean piston speed = 125 m/min; R.P.M. = 300 ; Diagram factor = 0.8; Mechanical efficiency = 90%.

Assume hyperbolic expansion and neglect compression effect of piston rod. If the steam consumption is 700 kg/h, calculate 1. stroke length ; 2. cylinder diameter ; and 3. brake thermal efficiency of the engine.

Solution. Given : $p_1 = 12$ bar ; $x = 0.95$; B.P. = 40 kW ; $c = (v_2 - v_c)/v_s = 0.5$; $p_b = 1.1$ bar ; $b = v_c/v_s = 0.05$; Mean piston speed = 125 m/min ; $N = 300$ r.p.m. ; $K = 0.8$; $\eta_m = 90\% = 0.9$; $m_s = 700 \text{ kg/h} = 11.67$ kg/min

1. *Stroke length*

Let L = Length of stroke in metres.

We know that mean piston speed,

$$125 = 2\,L\,N = 2\,L \times 300 = 600\,L$$

$$\therefore \quad L = 125/600 = 0.208 \text{ m Ans.}$$

2. *Cylinder diameter*

Let D = Cylinder diameter is metres.

$$\therefore \text{ Area,} \quad A = \frac{\pi}{4} \times D^2 = 0.7854\,D^2 \text{ m}^2$$

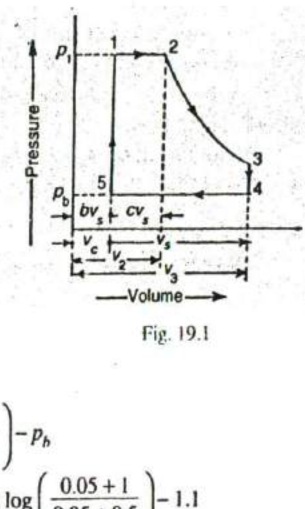


Fig. 19.1

We know that theoretical mean effective pressure,

$$^* p_m = p_1\,c + 2.3\,p_1\,(b + c) \log \left(\frac{b+1}{b+c} \right) - p_b$$

$$= 12 \times 0.5 + 2.3 \times 12\,(0.05 + 0.5) \log \left(\frac{0.05 + 1}{0.05 + 0.5} \right) - 1.1$$

$$= 6 + 15.18 \log 1.91 - 1.1 = 9.17 \text{ bar}$$

and actual mean effective pressure,

$$p_a = K \times p_m = 0.8 \times 9.17 = 7.34 \text{ bar}$$

We know that indicated power,

$$\text{I.P.} = \text{B.P.} / \eta_m = 40/0.9 = 44.44 \text{ kW}$$

We also know that indicated power (I.P.),

$$44.44 = \frac{200\,p_a\,L\,A\,N}{60} = \frac{200 \times 7.34 \times 0.208 \times 0.7854\,D^2 \times 300}{60} = 1200\,D^2$$

$$\therefore \quad D^2 = 0.037 \text{ or } D = 0.192 \text{ m Ans.}$$

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* Refer Chapter 17 Art. 17.8.

3. *Brake thermal efficiency*

From steam tables, corresponding to a pressure of 12 bar, we find that

$$h_{f1} = 798.4 \text{ kJ/kg, and } h_{fg1} = 1984.3 \text{ kJ/kg}$$

and corresponding to a pressure of 1.1 bar,

$$h_{fb} = 428.8 \text{ kJ/kg}$$

We know that enthalpy or total heat of steam supplied,

$$h_1 = h_{f1} + x \cdot h_{fg1} = 798.4 + 0.95 \times 1984.3 = 2683.5 \text{ kJ/kg}$$

$\therefore$ Brake thermal effi.ency

$$= \frac{\text{B.P.} \times 60}{11.67\,(h_1 - h_{fb})} = \frac{40 \times 60}{11.67\,(2683.5 - 428.8)} = 0.0912$$

$$= 9.12\ \% \textbf{ Ans.}$$

Example 19.7. *A steam engine is supplied with dry saturated steam at 7 bar and exhausts at 1.4 bar. The steam consumption was found to be 2 kg/min, when the engine output was 4.4 kW. Using steam tables or chart, find the relative efficiency of the engine.*

Solution. Given : $p_1 = 7$ bar ; $p_2 = 1.4$ bar ; $m_s = 2$ kg/min ; I.P. = 4.4 kW

From steam tables, corresponding to a pressure of 7 bar, we find that

$$h_1 = h_g = 2762 \text{ kJ/kg} \qquad \text{...(For dry saturated steam)}$$

and corresponding to a pressure of 1.4 bar,

$$h_{fb} = 458.4 \text{ kJ/kg}$$

We know that indicated thermal efficiency

$$= \frac{\text{I.P.} \times 60}{m_s\,(h_1 - h_{fb})} = \frac{4.4 \times 60}{2\,(2762 - 458.4)} = 0.0573 \text{ or } 5.73\%$$

Now the enthalpy or total heat of steam at 1.4 bar (h_2) may be obtained from Mollier chart by plotting the initial condition of steam at 14 bar on the saturation curve and then drawing a vertical line up to the pressure line of 1.4 bar. The corresponding value of h_2 is 2470 kJ/kg.

$$\therefore \text{ Rankine efficiency } = \frac{h_1 - h_2}{h_1 - h_{fb}} = \frac{2762 - 2470}{2762 - 458.4} = 0.127 \text{ or } 12.7\%$$

We know that relative efficiency

$$= \frac{\text{Thermal efficiency}}{\text{Rankine efficiency}} = \frac{0.0573}{0.127} = 0.451 \text{ or } 45.1\% \textbf{ Ans.}$$

19.3. Mass of Steam in the Engine Cylinder

A steam engine takes a certain mass of steam, from its boiler, during each stroke of the piston. This is known as *cylinder feed*. In addition to the cylinder feed, there is some steam left behind in the clearance space from the previous stroke. The steam left in the clearance space is known as *cushion steam*. Mathematically, total mass of steam in the engine cylinder,

$$m = m_s + m_c$$

m_s = Mass of steam used per hour, and

m_c = Mass of cushion steam.

The mass of steam used per hour may be calculated by measuring the mass of steam condensed in the condenser over a specified period. The mass of cushion steam is given by the relation ;

$$m_c = \frac{v_h}{v_{gh}}$$

where v_h = Volume of steam at any point (H) in the compression curve as shown in Fig. 19.2.

v_{gh} = Specific volume of steam at the given pressure at point H (from steam tables).

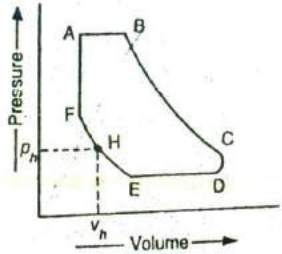


Fig. 19.2. p-v diagram of steam in the engine cylinder.

Example 19.8. *The volume and pressure at a certain point on the compression curve of an actual indicator diagram is 0.002 m³ and 2.4 bar respectively. Find the mass of cushion steam and the total mass of steam in the cylinder using expansion stroke. The steam consumption per minute is 6.5 kg and speed is 100 r.p.m. Assume the engine to be double acting.*

Solution. Given : $v_h = 0.002$ m³ ; $p_h = 2.4$ bar ; $m = 6.5$ kg/min ; $N = 100$ r.p.m.

Mass of cushion steam

From steam tables, corresponding to a pressure of 2.4 bar, we find that specific volume of steam at point H,

$$v_{gh} = 0.7464 \text{ m}^3/\text{kg}$$

We know that mass of cushion steam,

$$m_c = \frac{v_h}{v_{gh}} = \frac{0.002}{0.7464} = 0.0027 \text{ kg} \quad \textbf{Ans.}$$

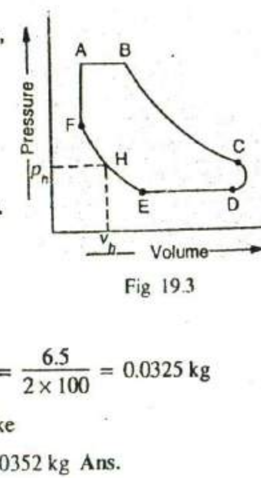


Fig 19.3

Total mass of steam

We know that mass of steam consumed per stroke,

$$m_s = \frac{\text{Mass of steam in kg/min}}{\text{Number of strokes /min}} = \frac{m}{2N} = \frac{6.5}{2 \times 100} = 0.0325 \text{ kg}$$

∴ Total mass of steam in the cylinder for expansion stroke

$$= m_s + m_c = 0.0325 + 0.0027 = 0.0352 \text{ kg} \quad \textbf{Ans.}$$

19.4. Missing Quantity

When a p-v curve for the mass of dry saturated steam is plotted on the actual indicator diagram, a curve (such as MN) is obtained. This curve is known as *saturation curve*. With the help of the saturation curve, the dryness fraction of steam, at all points of the expansion curve BC, is obtained. Thus, at any point K,

Dryness fraction of steam

$$= \frac{JK}{JL} = \frac{\text{Volume of steam in cylinder at } K}{\text{Volume of steam at } K\text{, if dry}}$$

Now the volume of steam represented by the line KL is called the volume of *missing quantity* at point K. The missing quantity per stroke at point K may also be regarded as the difference of the total mass of steam in the cylinder during expansion stroke and the mass of steam in the cylinder assuming dry saturated at point K. Therefore,

Missing quantity per stroke at K

= Total mass of steam in the cylinder during expansion stroke – Mass of steam in the cylinder assuming dry saturated at K

$$= (m_s + m_c) - \frac{v_k}{v_{gk}}$$

where v_k = Total volume of steam at point K, and

v_{gk} = Specific volume of dry saturated steam at pressure p_k (from steam tables).

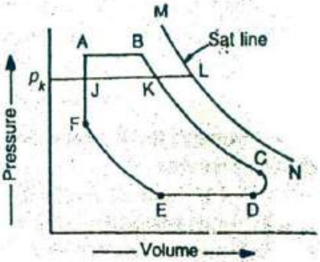


Fig. 19.4. Saturation curve in the p-v diagram of steam in the engine cylinder.

Notes : 1. The ratio of v_k / v_{gk} is known as indicated mass of steam (m_i) at any point K on the expansion curve. Therefore missing quantity per stroke

= Cylinder feed – Indicated mass of steam

2. The missing quantity is mainly due to cylinder condensation and a small amount of steam leakage past the valves and piston.

19.5. Methods of Reducing Missing Quantity or Cylinder Condensation

As a matter of fact, missing quantity is the loss of work in each stroke in an engine cylinder. It is, therefore, desirable to reduce the missing quantity. The cylinder condensation or missing quantity may be reduced by the following methods :

1. By the efficient steam jacketing of the cylinder walls.
2. By superheating the steam supplied to the engine cylinder.
3. By lagging in pipe from the boiler to the engine cylinder with a non-conducting material.
4. By compounding the expansion of steam in two cylinders, instead of one cylinder. Or, in other words, by keeping the expansion ratio small in each cylinder.
5. By increasing the speed of the engine.

Example 19.9. *The total mass of steam in an engine cylinder is 0.032 kg. It is found from the indicator diagram that at a pressure of 3.5 bar, the total cylinder volume occupied by steam is 0.0134 m^3. Find the dryness fraction of steam at this pressure and also the missing quantity.*

Solution. Given : $m = 0.032$ kg ; $p_k = 3.5$ bar ; $v_k = 0.0134$ m^3

From steam tables, we find that at a pressure of 3.5 bar, specific volume of steam at point K,

$$v_{gk} = 0.524 \text{ m}^3/\text{kg}$$

$\therefore$ Total volume of steam at point K, if dry

$$= 0.524 \times 0.032 = 0.0167 \text{ m}^3$$

Dryness fraction of steam

We know that dryness fraction of steam,

$$x = \frac{\text{Volume of steam in the cylinder at point } K}{\text{Volume of steam at point } K, \text{ if dry}} = \frac{0.0134}{0.0167}$$

$= 0.802$ **Ans**

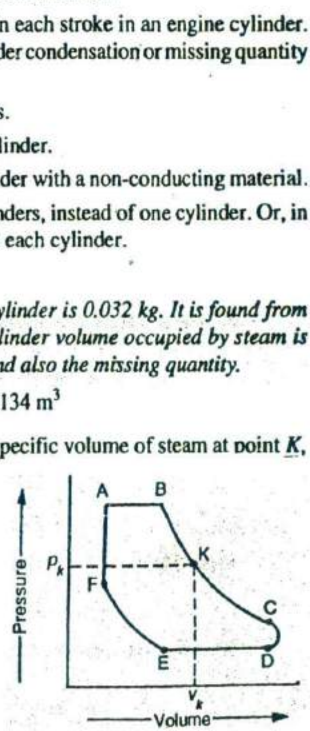


Fig. 19.5

Missing quantity

We also know that volume of missing quantity

$$= \text{Volume of steam at } K\text{, if dry} - \text{Volume of steam in the cylinder at point } K$$

$$= 0.0167 - 0.0134 = 0.0033 \text{ m}^3 \text{ Ans.}$$

Example 19.10. *Estimate the dryness fraction of steam in a cylinder at 0.7 of the stroke from the following data :*

R.P.M. = 100 ; Cut-off = 0.5 of stroke ; Steam condensed per minute = 45 kg/min ; Clearance = 8 % ; Swept volume = 0.1062 m³ ; Pressure of steam at 0.7 stroke = 4.2 bar ; Pressure of steam at 0.8 of return stroke on compression curve = 1.33 bar ; Volume of 1 kg of steam at 4.2 bar = 0.438 m³ ; Volume of 1 kg of steam at 1.33 bar = 1.296 m³.

Solution. Given : $N = 100$ r.p.m ; $v_B - v_A = 0.5\, v_s$; $m = 45$ kg/min ; $v_c = 0.08\, v_s$; $v_s = 0.1062\text{ m}^3$; $p_k = 4.2$ bar ; $p_h = 1.33$ bar ; $v_{gk} = 0.438\text{ m}^3/\text{kg}$; $v_{gh} = 1.296\text{ m}^3/\text{kg}$

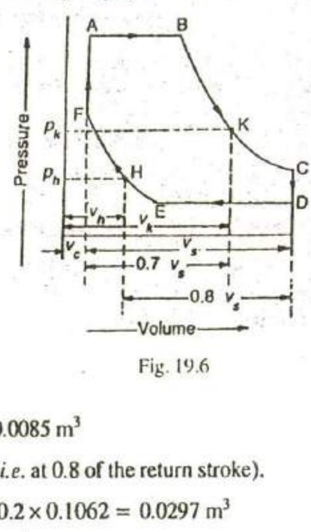


Fig. 19.6

We know that mass of steam used per stroke or cylinder feed,

$$m_s = \frac{\text{Mass of steam in kg/min}}{\text{No. of strokes/min}} = \frac{m}{N} = \frac{45}{100} = 0.45 \text{ kg}$$

. . . (Assuming single acting engine)

Clearance volume,

$$v_c = 0.08\, v_s = 0.08 \times 0.1062 = 0.0085 \text{ m}^3$$

and volume of steam at point H on the compression curve (*i.e.* at 0.8 of the return stroke).

$$v_h = v_c + (1 - 0.8)\, v_s = 0.0085 + 0.2 \times 0.1062 = 0.0297 \text{ m}^3$$

We know that mass of cushion steam,

$$m_c = \frac{v_h}{v_{gh}} = \frac{0.0297}{1.296} = 0.023 \text{ kg}$$

$\therefore$ Total mass of steam used during expansion stroke,

$$m_k = m_s + m_c = 0.45 + 0.023 = 0.473 \text{ kg}$$

Let x = Dryness fraction of steam at 0.7 of the stroke *i.e.* at point K,

Now volume of steam at point K (*i.e.* at 0.7 of the stroke),

$$v_k = v_c + 0.7\, v_s = 0.0085 + 0.7 \times 0.1062 = 0.083 \text{ m}^3$$

We know that mass of steam used during expansion stroke,

$$m_k = \frac{\text{Volume of steam at } K}{\text{Volume of steam at } K\text{, if dry}} = \frac{v_k}{x\, v_{gk}}$$

$$\therefore \quad x = \frac{v_k}{m_k\, v_{gk}} = \frac{0.083}{0.473 \times 0.438} = 0.4 \text{ Ans.}$$

Example 19.11. *The following results were obtained by measurements taken on an indicator diagram of a double acting steam engine :*

(i) Immediately after cut-off : Volume = 0.1 m^3 ; and pressure = 12.5 bar.

(ii) Immediately after compression has begun : Volume = 0.03 m^3 ; and pressure = 4 bar.

The speed of the engine was 105 r.p.m. and the steam supply per minute was 180 kg. Find the missing quantity in kg/min.

Solution. Given : $v_k = 0.1\ m^3$; $p_k = 12.5$ bar ; $v_h = 0.03\ m^3$; $p_h = 4$ bar ; $N = 105$ r.p.m. ; $m = 180$ kg/min

We know that mass of steam used per stroke, *i.e.* cylinder feed,

$$m_s = \frac{\text{Mass of steam used/min}}{\text{No. of strokes/min}} = \frac{m}{2N} = \frac{180}{2 \times 105} = 0.857 \text{ kg}$$

From steam tables, corresponding to a pressure of 12.5 bar, we find that specific volume of steam,

$$v_{gk} = 0.157 \text{ m}^3\text{/kg}$$

and corresponding to a pressure of 4 bar, specific volume of steam,

$$v_{gh} = 0.462 \text{ m}^3\text{/kg}$$

∴ Mass of cushion steam,

$$m_c = \frac{v_h}{v_{gh}} = \frac{0.03}{0.462} = 0.065 \text{ kg}$$

and indicated mass of steam,

$$m_i = \frac{v_k}{v_{gk}} = \frac{0.1}{0.157} = 0.637 \text{ kg}$$

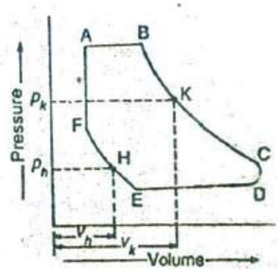


Fig. 19.7

We know that missing quantity per stroke

$$= (m_s + m_c) - m_i = (0.857 + 0.065) - 0.637 = 0.285 \text{ kg}$$

∴ Missing quantity/min

$$= 0.285 \times 2 \times 105 = 59.85 \text{ kg} \quad \textbf{Ans.}$$

Note : This is the missing quantity at the beginning of the expansion stroke.

19.6. Heat Balance Sheet

The complete record of heat supplied and heat rejected during a certain time (say one minute) by a steam engine in a tabulation form is known as *heat balance sheet*. The following values are generally required to complete the heat balance sheet of a steam engine.

1. *Heat supplied to cylinder per minute*

Let m_s = Mass of steam supplied to cylinder in kg/min,

m_j = Mass of steam supplied to jackets in kg/min, and

h = Total heat of steam supplied in kJ/kg.

∴ Heat supplied to cylinder/min.

$$= h\,(m_s + m_j) \text{ kJ/min} \qquad \ldots(i)$$

Note : If some water is received from receiver drain (m_r), then heat supplied to the cylinder/min.

$$= h\,(m_s + m_j + m_r) \text{ kJ/min}$$

2. *Heat absorbed in indicated power*

We know that heat absorbed in indicated power (I.P.) per minute

$$= \text{I.P.} \times 60 \text{ kJ/min} \qquad \ldots (ii)$$

3. *Heat rejected to the cooling water*

Let m_c = Mass of the cooling water/min,

t_o = Outlet temperature of cooling water, and

t_i = Inlet temperature of cooling water.

∴ Heat rejected to the cooling water

$$= m_c \times c_w (t_0 - t_i) = 4.2\, m_c (t_0 - t_i) \text{ kJ/min} \qquad \ldots (iii)$$

... ($\because$ Sp. heat of water, c_w = 4.2 kJ/kg K)

4. *Heat rejected in condensate*

Let t_c = Temperature of the condensate.

∴ Heat rejected in condensate

$$= m_s\, c_c\, t_c = 4.2\, m_s\, t_c \text{ kJ/min} \qquad \ldots (iv)$$

... ($\because$ Sp. heat of condensate, c_c = 4.2 kJ/kg K)

Note : The sum of heat rejected to the cooling water and condensate is known as heat rejected to exhaust steam.

5. *Heat rejected in jacket drain*

Let m_j = Mass of water drained for jackets/min, and

t_j = Temperature of jacket water.

∴ Heat rejected in jacket drain

$$= m_j\, c_w\, t_j = 4.2\, m_j\, t_j \text{ kJ/min} \qquad \ldots (v)$$

6. *Unaccounted heat*

There is always some loss of heat due to friction, leakage, radiation etc., which can not be determined experimentally. In order to complete the heat balance sheet, this loss is obtained by the difference of heat supplied to cylinder per minute and heat rejected in I.P., exhaust steam and jacket drain.

Finally, the heat balance sheet is prepared as given below :

S.No.	*Particulars*	*Heat in*	
		kJ/min	*%*
	Total heat supplied	...	*100*
1.	Heat absorbed in I.P.	...	...
2.	Heat in cooling water	...	...
3.	Heat in condensate	...	...
4.	Heat in jacket drain	...	...
5.	Unaccounted heat	...	...

Example 19.12. *The following observations were made during a trial of a jacketed simple steam engine :*

Pressure of steam supplied = *10 bar*

Cylinder feed	*= 13.5 kg/min*
Jacket feed	*= 1.5 kg/min*
Condition of cylinder and jacket feed	*= 95% dry*
Mass of circulating water	*= 220 kg/min*
Outlet temperature	*= 35° C*
Inlet temperature	*= 15° C*
Condensate temperature	*= 50° C*
Temperature of jacket drain	*= 150° C*
Indicated power	*= 80 kW*

Draw a heat balance sheet for the engine and also find indicated thermal efficiency.

Solution. Given : $p = 10$ bar ; $m_s = 13.5$ kg/min ; $m_j = 1.5$ kg/min ; $x = 95\% = 0.95$; $m_c = 220$ kg/min ; $t_o = 35°$ C ; $t_i = 15°$ C ; $t_c = 50°$ C ; $t_j = 150°$ C ; I.P. = 80 kW

Heat balance sheet

From steam tables, corresponding to a pressure of 10 bar, we find that

$$h_f = 762.6 \text{ kJ/kg}; \quad \text{and} \quad h_{fg} = 2013.6 \text{ kJ/kg}$$

We know that total heat in 1 kg of steam,

$$h = h_f + x\,h_{fg} = 762.6 + 0.95 \times 2013.6 = 2675.5 \text{ kJ/kg}$$

$\therefore$ Total heat supplied to the cylinder per minute

$$= h\,(m_s + m_j) = 2675.5\,(13.5 + 1.5) = 40\,132 \text{ kJ/min}$$

Heat absorbed in I.P. per minute

$$= \text{I.P.} \times 60 = 80 \times 60 = 4800 \text{ kJ/min} \qquad \ldots (i)$$

Heat rejected to cooling water per minute

$$= m_c\,c_w\,(t_o - t_i) = 220 \times 4.2\,(35 - 15) = 18\,480 \text{ kJ/min} \qquad \ldots (ii)$$

Heat in condensate per min

$$= m_s\,c_c\,t_c = 13.5 \times 4.2 \times 50 = 2835 \text{ kJ/min} \qquad \ldots (iii)$$

Heat rejected in jacket drain

$$= m_j\,c_w\,t_j = 1.5 \times 4.2 \times 150 = 945 \text{ kJ/min} \qquad \ldots (iv)$$

Unaccounted heat per min (by difference)

$$= 40\,132 - (4800 + 18\,480 + 2835 + 945) = 13\,072 \text{ kJ/min} \qquad \ldots (v)$$

Now prepare the heat balance sheet as given below :

S.No.	*Particulars*	*Heat in*	
		kJ/min	*%*
	Total heat supplied	40 132	100
1.	Heat absorbed in I.P.	4800	11.96
2.	Heat in cooling water	18 480	46.05
3.	Heat in condensate	2835	7.06
4.	Heat in jacket drain	945	2.35
5.	Unaccounted heat	13 072	32.58

Indicated thermal efficiency

We know that heat equivalent to I.P./min

$$= 4800 \text{ kJ/min}$$

and net heat supplied/min = Total heat supplied − Heat in condensate

$$= 40\,132 - 2835 = 37\,297 \text{ kJ/min}$$

∴ Indicated thermal efficiency

$$= \frac{\text{Heat equivalent to I.P./min}}{\text{Net heat supplied / min}} = \frac{4800}{37\,297} = 0.1287 \text{ or } 12.87\ \% \text{ Ans.}$$

Example 19.13. *The following observations were recorded during trial on a jacketed double acting compound steam engine supplied with dry saturated steam :*

I.P. of the steam engine	*= 180 kW*
Pressure of steam supplied	*= 6 bar*
Receiver pressure	*= 2.5 bar*
Cylinder feed	*= 17.8 kg/min*
Discharge from jacket drain	*= 2.2 kg/min*
Discharge from receiver drain	*= 1.4 kg/min*
Mass of circulating water	*= 350 kg/min*
Rise of temperature in cooling water	*= 25° C*
Condensate temperature	*= 50° C*
Temperature of jacket drain	*= 154.5° C*
Average brake torque	*= 4500 N-m*
Engine speed	*= 250 r.p.m.*

Find mechanical and brake thermal efficiency of the steam engine. Also prepare heat balance sheet on minute as well as percentage basis.

Solution. Given : I.P. = 180 kW ; $p = 6$ bar ; $p_r = 2.5$ bar ; $m_s = 17.8$ kg/min ; $m_j = 2.2$ kg/min ; $m_r = 1.4$ kg/min ; $m_w = 350$ kg/min ; $t_o - t_i = 25°$ C ; $t_c = 50°$ C ; $t_j = 154.5°$ C ; $T = 4500$ N-m; $N = 250$ r.p.m.

Mechanical efficiency

We know that brake power of the engine,

$$\text{B.P.} = \frac{2\pi N T}{60} = \frac{2\pi \times 250 \times 4500}{60} = 117\,800 \text{ W} = 117.8 \text{ kW}$$

∴ Mechanical efficiency,

$$\eta_m = \frac{\text{B.P.}}{\text{I.P.}} = \frac{117.8}{180} = 0.654 \text{ or } 65.4\ \% \text{ Ans.}$$

Brake thermal efficiency

We know that heat in B.P. per minute

$$= \text{B.P.} \times 60 = 117.8 \times 60 = 7068 \text{ kJ/min}$$

Heat in condensate per minute

$$= m_s\, c_c\, t_c = 17.8 \times 4.2 \times 50 = 3738 \text{ kJ/min}$$

From steam tables, corresponding to a pressure of 6 bar, we find that heat in 1 kg of dry saturated steam,

$$h = 2755.5 \text{ kJ/kg}$$

$\therefore$ Heat supplied to the cylinder per minute

$$= h(m_s + m_j + m_r) = 2755.5(17.8 + 2.2 + 1.4) = 58\,970 \text{ kJ/min}$$

and net heat supplied per minute

$$= \text{Total heat supplied} - \text{Heat in condensate}$$

$$= 58\,970 - 3738 = 55\,232 \text{ kJ/min}$$

$\therefore$ Brake thermal efficiency

$$= \frac{\text{Heat in B.P. per minute}}{\text{Net heat supplied per minute}} = \frac{7068}{55\,232}$$

$$= 0.128 \text{ or } 12.8\ \% \textbf{ Ans.}$$

Heat balance sheet

Heat in I.P. per minute

$$= \text{I.P} \times 60 = 180 \times 60 = 10\,800 \text{ kJ/min}$$

We know that heat rejected to the cooling water per minute

$$= m_w c_w (t_o - t_i) = 350 \times 4.2 \times 25 = 36\,750 \text{ kJ/min}$$

and heat rejected in jacket drain per minute

$$= m_j c_w t_j = 2.2 \times 4.2 \times 154.5 = 1428 \text{ kJ/min}$$

From steam tables, corresponding to a receiver pressure of 2.5 bar, we find that heat in 1 kg of saturated water,

$$h_f = 535.4 \text{ kJ/kg}$$

$\therefore$ Heat rejected in receiver drain per min

$$= 1.4 \times 535.4 = 750 \text{ kJ/min}$$

Unaccounted heat per minute (by difference)

$$= 58\,970 - (10\,800 + 36\,750 + 3738 + 1428 + 750) = 5504 \text{ kJ/min}$$

Now prepare the heat balance sheet as given below :

S.No.	*Particulars*	*Heat in*	
		kJ/min	*%*
	Total heat supplied	58 970	100
1.	Heat absorbed in I.P.	10 800	18.31
2.	Heat in cooling water	36 750	62.32
3.	Heat in condensate	3738	6.34
4.	Heat in jacket drain	1428	2.42
5.	Heat in receiver drain	750	1.27
6.	Unaccounted heat	5504	9.34

19.7. Governing of Steam Engines

As a matter of fact, simple and compound steam engines are always designed to run at a particular speed. But in actual practice, load on the engine keeps on fluctuating from time to time. The change of load, on the steam engine, is sure to change its speed and rate of steam flow. It has been observed that if load on the steam engine is decreased, without changing the quantity of steam, the engine will run at a higher speed. Similarly, if load on the steam engine is increased, without changing the quantity of steam, the engine will run at a lower speed.

Now, in order to have a high efficiency of a steam engine, at different load conditions, its speed must be kept constant as far as possible. The process of providing any arrangement, which will keep the speed constant (according to the changing load conditions) is known as *governing of steam engines*.

We know that work done in a cylinder of a steam engine, is equal to the area of the indicator diagram to some scale. It is thus obvious, that if the engine is subjected to an increased load, more work must be done in the cylinder. Similarly, if the engine is subjected to a reduced load, less work must be done in the cylinder. Now in order to do a greater amount of work in the cylinder, area of the indicator diagram must be increased. Similarly, in order to do a less amount of work in the cylinder, area of the indicator diagram must be decreased. A little consideration will show, that in order to increase the area of indicator diagram either steam pressure or volume of the intake steam should be increased (or decreased for decrease in area of the indicator diagram). These days, both the above mentioned methods are used for the governing of a steam engine. In the following pages, we shall discuss the governing of simple and compound steam engines.

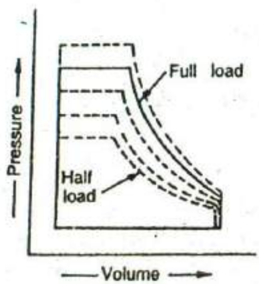
19.8. Governing of Simple Steam Engines

The governing of simple steam engines is done by a number of methods. But the following two methods are important from the subject point of view :

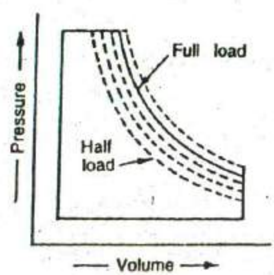
1. Throttle governing, and 2. Cut-off governing.

These two methods of governing of simple steam engines are discussed, in detail, in the following pages.

19.9. Throttle Governing of Simple Steam Engines



(*a*) Throttle governing.



(*b*) Cut-off governing.

Fig. 19.8. Governing of simple steam engines.

The throttle governing, of simple steam engines, is a method of controlling the engine output by varying pressure of the intake steam. The pressure of intake steam is varied by opening or closing the throttle valve under the control of a centrifugal governor. This is illustrated in Fig. 19.8 (*a*). In this figure, actual indicator diagram of the simple steam engine at full load on p-v diagram is shown. Whenever load on the engine is decreased, it tends to run the engine at a higher speed. Now, in order to run the engine at the designed speed, less work is required to be done in the engine cylinder. Or in other words, area of the indicator diagram is required to be reduced. In throttle governing, area of the indicator diagram is reduced by reducing pressure of the admission steam as shown in Fig. 19.8 (*a*). Similarly, if the load further decreases, pressure of the admission steam is also further reduced.

Notes : 1. In throttle governing, cut-off of the engine cylinder remains the same.

2. If load on the engine is increased, then pressure of the admission steam is also increased to suit the increased load.

3. The throttle governing of a simple steam engine results in the reduction of its thermal efficiency, and as such is somewhat wasteful with regard to the steam. The reason for the same is that at any load (below the full load), the full pressure of the steam is not used. Moreover, due to constant cut-off, a large quantity of steam is used in every stroke. This also tends to lower the thermal efficiency of the engine.

19.10. Cut-off Governing of Simple Steam Engines

The cut-off governing, of a simple steam engine, is a method of controlling the engine output by varying volume of intake steam. This is done by varying the cut-off point by a slide valve under the control of a centrifugal governor. This is illustrated in Fig. 19.8 (*b*). In this figure, actual indicator diagram of the simple steam engine at full load on *p*-*v* diagram is shown. Whenever load on the engine is decreased, it tends to run the engine at a higher speed. Now, in order to run the engine at the designed speed, less work is required to be done in the engine cylinder. Or in other words, area of the indicator diagram is required to be reduced. In cut-off governing, area of the indicator diagram is reduced by reducing volume of the intake steam, as shown in Fig. 19.8 (*b*). Similarly, if the load further decreases, volume of the intake steam is also further reduced.

Notes : 1. In a cut-off governing, pressure of the intake steam remains the same.

2. If load on the engine is increased, then pressure of the admission steam is also increased to suit the increased load.

3. This method is more economical and efficient. That is why, these days, cut-off governing is mostly used. But it requires a special valve gear.

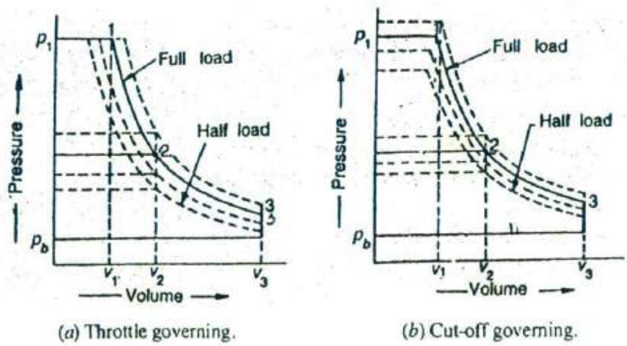
19.11 Governing of Compound Steam Engines

The governing of compound steam engines is done by a number of methods. But, like the governing of simple steam engines, the following two methods are important from the subject point of view :

1. Throttle governing, and 2. Cut-off governing.

These two methods of governing compound steam engines are discussed, in detail, in the following pages.

19.12. Throttle Governing of Compound Steam Engines



(*a*) Throttle governing. (*b*) Cut-off governing.

Fig. 19.9. Governing of compound steam engines.

The throttle governing, of a compound steam engine, is a method of controlling the engine output by varying pressure of the intake steam. The pressure of intake steam is varied by opening or closing the throttle valve under the control of a centrifugal governor. This is illustrated in Fig. 19.9 (*a*). In this figure, indicator diagram of a compound steam engine is shown with complete expansion in H.P. cylinder and incomplete expansion in L.P. cylinder at full load. Whenever load on the engine is decreased, it tends to run the engine at a higher speed. Now in order to run the engine at the designed speed, less work is required to be done in the engine cylinders. Or in other words, area of the indicator diagram is required to be reduced. In throttle governing, area of the indicator diagram is reduced by reducing pressure of the admission steam as shown in Fig. 19.9 (*a*). Similarly, if the load further decreases, pressure of the admission steam is also further reduced.

Notes : 1. In throttle governing, cut-off point of the engine cylinder remains the same.

2. If load on the engine is increased, then pressure of the admission steam is also increased to suit the increased load.

3. In throttle governing, work done in H.P. cylinder is greatly reduced, whereas work done in L.P. cylinder is slightly reduced.

19.13. Cut-off Governing of Compound Steam Engines

The cut-off governing of a compound steam engine, is a method of controlling the engine output by varying volume of intake steam in the H.P. cylinder. This is done by varying the cut-off point under the control of a centrifugal governor. This is illustrated in Fig. 19.9 (*b*). In this figure, actual indicator diagram of a compound steam engine at full load on p-v diagram is shown. Whenever load on the engine is decreased, it tends to run the engine at a higher speed. Now, in order to run the engine at the designed speed, less work is required to be done in the engine cylinder. Or in other words, area of the indicator diagram is required to be reduced. In this method, area of the indicator diagram is reduced by reducing the volume of the intake steam as shown in Fig. 19.9 (*b*). Similarly, if the load further decreases, volume of the intake steam is also further reduced.

From the geometry of the indicator diagram, we see that when volume of the intake steam is reduced, it will reduce the pressure of steam at the release point of H.P. cylinder (or admission into the L.P. cylinder). This will happen as the volume of H.P. cylinder (v_2) will remain constant. In such a case, area of H.P. cylinder indicator diagram will slightly increase ; whereas area of L.P. cylinder indicator diagram will considerably decrease. The net effect on the indicator diagram will be to reduce its total area. Or in other words to reduce the total work done in both the cylinders.

Notes : 1. In this method, pressure of the intake steam remains the same.

2. If load on the engine is increased, then volume of the admission steam is also increased to suit the increased load. In this case, area of the H.P. cylinder indicator diagram will slightly decrease; whereas area of the L.P. cylinder indicator diagram will considerably increase. The net effect on the indicator diagram will be to increase its total area.

3. If the engine runs on a very little load, area of the L.P. cylinder indicator diagram may almost become negligible.

19.14. Steam Consumption (Willian's Law)

The amount of steam used by an engine is measured by weighing the condensate collected from the condenser into which the engine exhausts.

When the steam consumption per hour is plotted against the indicated power (I.P.) during a test on a throttle governed engine, it will be a straight line. This shows that the steam consumption per hour is directly proportional to I.P. It is called as *Willian's law* and the straight line is called *Willian's line* as shown in Fig. 19.10.

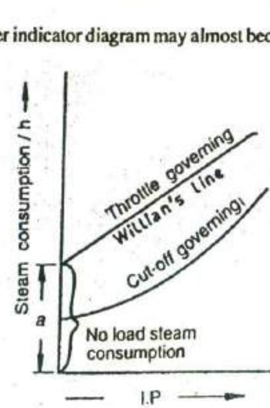


Fig. 19.10. Willian's law for steam consumption.

Willian's law holds good only for a throttle-governed engine, because the ratio of expansion remains constant. This condition is not fulfilled in a cut-off governed engine. The Willian's line follows the law :

$$m = a + b \times \text{I.P.}$$

where m = Steam consumption per hour,

a = A constant, *i.e.* no load consumption per hour,

b = Another constant representing the shape of the Willian's line, and

I.P. = Indicated power.

Example 19.14. *A throttle-governed steam engine develops 15 kW with 280 kg of steam used per hour and 37.5 kW with 520 kg of steam per hour. Determine the steam in kg/h when steam engine develops 20 kW.*

Solution. Given : $m_1 = 280$ kg/h when I.P. = 15 kW ; $m_2 = 520$ kg/h when I.P. = 37.5 kW

We know that Willian's law is

$$m = a + b \times \text{I.P.}$$

$$\therefore \quad 280 = a + b \times 15 \qquad \ldots (i)$$

and $$520 = a + b \times 37.5 \qquad \ldots (ii)$$

Subtracting equation (*i*) from equation (*ii*),

$$240 = 22.5\, b \quad \text{or} \quad b = 10.67$$

Substituting the value of $b = 10.67$ in equation (i), we get $a = 120$.

Now the equation of Willian's line becomes

$$m = 120 + 10.67 \times \text{I.P.}$$

$\therefore$ Steam cosumption at 20 kW

$$= 120 + 10.67 \times 20 = 333.4 \text{ kg/h} \quad \textbf{Ans.}$$

Example 19.15. *A steam engine of 400 kW I.P., governed by throttle control consumes 10 kg of steam per hour per I.P. at full load and 12.5 kg at half load. Find the steam consumption, if it runs at two-third the load for 4 hours.*

Solution. Given : I.P. = 400 kW

We know that steam consumption at full load

$$= 10 \text{ kg/I.P./h}$$

$\therefore$ Total steam consumption per hour at full load *i.e.* at 400 kW,

$$m_1 = 10 \times 400 = 4000 \text{ kg/h}$$

We also know that steam consumption at half load

$$= 12.5 \text{ kg/I.P./h}$$

$\therefore$ Total steam consumption per hour at half load *i.e.* at 200 kW,

$$m_2 = 12.5 \times 200 = 2500 \text{ kg/h}$$

According to Willian's law :

$$4000 = a + b \times 400 \qquad \ldots (i)$$

and $$2500 = a + b \times 200 \qquad \ldots (ii)$$

Subtracting equation (*ii*) from (*i*),

$$1500 = 200\, b \quad \text{or} \quad b = 7.5$$

Substituting the value of $b = 7.5$ in equation (*i*), we get $a = 1000$.

Now the equation of Willian's line becomes

$$m = 1000 + 7.5 \times \text{I.P.}$$

$\therefore$ Steam consumption at 2/3 load *i.e.* at $2 \times 400/3$ kW,

$$= 1000 + 7.5 \times \frac{2}{3} \times 400 = 3000 \text{ kg/h}$$

and steam consumption for 4 hours

$$= 4 \times 3000 = 12\,000 \text{ kg} \textbf{ Ans.}$$

Example 19.16. *A steam engine with throttle governing and running at 140 r.p.m. develops 4.2 kW and 9.5 kW indicated power while steam consumption is 104 kg/h and 179 kg/h respectively. The steam is supplied to the engine at a pressure of 6 bar and 0.9 dryness fraction. The condensate temperature is 40° C. Find the indicated thermal efficiency of the engine when it runs at 140 r.p.m. and develops 7.5 kW.*

Solution. Given : $^*N = 140$ r.p.m. ; $(\text{I.P.})_1 = 4.2$ kW ; $(\text{I.P.})_2 = 9.5$ kW ; $m_1 = 104$ kg/h ; $m_2 = 179$ kg/h ; $p = 6$ bar ; $x = 0.9$; $t_c = 40°$ C

We know that Willian's law is

$$m = a + b \times \text{I.P.}$$

$$\therefore \quad 104 = a + b \times 4.2 \qquad \ldots (i)$$

and $$179 = a + b \times 9.5 \qquad \ldots (ii)$$

Subtracting equation (*i*) from (*ii*),

$$75 = 5.3\,b \quad \text{or} \quad b = 14.15$$

Substituting the value of $b = 14.15$ in equation (*i*), we get $a = 44.57$.

Now the equation of Willian's line becomes

$$m = 44.57 + 14.15 \times \text{I.P.}$$

$\therefore$ Steam consumption when engine develops 7.5 kW,

$$m_s = 44.57 + 14.15 \times 7.5 = 150.7 \text{ kg/h} = 2.51 \text{ kg/min}$$

From steam tables corresponding to a pressure of 6 bar, we find that

$$h_{f1} = 670.4 \text{ kJ/kg} \text{ ; and } h_{fg1} = 2085 \text{ kJ/kg}$$

and corresponding to a condensate temperature of 40° C,

$$h_{fb} = 167.5 \text{ kJ/kg}$$

We know that total heat of steam supplied,

$$h_1 = h_{f1} + x\,h_{fg1} = 670.4 + 0.9 \times 2085 = 2547 \text{ kJ/kg}$$

$\therefore$ Indicated thermal efficiency

$$= \frac{\text{I.P.} \times 60}{m_s\,(h_1 - h_{fb})} = \frac{7.5 \times 60}{2.51\,(2547 - 167.5)} = 0.075 \text{ or } 7.5\% \textbf{ Ans.}$$

EXERCISES

1. A double acting steam engine with piston diameter 275 mm, stroke length 650 mm and cut-off 50% of stroke length is supplied steam at a pressure of 7 bar. The back pressure is 1.2 bar. Assuming a diagram factor of 0.75, find the indicated power of the engine when it runs at 250 r.p.m. Also find mechanical efficiency of the engine if its brake power is 100 kW. Neglect clearance. [**Ans.** 114 kW ; 87.7 %]

---

* Superfluous data.

2. The following data pertains to a single cylinder double acting steam engine :

Admission pressure = 14 bar ; Back pressure = 0.35 bar ; Cut-off = 0.4 of stroke ; Diagram factor = 0.7 ; Cylinder diameter = 300 mm ; Stroke = 1.5 Cylinder bore ; Mechanical efficieney = 80%.

Neglecting clearance, estimate brake power developed by the engine running at 200 r.p.m.

[Ans. 123.2 kW]

3. A single cylinder double acting steam engine of 280 mm bore and 450 mm stroke works on a supply pressure of 10 bar and back pressure of 0.15 bar. The cut-off takes place at 25% of stroke. Assuming a diagram factor of 0.7 and engine speed 3 revolutions per second, find the I.P. of the engine. If the engine consumes 15 kg of dry saturated steam per minute, find the indicated thermal efficiency of the engine. Neglect clearance.

[Ans. 10.62%]

4. A single cylinder double acting steam engine with cylinder diameter 150 mm ; stroke length 200 mm is required to develop 20 kW of indicated power at 300 r.p.m. with cut-off at 20% of the stroke. Find the admission pressure if the diagram factor is 0.72 and back pressure is 0.28 bar. Also calculate the indicated thermal efficiency of the engine, when it receives 240 kg of dry steam per hour. **[Ans. 15.6 bar ; 11.96 %]**

5. The following observations were made during the trial of a steam engine :

Radius of the brake wheel = 500 mm ; Load on the brake = 1350 N ; Spring balance reading = 50 N ; Steam supplied per minute = 7 kg ; Admission pressure of steam = 11 bar ; Quality of steam = 0.98 dry ; Back pressure = 1.1 bar ; Engine speed = 240 r.p.m.

Find the brake thermal efficiency of the engine. **[Ans. 8.5%]**

6. The following data relates to a single cylinder double acting steam engine using dry saturated steam :

Bore = 300 mm ; Stroke = 450 mm ; Hypothetical mean effective pressure = 3.5 bar ; Cut-off = 25% of stroke ; Exhaust pressure = 0.5 bar ; Steam consumption = 560 kg/h ; Speed = 160 r.p.m. ; Diagram factor = 0.8 ; Mechanical efficiency = 80%.

Calculate brake thermal efficiency. Assume hyperbolic expansion and neglect clearance and compression. **[Ans. 10.1%]**

7. A single cylinder double acting steam engine is supplied with dry saturated steam at 11 bar. The bore is 300 mm and the stroke 375 mm. The engine runs at 250 r.p.m. with a cut-off ratio of 1/3 and against a back pressure of 0.36 bar. If the diagram factor is 0.8, calculate the indicated power if mechanical efficiency is 85% and brake thermal efficiency 15%. Calculate also the brake power and the specific steam consumption in kg/B.P./h. Assume that the condensate is used as the feed water to the boiler and the expansion of steam is hyperbolic. Neglect clearance **[Ans. 129.5 kW ; 110 kW ; 9.7 kg/B.P./h]**

8. The following observations were recorded during a trial of a single cylinder double acting non-condensing steam engine :

Cylinder diameter = 250 mm ; Piston rod diameter = 50 mm ; Stroke = 350 mm ; Speed = 240 r.p.m.; Cut-off = 0.3 of stroke ; Length of indicator diagram = 53 mm ; Area of indicator diagram for cover end = 1570 mm^2 ; Area of indicator diagram for crank end = 1440 mm^2 ; Spring number = 120 bar/m ; Diameter of brake wheel = 1.67 m ; Diameter of brake rope = 20 mm ; Dead load on the brake = 2000 N ; Reading of spring balance = 200 N ; Pressure of steam supplied = 10.5 bar ; Dryness fraction of steam supplied = 0.9.

Find : 1. Indicated power ; 2. Brake power ; 3. Mechanical efficiency ; 4. Specific steam consumption on I.P. and B.P. basis ; 5. Indicated thermal efficiency ; and 6. Brake thermal efficiency.

[Ans. 45.9 kW ; 38.23 kW ; 83.3% ; 19.35 kg/kWh. 23.23 kg/kWh ; 7.2% ; 6%]

9. A steam engine uses 500 kg of dry saturated steam per hour at a pressure of 20 bar and exhaust takes place at a pressure of 0.2 bar with dryness fraction of 0.78. Find Rankine efficiency and relative efficiency of the engine, if it develops 40 kW at full load. **[Ans. 26.8% ; 50.3%]**

10. The volume and pressure at a certain point on the compression curve of an actual indicator diagram is 0.028 m^3 and 4 bar respectively. Estimate the total mass of steam in the cylinder per stroke when the steam supplied is 210 kg/min at 105 r.p.m. **[Ans. 1.06 kg]**

11. In a double acting engine running at 100 r.p.m., the following values were noted :

(*i*) Immediately after cut-off :

Volume = 0.0588 m^3 ; pressure = 5 bar (*ii*) Immediately after compression has begun :

Volume = 0.0211 m^3 ; pressure = 1.4 bar

Find the missing quantity in kg/min. The steam supplied is 40 kg/min. [Ans. 18 kg/min]

12. The following values were recorded during a test on a single cylinder double acting jacketed steam engine :

Pressure of steam supplied = 5.5 bar ; Mass of steam used in engine cylinder = 5.4 kg/min ; Mass of steam used in jacket = 0.5 kg/min ; Dryness fraction of steam = 0.95 ; Mass of circulating water = 126.5 kg/min ; Rise in temperature of circulating water = 20° C ; Condensate temperature = 40° C ; Temperature of jacket drain = 140° C ; Indicated power = 25 kW.

Draw a heat balance sheet for the engine and find its indicated thermal efficiency. [Ans. 8.16%]

13. The following observations were recorded during a trial on a single cylinder double acting condensing type steam engine :

Pressure of steam supply = 8 bar ; Mass of condensate available from condenser per hour = 95 kg ; Quality of steam supplied to the engine = dry and saturated ; Condenser vacuum = 300 mm of Hg ; Engine speed = 150 r.p.m. ; Mean effective pressure on cover end side = 1.52 bar ; Mean effective pressure on crank end side = 1.22 bar ; Diameter of cylinder = 200 mm ; Length of stroke = 300 mm ; Piston rod diameter = 38 mm ; Net load on the brake = 950 N ; Effective diameter of brake wheel = 0.75 m ; Mass of cooling water per minute = 45 kg ; Rise in temperature of cooling water = 18° C ; Condensate temperature = 55° C.

Find the mechanical efficiency and brake thermal efficiency. Also draw a heat balance sheet.

[Ans. 88.14%; 8.36%]

14. A reciprocating steam engine, governed by throttling, uses 530 kg of steam per hour, when developing 55 kW I.P. It uses 2160 kg per hour when developing 280 kW I.P. Find the approximate power of this engine when the steam consumption is 1580 kg/h, assuming the Willian's relation holds good.

[Ans. 200 kW]

15. A throttle governed steam engine requires 500 kg of steam per hour while developing 37.5 kW and 2000 kg per hour when developing 187.5 kW. Find the thermal efficiency of the engine when it develops 115 kW assuming the steam supplied to be dry saturated at a pressure of 15 bar and exhaust pressure to be 0.3 bar.

[Ans. 13%]

QUESTIONS

1. What is meant by 'efficiency' of a steam engine ?
2. Differentiate between :
   (*a*) Mechanical efficiency and overall efficiency.
   (*b*) Indicated thermal efficiency and brake thermal efficiency.
   (*c*) Relative efficiency and Rankine efficiency.
3. Define missing quantity. Discuss the method to find it.
4. What are the methods used to reduce cylinder condensation in steam engine ?
5. Explain the term 'heat balance sheet'. What important light does it throw on the working of a steam engine.
6. What do you understand by the term 'governing of steam engines'? Explain its necessity.
7. Discuss the methods of governing a simple steam engine.
8. Explain clearly, the effect on the distribution of work between the cylinders of a two cylinder compound steam engine, when governing is done by (*a*) throttling, and (*b*) the alterations of cut-off in the H.P. cylinder.
9. State and explain Willian's law.

OBJECTIVE TYPE QUESTIONS

1. There is always some steam left in the clearance space from the previous stroke. This steam left in the clearance space is known as

(*a*) wet steam (*b*) saturated steam (*c*) superheated steam (*d*) cushion steam

2. The missing equality per stroke is equal to

(*a*) cylinder feed – indicated mass of steam

(*b*) cylinder feed + indicated mass of steam

(*c*) mass of cushion steam + indicated mass of steam

(*d*) mass of cushion steam – cylinder feed

3. The throttle governing of steam engines is a method of controlling the engine output by varying

(*a*) volume of intake steam (*b*) pressure of intake steam

(*c*) temperature of intake steam (*d*) all of these

4. Willian's line for a steam engine is a straight line relationship between the steam consumption per hour and

(*a*) indicated power (*b*) brake power (*c*) efficiency (*d*) pressure of steam

5. Wilian's law holds good for .... governed engine.

(*a*) cut-off (*b*) throttle

ANSWERS

1. (*d*) 2. (*a*) 3. (*b*) 4. (*a*) 5. (*b*)

20

Steam Condensers

1. Introduction. 2. Advantages of a Condenser in a Steam Power Plant. 3. Requirements of a Steam Condensing Plant. 4. Classification of Condensers. 5. Jet Condensers. 6. Types of Jet Condensers. 7. Parallel Flow Jet Condensers. 8. Counterflow or Low Level Jet Condensers. 9. Barometric or High Level Jet Condensers. 10. Ejector Condensers. 11. Surface Condensers. 12. Types of Surface Condensers. 13. Down Flow Surface Condensers. 14. Central Flow Condensers. 15. Regenerative Surface Condensers. 16. Evaporative Condensers. 17. Comparison of Jet and Surface Condensers. 18. Mixture of Air and Steam (Dalton's Law of Partial Pressures). 19. Measurement of Vacuum in a Condenser. 20. Vacuum Efficiency. 21. Condenser Efficiency. 22. Mass of Cooling Water Required or Condensation of Steam. 23. Sources of Air into the Condenser. 24. Effects of Air Leakage. 25. Air Pump. 26. Edwards Air Pump. 27. Cooling Towers. 28. Types of Cooling Towers.

20.1. Introduction

A steam condenser is a closed vessel into which the steam is exhausted, and condensed after doing work in an engine cylinder or turbine. A steam condenser has the following two objects :

1. The primary object is to maintain a low pressure (below atmospheric pressure) so as to obtain the maximum possible energy from steam and thus to secure a high efficiency.
2. The secondary object is to supply pure feed water to the hot well, from where it is pumped back to the boiler.

Note : The low pressure is accompanied by low temperature and thus all condensers maintain a vacuum under normal conditions. The condensed steam is called condensate. The temperature of condensate is higher on leaving the condenser than that of circulating water at inlet. It is thus obvious, that the condensate will have a considerable liquid heat.

20.2. Advantages of a Condenser in a Steam Power Plant

Following are the main advantages of incorporating a condenser in a steam power plant :

1. It increases expansion ratio of steam, and thus increases efficiency of the plant.
2. It reduces back pressure of the steam, and thus more work can be obtained.
3. It reduces temperature of the exhaust steam, and thus more work can be obtained.
4. The reuse of condensate (*i.e.* condensed steam) as feed water for boilers reduces the cost of power generation.
5. The temperature of condensate is higher than that of fresh water. Therefore the amount of heat supplied per kg of steam is reduced.

20.3. Requirements of a Steam Condensing Plant

The principle requirements of a condensing plant, as shown in Fig. 20.1, are as follows :

1. *Condenser.* It is a closed vessel is which steam is condensed. The steam gives up heat energy to coolant (which is water) during the process of condensation.

2. *Condensate pump.* It is a pump, which removes condensate (*i.e.* condensed steam) from the condenser to the hot well.
3. *Hot well.* It is a sump between the condenser and boiler, which receives condensate pumped by the condensate pump.

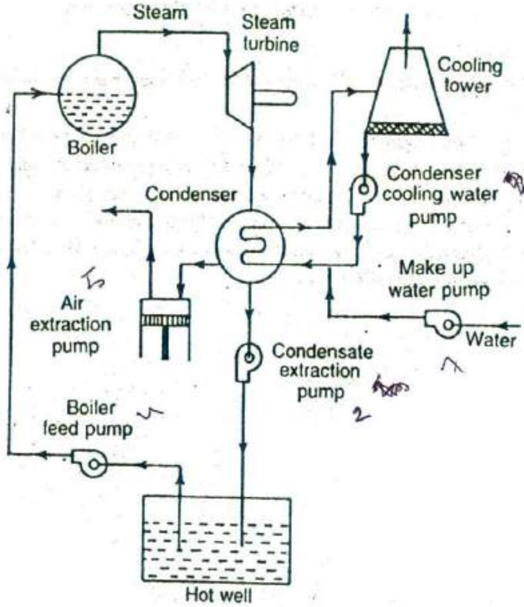


Fig. 20.1. Steam condensing plant.

4. *Boiler feed pump.* It is a pump, which pumps the condensate from the hot well to the boiler. This is done by increasing the pressure of condensate above the boil- pressure.
5. *Air extraction pump.* It is a pump which extracts (*i.e.* removes) air from the condenser.
6. *Cooling tower.* It is a tower used for cooling the water which is discharged from the condenser.
7. *Cooling water pump.* It is a pump, which circulates the cooling water through the condenser.

20.4. Classification of Condensers

The steam condensers may be broadly classified into the following two types, depending upon the way in which the steam is condensed :

1. Jet condensers or mixing ty_e condensers, and
2. Surface condensers or non-mixing type condensers.

20.5. Jet Condensers

These days, the jet condensers are seldom used because there is some loss of condensate during the process of condensation and high power requirements for the pumps used. Moreover, the condensate can not be used as feed water to the boiler as it is not free from salt. However, jet condensers may be used at places where water of good quality is easily available in sufficient quantit

20.6. Types of Jet Condensers

The jet condensers may be further classified, on the basis of the direction of flow of the condensate and the arrangement of the tubing system, into the following four types :

1. Parallel flow jet condenser, 2. Counterflow or low level jet condenser, 3. Barometric or high level jet condenser, and 4. Ejector condenser.

These condensers are discussed, in detail, in the following pages.

20.7. Parallel Flow Jet Condensers

In parallel flow jet condensers, both the steam and water enter at the top, and the mixture is removed from the bot om.

The principle of this condenser is shown in Fig. 20.2. The exhaust steam is condensed when it mixes up with water. The condensate, cooling water and air flow downwards and are removed by two separate pumps known as air pump and condensate pump. Sometimes, a single pump known as wet air pump, is also used to remove both air and condensate. But the former gives a greater vacuum. The condensate pump delivers the condensate to the hot well, from where surplus water flows to the cooling water tank through an overflow pipe.

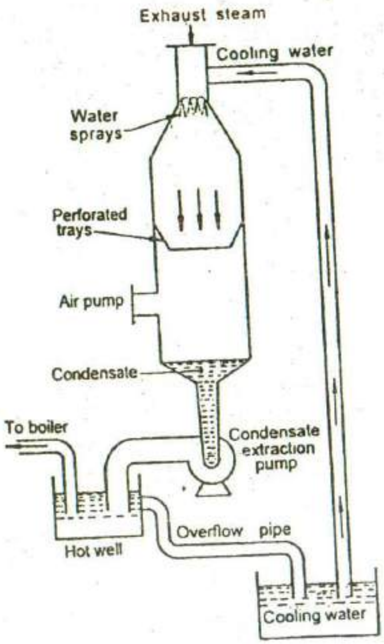


Fig. 20.2. Parallel flow jet condenser.

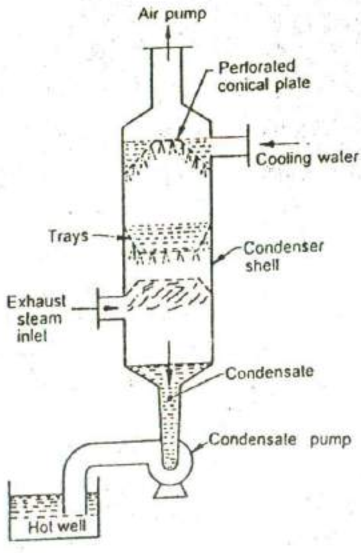


Fig. 20.3. Counterflow jet condenser.

20.8. Counterflow or Low Level Jet Condensers

In counterflow or low level jet condensers, the exhaust steam enters at the bottom, flows upwards and meets the downcoming cooling water.

The vacuum is created by the air pump, placed at the top of the condenser shell. This draws the supply of cooling water, which falls in a large number of jets, through perforated conical plate as shown in Fig. 20.3. The falling water is caught in the trays, from which it escapes in a second series of jets and meets the exhaust steam entering at the bottom. The rapid condensation occurs, and the condensate and cooling water descends through a vertical pipe to the condensate pump, which delivers it to the hot well.

20.9. Barometric or High Level Jet Condensers

These condensers are provided at a high level with a long vertical discharge pipe as shown in Fig. 20.4. In high level jet condensers, exhaust steam enters at the bottom, flows upwards and meets the downcoming cooling water in the same way as that of low level jet condenser. The vacuum is created by the air pump, placed at the top of the condenser shell. The condensate and cooling water descends through a vertical pipe to the hot well without the aid of any pump. The surplus water from the hot well flows to the cooling water tank through an overflow pipe.

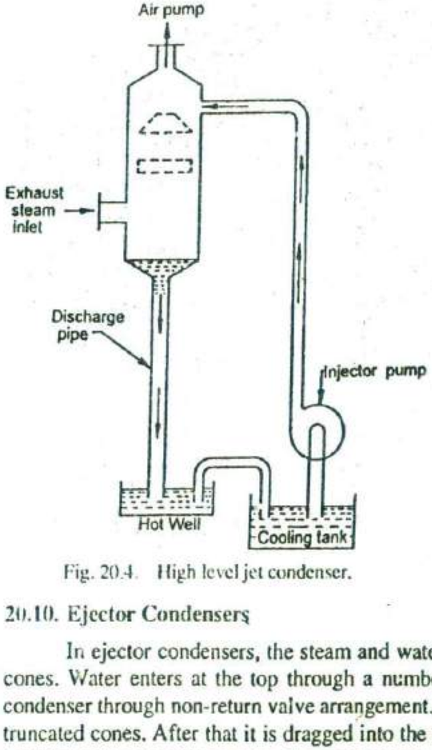


Fig. 20.4. High level jet condenser.

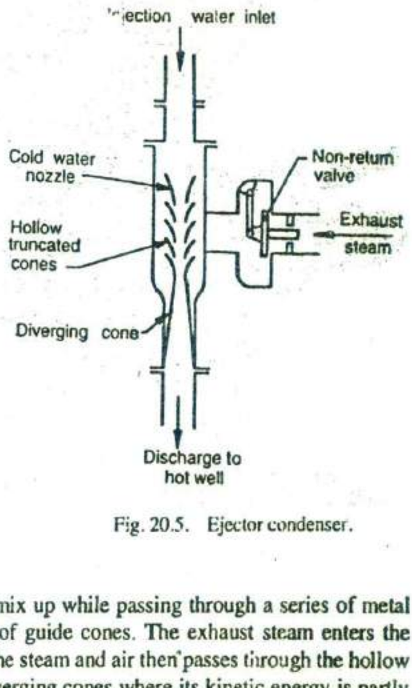


Fig. 20.5. Ejector condenser.

20.10. Ejector Condensers

In ejector condensers, the steam and water mix up while passing through a series of metal cones. Water enters at the top through a number of guide cones. The exhaust steam enters the condenser through non-return valve arrangement. The steam and air then passes through the hollow truncated cones. After that it is dragged into the diverging cones where its kinetic energy is partly transformed to pressure energy. The condensate and cooling water is then discharged to the hot well as shown in Fig. 20.5.

20.11. Surface Condensers

A surface condenser has a great advantage over the jet condensers, as the condensate does not mix up with the cooling water. As a result of this, whole condensate can be reused in the boiler. This type of condenser is essential in ships which can carry only a limited quantity of fresh water for the boilers. It is also widely used in land installations, where inferior water is available or the better quality of water for feed is to be used economically.

Fig. 20.6 shows a longitudinal section of a two pass surface condenser. It consists of a horizontal cast iron cylindrical vessel packed with tubes, through which the cooling water flows. The ends of the condenser are cut off by vertical perforated type plates into which water tubes are fixed. This is done in such a manner that the leakage of water into the centre condensing space is prevented.

The water tubes pass horizontally through the main condensing space for the steam. The steam enters at the top and is forced to flow downwards over the tubes due to the suction of the extraction pump at the bottom.

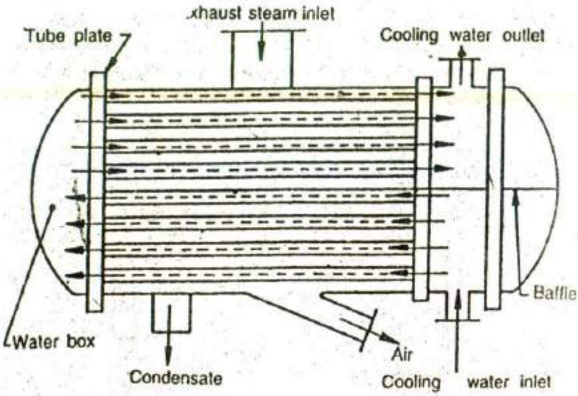


Fig. 20.6. Surface condenser.

The cooling water flows in one direction through the lower half of the tubes and returns in opposite direction through the upper half, as shown in Fig. 20.6.

20.12. Types of Surface Condensers

The surface condensers may be further classified on the basis of the direction of flow of the condensate, the arrangement of tubing system and the position of the extraction pump, into the following four types :

1. Down flow surface condenser, 2. Central flow surface condenser, 3. Regenerative surface condenser, and 4. Evaporative condenser.

These condensers are discussed, in detail, in the following pages.

20.13. Down Flow Surface Condensers

In down flow surface condensers, the exhaust steam enters at the top and flow downwards over the tubes due to force of gravity as well as suction of the extraction pump fitted at the bottom. The condensate is collected at the bottom and then pumped by the extraction pump. The dry air pump suction pipe, which is provided near the bottom, is covered by a baffle so as to prevent the entry of condensed steam into it, as shown in Fig. 20.7.

As the steam flows perpendicular to the direction of flow of cooling water (inside the tubes), this is also called a cross-surface condenser.

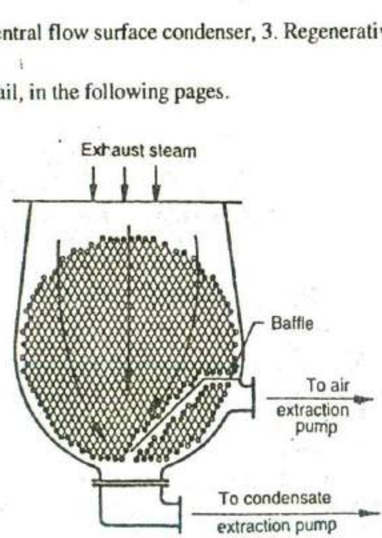


Fig. 20.7. Down flow surface condenser.

20.14. Central Flow Surface Condensers

In central flow surface condensers, the exhaust steam enters at the top and flow downwards. The suction pipe of the air extraction pump is placed in the centre of the tube nest as shown in Fig. 20.8. This causes the steam to flow radially inwards over the tubes towards the suction pipe. The condensate is collected at the bottom and then pumped by the extraction pump as shown in Fig. 20.8.

The central flow surface condenser is an improvement over the down flow type as the steam is directed radially inwards by a volute casing around the tube nest. It, thus, gives an access to the whole periphery of the tubes.

Fig. 20.8. Central flow surface condenser.

20.15. Regenerative Surface Condensers

In regenerative surface condensers, the condensate is heated by a regenerative method. The condensate after leaving the tubes is passed through the exhaust steam from the engine or turbine. It thus, raises its temperature for use as feed water for the boiler.

20.16. Evaporative Condenser

The steam to be condensed enters at the top of a series of pipes outside of which a film of cold water is falling. At the same time, a current of air circulates over the water film, causing rapid evaporation of some of the cooling water. As a result of this, the steam circulating inside the pipe is condensed. The remaining cooling water is collected at an increased temperature and is reused. Its original temperature is restored by the addition of the requisite quantity of cold water.

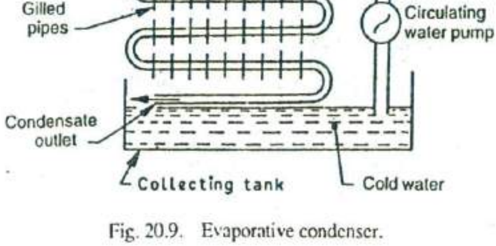


Fig. 20.9. Evaporative condenser.

The evaporative condensers are provided when the circulating water is to be used again and again. These condensers consist of sheets of gilled piping, which is bent backwards and forwards and placed in a vertical plane, as shown in Fig. 20.9.

20.17. Comparison of Jet and Surface Condensers

Following are the important points of comparison between jet and surface condensers :

S. No.	Jet condensers	Surface condensers
1.	Cooling water and steam are mixed up.	Cooling water and steam are not mixed up.
2.	Less suitable for high capacity plants.	More suitable for high capacity plants.
3.	Condensate is wasted.	Condensate is reused.
4.	It requires less quantity of circulating water.	It requires a large quantity of circulating water.
5.	The condensing plant is economical and simple.	The condensing plant is costly and complicated.
6.	Its maintenance cost is low.	Its maintenance cost is high.
7.	More power is required for air pump.	Less power is required for air pump.
8.	High power is required for water pumping.	Less power is required for water pumping.

20.18. Mixture of Air and Steam (Dalton's Law of Partial Pressures)

It states "*The pressure of the mixture of air and steam is equal to the sum of the pressures, which each constitutent would exert, if it occupied the same space by itself.*" Mathematically, pressure in the condenser containing mixture of air and steam,

$$p_c = p_a + p_s$$

where p_a = Partial pressure of air, and

p_s = Partial pressure of steam.

Note : In most of the cases, we are required to find partial pressure of air, therefore Dalton's law may also be used as :

$$p_a = p_c - p_s$$

Example 20.1. *The following observations were recorded during a condenser test ;*

Vacuum reading = 700 mm of Hg ; Barometer reading = 760 mm of Hg ; Condensate temperature = 34° C

Find : 1. Partial pressure of air, and 2. Mass of air per m³ of condenser volume.

Solution. Given : Vacuum reading = 700 mm of Hg ; Barometer reading = 760 mm of Hg ; $T = 34° \text{C} = 34 + 273 = 307 \text{ K}$

1. *Partial pressure of air*

We know that pressure in the condenser

$$p_c = \text{Barometer reading} - \text{Vacuum reading}$$

$$= 760 - 700 = 60 \text{ mm of Hg}$$

$$= 60 \times 0.001\,33 = 0.0798 \text{ bar} \quad \ldots (\because 1 \text{ mm of Hg} = 0.001\,33 \text{ bar})$$

From steam tables, corresponding to a temperature of 34° C, we find that pressure of steam,

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

∴ Partial pressure of air,

$$p_a = p_c - p_s = 0.0798 - 0.0532 = 0.0266 \text{ bar} \quad \textbf{Ans.}$$

2. *Mass of air per m^3 of condenser volume*

We know that mass of air per m^3 of condenser volume,

$$m_a = \frac{p_a v}{R T} = \frac{0.0266 \times 10^5 \times 1}{287 \times 307} = 0.03 \text{ kg} \textbf{ Ans.}$$

$$\ldots (p_a \text{ is taken in N/m}^2 \text{ and } R \text{ for air} = 287 \text{ J/kg K})$$

20.19. Measurement of Vacuum in a Condenser

The vacuum, in general may be defined as the difference between the atmospheric pressure and the absolute pressure. In the study of condensers, the vacuum is generally converted to correspond with a standard atmospheric pressure, which is taken as the barometric pressure of 760 mm of mercury. Mathematically, vacuum gauge reading corrected to standard barometer or in other words

$$\text{Corrected vacuum} = 760 - (\text{Barometer reading} - \text{Vacuum gauge reading})$$

Note : We know that

$$\text{Atmospheric pressure} = 760 \text{ mm of Hg} = 1.013 \text{ bar}$$

$$\therefore 1 \text{ mm of Hg} = \frac{1.013}{760} = 0.00133 \text{ bar} = 133 \text{ N/m}^2 \qquad \ldots (\because 1 \text{ bar} = 10^5 \text{ N/m}^2)$$

Example 20.2. *A vacuum gauge fitted to a condenser reads 680 mm of Hg, when the barometer reads 750 mm of Hg. Determine the corrected vacuum in terms of mm of Hg and bar.*

Solution. Given : Vacuum gauge reading = 680 mm of Hg ; Barometer reading = 750 mm of Hg

We know that pressure in the condenser

$$= 750 - 680 = 70 \text{ mm of Hg}$$

and corrected vacuum

$$= 760 - 70 = 690 \text{ mm of Hg} \textbf{ Ans.}$$

$$= 690 \times 0.001\,33 = 0.918 \text{ bar} \textbf{ Ans.}$$

20.20. Vacuum Efficiency

The minimum absolute pressure (also called ideal pressure) at the steam inlet of a condenser is the pressure corresponding to the temperature of the condensed steam. The corresponding vacuum (called ideal vacuum) is the maximum vacuum that can be obtained in a condensing plant, with no air present at that temperature. The pressure in the actual condenser is greater than the ideal pressure by an amount equal to the pressure of air present in the condenser. The ratio of the actual vacuum to the ideal vacuum is known as *vacuum efficiency*. Mathematically, vacuum efficiency,

$$\eta_v = \frac{\text{Actual vacuum}}{\text{Ideal vacuum}}.$$

where Actual vacuum = Barometric pressure – Actual pressure

and Ideal vacuum = Barometric pressure – Ideal pressure

Example 20.3. *Calculate the vacuum efficiency from the following data :*

Vacuum at steam inlet to condenser = 700 mm of Hg ; Barometer reading = 760 mm of Hg ; Hot well temperature = 30° C.

Solution. Given : Vacuum reading or actual vacuum = 700 mm of Hg ; Barometer reading = 760 mm of Hg ; $t = 30°$ C

We know that pressure in the condenser

$$= 760 - 700 = 60 \text{ mm of Hg}$$

From steam tables, corresponding to a temperature of 30° C, we find that ideal pressure of steam,

$$= 0.0424 \text{ bar} = \frac{0.0424}{0.001\,33} = 31.88 \text{ mm of Hg}$$

We know that ideal vacuum = Barometer reading – Ideal pressure

$$= 760 - 31.88 = 728.12 \text{ mm of Hg}$$

$\therefore$ Vacuum efficiency, $\eta_v = \dfrac{\text{Actual vacuum}}{\text{Ideal vacuum}} = \dfrac{700}{728.12} = 0.9614$ or $96.14\ \%$ **Ans.**

Example 20.4. *The vacuum efficiency of a condenser is 96%. The temperature of condensate is 40° C. If the barometer reads 752 mm of Hg, find the vacuum gauge reading of the condenser.*

Solution. Given : $\eta_v = 96\% = 0.96$; $t = 40°$ C ; Barometer reading = 752 mm of Hg

From steam tables, corresponding to a temperature of 40° C, we find that ideal pressure of steam,

$$= 0.0737 \text{ bar} = \frac{0.0737}{0.001\ 33} = 55.4 \text{ mm of Hg}$$

$\therefore$ Ideal vacuum = Barometer reading – Ideal pressure

$$= 752 \quad 55.4 = 696.6 \text{ mm of Hg}$$

We know that vacuum efficiency (η_v),

$$0.96 = \frac{\text{Actual vacuum}}{\text{Ideal vacuum}} = \frac{\text{Actual vacuum}}{696.6}$$

$\therefore$ Actual vacuum or vacuum gauge reading of the condenser

$$= 0.96 \times 696.6 = 668.74 \text{ mm of Hg} \textbf{ Ans.}$$

Example 20.5. *In a surface condenser, the vacuum maintained is 700 mm of Hg. The barometer reads 754 mm. If the temperature of condensate is 18° C, determine : 1. mass of air per kg of steam ; and 2. vacuum efficiency.*

Solution. Given : Actual vacuum = 700 mm of Hg ; Barometer reading = 754 mm of Hg ; $T = 18°\text{C} = 18 + 273 = 291$ K

We know that pressure in the condenser,

$$p_c = 754 - 700 = 54 \text{ mm of Hg}$$

From steam tables, corresponding to 18° C, we find that absolute or ideal pressure of steam,

$$p_s = 0.0206 \text{ bar} = \frac{0.0206}{0.001\ 33} = 15.5 \text{ mm of Hg}$$

and specific volume of steam, $v_s = 65.09 \text{ m}^3/\text{kg}$

Mass of air per kg of steam

We know that pressure of air (as per Dalton's law),

$$p_a = p_c - p_s = 54 - 15.5 = 38.5 \text{ mm of Hg}$$

$$= 38.5 \times 0.001\ 33 = 0.0512 \text{ bar} = 0.0512 \times 10^5 \text{ N/m}^2$$

and mass of air per kg of steam,

$$m_a = \frac{p_a v}{R\,T} = \frac{0.0512 \times 10^5 \times 65.09}{287 \times 291} = 4 \text{ kg } \textbf{Ans.} \quad \ldots (\because pv = mRT)$$

Vacuum efficiency

We know that ideal vacuum = Barometer reading – Ideal pressure

$$= 754 - 15.5 = 738.5 \text{ mm of Hg}$$

and vacuum efficiency, $\eta_v = \dfrac{\text{Actual vacuum}}{\text{Ideal vacuum}} = \dfrac{700}{738.5} = 0.948$ or 94.8% **Ans.**

Example 20.6. *A surface condenser fitted with separate air and water extraction pumps, has a portion of the tubes near the air pump suction screened off from the steam so that the air is cooled below the condensate temperature. The steam enters the condenser at 38° C and the condensate is removed at 37° C. The air removed has a temperature of 36° C. If the total air infiltration from all sources together is 5 kg/h, determine the volume of air handled by the air pump per hour. What would be the corresponding value of the air handled if a combined air and condensate pump was employed ? Assume uniform pressure in the condenser.*

Solution. Given : $T_s = 38°\text{C} = 38 + 273 = 311\text{ K}$; $T_c = 37°\text{C} = 37 + 273 = 310\text{ K}$; $T_a = 36°\text{C} = 36 + 273 = 309\text{ K}$; $m_a = 5\text{ kg/h}$

1. *Volume of air handled by the air pump per hour*

Since the pressure at entry to the condenser (p_c) is equal to the pressure of steam corresponding to 38° C, therefore from steam tables,

$$p_c = 0.0662\text{ bar}$$

and pressure of steam at the air pump suction, corresponding to 36° C (from steam tables),

$$p_s = 0.0594\text{ bar}$$

∴ Pressure of air at the air pump suction (as per Dalton's law),

$$p_a = p_c - p_s = 0.0662 - 0.0594 \doteq 0.0068\text{ bar}$$

$$= 0.0068 \times 10^5 = 680\text{ N/m}^2$$

We know that volume of air handled by the air pump,

$$v_a = \frac{m_a R T_a}{p_a} = \frac{5 \times 287 \times 309}{680} = 652\text{ m}^3\text{/h} \textbf{ Ans.}$$

2. *Volume of air handled when a combined air and condensate pump is employed*

From steam tables, corresponding to a condensate temperature of 37° C, we find that pressure of steam,

$$p_s = 0.0627\text{ bar}$$

∴ Pressure of air (as per Dalton's law),

$$p_a = p_c - p_s = 0.0662 - 0.0627 = 0.0035\text{ bar}$$

$$= 0.0035 \times 10^5 = 350\text{ N/m}^2$$

We know that volume of air handled,

$$v_a = \frac{m_a R T_c}{p_a} = \frac{5 \times 287 \times 310}{350} = 1271\text{ m}^3\text{/h} \textbf{ Ans.}$$

Example 20.7. *The air leakage into a surface condenser operating with a steam turbine is estimated as 84 kg/h. The vacuum near the inlet of air pump is 700 mm of Hg when barometer reads 760 mm of Hg. The temperature at inlet of vacuum pump is 20° C. Calculate :*

1. The minimum capacity of the air pump is m³/h ; 2. The dimensions of the reciprocating air pump to remove the air if it runs at 200 r.p.m. Take L / D ratio = 1.5 and volumetric efficiency = 100 percent ; and 3. The mass of vapour extracted per minute.

Solution. Given : $m_a = 84\text{ kg/h}$; Vacuum = 700 mm of Hg ; Barometer reading = 760 mm of Hg ; $T = 20°\text{C} = 20 + 273 = 293\text{ K}$

1. *Minimum capacity of the air pump*

We know that pressure in the condenser,

$$p_c = \text{Barometer reading} - \text{Condenser Vacuum}$$

$$= 760 - 700 = 60 \text{ mm of Hg} = 60 \times 0.001\,33 = 0.0798 \text{ bar}$$

From steam tables, corresponding to a temperature of 20° C, we find that pressure of steam,

$$p_s = 0.0234 \text{ bar}$$

∴ Pressure of air (as per Dalton's law),

$$p_a = p_c - p_s = 0.0798 - 0.0234 = 0.0564 \text{ bar}$$

$$= 0.0564 \times 10^5 = 5640 \text{ N/m}^2$$

We know that minimum capacity of the air pump,

$$v_a = \frac{m_a R T}{p_a} = \frac{84 \times 287 \times 293}{5640} = 1252.4 \text{ m}^3\text{/h} \textbf{ Ans.}$$

2 *Dimensions of the reciprocating pump*

Let D = Diameter of the cylinder is metres,

L = Length of the stroke in metres = $1.5\,D$ ... (Given)

η_v = Volumetric efficiency = 100% = 1

N = Speed of the pump = 200 r.p.m. ... (Given)

We know that minimum capacity of the air pump (v_a),

$$\frac{1252.4}{60} = \frac{\pi}{4} \times D^2 \times L \times N = \frac{\pi}{4} \times D^2 \times 1.5\,D \times 200 = 235.6\,D^3$$

... (v_a is taken in m³/min)

∴ $D^3 = 0.0886$ or $D = 0.446$ m **Ans.**

and $L = 1.5\,D = 1.5 \times 0.446 = 0.669$ m **Ans.**

3. *Mass of vapour extracted per minute*

From steam tables, corresponding to a temperature of 20° C, we find that specific volume of steam,

$$v_g = 57.84 \text{ m}^3\text{/kg}$$

∴ Mass of vapour extracted per minute

$$= \frac{v_a}{v_g} = \frac{1252.4}{60 \times 57.84} = 0.361 \text{ kg/min} \textbf{ Ans.}$$

Example 20.8. *The vacuum at the extraction pipe in a condenser is 710 mm of mercury and the temperature is 35.82° C. The barometer reads 760 mm of mercury. The air leakage into the condenser is 4 kg per 10 000 kg of steam. Determine : 1. the volume of air to be dealt with by the dry air pump per kg of steam entering the condenser, and 2. the mass of water vapour associated with this air.*

Take R = 287 J/kg K for air.

Solution. Given : Vacuum = 710 mm of Hg ; T = 35.82° C = 35.82 + 273 = 308.82 K ; Barometer reading = 760 mm of Hg ; m_a = 4 kg per 10 000 kg of steam = 0.0004 kg / kg of steam

1. *Volume of air per kg of steam entering the condenser*

Let v_a = Volume of air per kg of steam entering the condenser.

We know that pressure in the condenser

$$p_c = \text{Barometer reading} - \text{Condenser vacuum}$$

$$= 760 - 710 = 50 \text{ mm of Hg} = 50 \times 0.001\,33 = 0.0665 \text{ bar}$$

From steam tables, corresponding to the temperature of 35.82° C, we find that the pressure of steam,

$$p_s = 0.0588 \text{ bar}$$

∴ Pressure of air (as per Dalton's law),

$$p_a = p_c - p_s = 0.0665 - 0.0588 = 0.0077 \text{ bar}$$

$$= 0.0077 \times 10^5 = 770 \text{ N/m}^2$$

We know that $p_a v_a = m_a R T$

$$\therefore \quad v_a = \frac{m_a R T}{p_a} = \frac{0.0004 \times 287 \times 308.82}{770} = 0.046 \text{ m}^3\text{/kg of steam} \textbf{ Ans.}$$

2. *Mass of water vapour associated with this air*

From steam tables, corresponding to a temperature of 35.82° C, we find that specific volume of steam,

$$v_g = 24.2 \text{ m}^3\text{/kg}$$

∴ Mass of water vapour associated with the air

$$= \frac{v_a}{v_g} = \frac{0.046}{24.2} = 0.0019 \text{ kg} \textbf{ Ans.}$$

Example 20.9. *The air entering a steam condenser with steam is estimated at 6 kg per hour. The temperature at inlet to air cooler section is 30° C and at the outlet 26° C. The vacuum in the shell is essentially constant throughout and is 721 mm of Hg, while the barometer reads 758 mm of Hg. Calculate :*

1. *The volume of air entering the cooling section per hour ;*
2. *The mass of moisture contained in the air ; and*
3. *The mass of steam condensed per hour in the cooling section.*

Solution. Given : $m_a = 6$ kg/h ; $T_1 = 30°\text{ C} = 30 + 273 = 303$ K ; $T_2 = 26°\text{ C} = 26 + 273 = 299$ K ; Condenser vacuum = 721 mm of Hg ; Barometer reading = 758 mm of Hg

We know that pressure in the condenser,

$$p_c = \text{Barometer reading} - \text{Condenser vacuum}$$

$$= 758 - 721 = 37 \text{ mm of Hg} = 37 \times 0.001\,33 = 0.0492 \text{ bar}$$

Considering the inlet to the cooling section. From steam tables, corresponding to a temperature of 30° C, we find that pressure of steam,

$$p_s = 0.0424 \text{ bar}$$

∴ Pressure of air (as per Dalton's law),

$$p_a = p_c - p_s = 0.0492 - 0.0424 = 0.0068 \text{ bar}$$

$$= 0.0068 \times 10^5 = 680 \text{ N/m}^2$$

1. *Volume of air entering the cooling section per hour*

We know that volume of air entering the cooling section,

$$v_a = \frac{m_a R T_1}{p_a} = \frac{6 \times 287 \times 303}{680} = 767 \text{ m}^3/\text{h} \text{ Ans.}$$

2. *Mass of moisture in the air*

From steam tables, corresponding to a temperature of 30° C (or pressure 0.0424 bar), we find that specific volume of steam

$$v_g = 32.93 \text{ m}^3/\text{kg}$$

∴ Mass of moisture or steam associated with the air,

$$m_1 = \frac{v_a}{v_g} = \frac{767}{32.93} = 23.3 \text{ kg/h} \text{ Ans.}$$

3. *Mass of steam condensed per hour in the cooling section*

Now considering the outlet of the cooling section. From steam tables, corresponding to a temperature of 26° C, we find that pressure of steam,

$$p_s = 0.0336 \text{ bar}$$

∴ Pressure of air (as per Dalton's law),

$$p_a = p_c - p_s = 0.0492 - 0.0336 = 0.0156 \text{ bar}$$

$$= 0.0156 \times 10^5 = 1560 \text{ N/m}^2$$

We know that volume of air at outlet to the cooling section,

$$v_a = \frac{m_a R T_2}{p_a} = \frac{6 \times 287 \times 299}{1560} = 330 \text{ m}^3/\text{h}$$

From steam tables, corresponding to a temperature of 26° C (or pressure 0.0336 bar), we find that specific volume of steam,

$$v_g = 41.034 \text{ m}^3/\text{kg}$$

and mass of steam associated with the air,

$$m_2 = \frac{v_a}{v_g} = \frac{330}{41.034} = 8.04 \text{ kg/h}$$

∴ Mass of steam condensed

$$= m_1 - m_2 = 23.3 - 8.04 = 15.26 \text{ kg/h} \text{ Ans.}$$

20.21. Condenser Efficiency

The condenser efficiency may be *defined as the ratio of temperature rise of cooling water to the vacuum temperature *minus* inlet cooling water temperature. Mathematically, condenser efficiency,

$$\eta_c = \frac{\text{Temperature rise of cooling water}}{\text{Vacuum temperature} - \text{Inlet cooling water temperature}}$$

$$= \frac{t_o - t_i}{t_v - t_i}$$

* This definition was proposed by M/s C.A. Parsons & Co., well known manufacturers of steam turbines. Today this definition is widely used.

where t_o = Outlet temperature of cooling water,

t_i = Inlet temperature of cooling water,

t_v = Vacuum temperature. It is the saturation temperature corresponding to the condenser pressure.

Example 20.10. *The inlet and outlet temperatures of cooling water in a condenser are 27° C and 35° C respectively. If the vacuum in the condenser is 700 mm of Hg against barometric pressure of 760 mm of Hg, calculate the efficiency of the condenser.*

Solution. Given : $t_i = 27°$ C ; $t_0 = 35°$ C ; Condenser vacuum = 700 mm of Hg ; Barometric pressure = 760 mm of Hg

We know that pressure in the condenser,

$$= 760 - 700 = 60 \text{ mm of Hg}$$

$$= 60 \times 0.001\,33 = 0.0798 \text{ bar}$$

From steam tables, corresponding to a pressure of 0.0798 bar, we find that vacuum temperature,

$$t_v = 41.5° \text{C}$$

∴ Condenser efficiency,

$$\eta_c = \frac{t_o - t_i}{t_v - t_i} = \frac{35 - 27}{41.5 - 27} = 0.552 \text{ or } 55.2\,\% \text{ Ans.}$$

Example 20.11. *The following data were obtained from the test of a surface condenser :*

Condenser vacuum = 711 mm of Hg ; Hot well temperature = 32° C ; Inlet temperature of circulated water = 12° C ; Outlet temperature of circulated water = 28° C ; Barometer reading = 760 mm of Hg.

Compute the vacuum efficiency and the efficiency of the condenser.

Solution. Given : Condenser vacuum or actual vacuum = 711 mm of Hg ; $t_v = 32°$ C ; $t_i = 12°$ C ; $t_o = 28°$ C ; Barometer reading = 760 mm of Hg

Vacuum efficiency

We know that pressure in the condenser,

$$p_c = 760 - 711 = 49 \text{ mm of Hg}$$

From steam tables, corresponding to a temperature of 32° C, we find that ideal pressure of steam

$$= 0.0475 \text{ bar} = \frac{0.0475}{0.001\,33} = 35.7 \text{ mm of Hg}$$

We know that ideal vacuum

$$= \text{Barometer reading} - \text{Ideal pressure}$$

$$= 760 - 35.7 = 724.3 \text{ mm of Hg}$$

∴ Vacuum efficiency,

$$\eta_v = \frac{\text{Actual vacuum}}{\text{Ideal vacuum}} = \frac{711}{724.3} = 0.9816 \text{ or } 98.16\,\% \text{ Ans.}$$

Condenser efficiency

We know that condenser efficiency,

$$\eta_c = \frac{t_o - t_i}{t_v - t_i} = \frac{28 - 12}{32 - 12} = 0.8 \quad \text{or} \quad 80\% \ \textbf{Ans.}$$

20.22. Mass of Cooling Water Required for Condensation of Steam

In the previous articles, we have discussed various types of condensers and their working. Now we shall discuss the amount of cooling water required for the condensation of steam.

Let m_w = Mass of cooling water,

m_s = Mass of steam condensed (*i.e.* condensate),

h = Total heat of steam entering the condenser,

h_{f1} = Total heat in condensate.

t_i = Inlet temperature of circulating water, and

t_o = Outlet temperature of circulating water.

We know that heat lost by steam

$$= m_s (h - h_{f1})$$

Heat gained by cooling water

$$= m_w c_w (t_o - t_i)$$

We also know that heat gained by cooling water

$$= \text{Heat lost by steam}$$

$$m_w c_w (t_o - t_i) = m_s (h - h_{f1})$$

$$\therefore \quad m_w = \frac{m_s (h - h_{f1})}{c_w (t_o - t_i)}$$

Note : The above equation is applicable to both jet and surface condensers.

Example 20.12. *A surface condenser is designed to handle 10 000 kg of steam per hour. The steam enters at 0.08 bar 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. Given : m_s = 10 000 kg/h ; p = 0.08 bar ; x = 0.9 ; $t_0 - t_i$ = 10° C

From steam tables, corresponding to a pressure of 0.08 bar, we find that

$$h_f = 173.9 \text{ kJ/kg} ; \ h_{fg} = 2403.2 \text{ kJ/kg} ; \text{ and } t = 41.5° \text{ C}$$

$\therefore$ Total heat of the entering steam,

$$h = h_f + x\,h_{fg} = 173.9 + 0.9 \times 2403.2 = 2336.8 \text{ kJ/kg}$$

Since the condensate temperature is equal to the saturation temperature of 41.5° C, therefore heat in condensate corresponding to 41.5° C,

$$h_{f1} = 173.9 \text{ kJ/kg}$$

We know that the cooling water flow rate per hour,

$$m_w = \frac{m_s (h - h_{f1})}{c_w (t_0 - t_i)} = \frac{10\,000\,(2336.8 - 173.9)}{4.2 \times 10} = 514\,980 \text{ kg/h} \ \textbf{Ans.}$$

Example 20.13. *In a condenser test, the following observations were made :*

Vacuum = 690 mm of Hg ; Barometer reading = 750 mm of Hg ; Mean temperature of condensation = 35° C ; Hot well temperature = 28° C ; Mass of cooling water = 50 000 kg/h ; Inlet temperature = 17° C ; Outlet temperature = 30° C ; Mass of condensate per hour = 1250 kg.

Find : 1. The mass of air present per m³ of condenser volume ; 2. The state of steam entering the condenser ; and 3. The vacuum efficiency.

Take R for air = 287 J/kg K.

Solution. Given : Vacuum = 690 mm of Hg ; Barometer reading = 750 mm of Hg ; $t_c = 35°$ C ; $t_h = 28°$ C ; $m_w = 50\,000$ kg/h ; $t_i = 17°$ C ; $t_o = 30°$ C ; $m_s = 1250$ kg/h ; $R = 287$ J/kg K

1. *Mass of air present per m³ of condenser volume*

We know that pressure in the condenser,

$$p_c = 750 - 690 = 60 \text{ mm of Hg} = 60 \times 0.001\,33 = \mathbf{0.08\ bar}$$

From steam tables, corresponding to a condensation temperature of 35° C, we find that the pressure of steam,

$$\therefore\ p_s = 0.0562 \text{ bar}$$

$\therefore$ Pressure of air (as per Dalton's law),

$$p_a = p_c - p_s = 0.08 - 0.0562 = 0.0238 \text{ bar}$$

$$= 0.0238 \times 10^5 = 2380 \text{ N/m}^2$$

We know that mass of air per m³ of condenser volume,

$$m_a = \frac{p_a v}{R\,T} = \frac{2380 \times 1}{287\,(35 + 273)} = 0.027 \text{ kg } \textbf{Ans.}$$

2. *State of steam entering the condenser*

Let x = Dryness fraction (*i.e.* state) of steam entering the condenser.

From steam tables, corresponding to a pressure of 0.0562 bar (or 35° C), we find that

$$h_f = 146.6 \text{ kJ/kg}, \text{ and } h_{fg} = 2418.8 \text{ kJ/kg}$$

and corresponding to a hot well temperature of 28° C,

$$h_{f1} = 117.3 \text{ kJ/kg}$$

We know that total heat of entering steam,

$$h = h_f + x\,h_{fg} = 146.6 + x \times 2418.8 \text{ kJ/kg}$$

We also know that mass of cooling water (m_w),

$$50\,000 = \frac{m_s\,(h - h_{f1})}{c_w\,(t_o - t_i)} = \frac{1250\,[146.6 + x \times 2418.8 - 117.3]}{4.2\,(30 - 17)}$$

$$= 22.894\,(29.3 + x \times 2418.8)$$

or $$29.3 + x \times 2418.8 = 50\,000 / 22.894 = 2184$$

$$x = \frac{2184 - 29.3}{2418.8} = 0.89 \text{ Ans.}$$

3. *Vacuum efficiency*

We know that corresponding to a condensation temperature of 35° C, ideal pressure of steam

$$= 0.0562 \text{ bar} = \frac{0.0562}{0.001\,33} = 42.25 \text{ mm of Hg}$$

$\therefore$ Ideal vacuum = Barometer reading – Ideal pressure

$$= 750 - 42.25 = 707.75 \text{ mm of Hg}$$

We know that vacuum efficiency,

$$\eta_v = \frac{\text{Actual vacuum}}{\text{Ideal vacuum}} = \frac{690}{707.75} = 0.975 \text{ or } 97.5\% \text{ Ans.}$$

Example 20.14. *The following observations were recorded during a test on a steam condenser :*

Barometer reading	*= 765 mm of Hg*
Condenser vacuum	*= 710 mm of Hg*
Mean condenser temperature	*= 35° C*
Condensate temperature	*= 28° C*
Condensate collected per hour	*= 2 tonnes*
Quantity of cooling water per hour	*= 60 tonnes*
Temperature of cooling water at inlet	*= 10° C*
Temperature of cooling water at outlet	*= 25° C*

Find : 1. vacuum corrected to the standard barometer reading ; 2. vacuum efficiency of the condenser ; 3. undercooling of the condensate ; 4. condenser efficiency ; 5. quality of the steam entering the condenser ; 6. mass of air per m^3 of condenser volume ; and 7. mass of air per kg of uncondensed steam.

Solution. Given : Barometer reading = 765 mm of Hg ; Condenser vacuum = 710 mm of Hg ; $T = 35° \text{C} = 35 + 273 = 308$ K ; $t_c = 28°$ C ; $m_s = 2$ t/h = 2000 kg/h ; $m_w = 60$ t/h = 60 000 kg/h ; $t_i = 10°$ C ; $t_o = 25°$ C

1. *Vacuum corrected to the standard barometer reading*

We know that absolute pressure in the condenser

$$= \text{Barometer reading} - \text{Condenser vacuum}$$

$$= 765 - 710 = 55 \text{ mm of Hg}$$

and vacuum corrected to the standard barometer reading (assuming 760 mm of Hg)

$$= 760 - 55 = 705 \text{ mm of Hg} \text{ Ans.}$$

2. *Vacuum efficiency of the condenser*

From steam tables, corresponding to the mean condenser temperature of 35° C, we find that ideal pressure of steam,

$$p_s = 0.0562 \text{ bar} = \frac{0.0562}{0.001\,33} = 42.2 \text{ mm of Hg}$$

We know that ideal vacuum

$$= \text{Barometer pressure} - \text{Ideal pressure}$$

$$= 765 - 42.2 = 722.8 \text{ mm of Hg}$$

and vacuum efficiency, $\eta_v = \dfrac{\text{Actual vacuum}}{\text{Ideal vacuum}} = \dfrac{710}{722.8} = 0.982$ or 98.2% **Ans.**

3. *Undercooling of the condensate*

We know that undercooling of the condensate

$$= \text{Mean condenser temp.} - \text{Condensate temp.}$$

$$= 35 - 28 = 7^\circ \text{ C} \textbf{ Ans.}$$

4. *Condenser efficiency*

We have already found that pressure in the condenser

$$p_c = 765 - 710 = 55 \text{ mm of Hg}$$

$$= 55 \times 0.001\ 33 = 0.073 \text{ bar}$$

From steam tables, corresponding to a pressure of 0.073 bar, we find that vacuum temperature

$$t_v = 39.83^\circ \text{C}$$

$\therefore$ Condenser efficiency,

$$\eta_c = \frac{\text{Temperature rise of cooling water}}{\text{Vacuum temperature} - \text{Inlet cooling temperature}} = \frac{t_o - t_i}{t_v - t_i}$$

$$= \frac{25 - 10}{39.83 - 10} = 0.503 \quad \text{or} \quad 50.3\ \% \textbf{ Ans.}$$

5. *Quality of steam entering the condenser*

Let x = Quality of steam entering the condenser.

From steam tables, corresponding to a pressure of 0.073 bar, we find that

$$h_f = 166.7 \text{ kJ/kg}; \text{ and } h_{fg} = 2407.4 \text{ kJ/kg}$$

and corresponding to a condensate temperature of 28° C, heat in condensate,

$$h_{f1} = 117.3 \text{ kJ/kg}$$

We know that total heat of entering steam,

$$h = h_f + x\, h_{fg} = 166.7 + x \times 2407.4$$

We also know that mass of cooling water (m_w),

$$60\ 000 = \frac{m_s\,(h - h_{f1})}{c_w\,(t_o - t_i)} = \frac{2000\,(166.7 + x \times 2407.4 - 117.3)}{4.2\,(25 - 10)}$$

$$= 31.7\,(49.4 + x \times 2407.4)$$

or $$x \times 2407.4 = \frac{60\ 000}{31.7} - 49.4 = 1843.3$$

$\therefore$ $$x = 0.76 \textbf{ Ans.}$$

6. *Mass of air per m³ of condenser volume*

We know that absolute pressure of air (as per Dalton's law),

$$p_a = p_c - p_s = 0.073 - 0.0562 = 0.0168 \text{ bar}$$

$$= 0.0168 \times 10^5 = 1680 \text{ N/m}^2$$

$\therefore$ Mass of air per m³ of condenser volume,

$$m_a = \frac{p_a\, v}{R\, T} = \frac{1680 \times 1}{287 \times 308} = 0.019 \text{ kg} \textbf{ Ans.} \quad \ldots (\because pv = mRT \text{ and } v = 1 \text{ m}^3)$$

7. *Mass of air per kg of uncondensed steam*

From steam tables, corresponding to 35° C (*i.e.* mean condenser temperature), specific volume of steam,

$$v_g = 25.245 \text{ m}^3/\text{kg}$$

Thus air associated with 1 kg of steam at 35° C will occupy the same volume *i.e.* 25.245 m³.

∴ Mass of air per kg of uncondensed steam,

$$m_a = \frac{p_a v_g}{RT} = \frac{1680 \times 25.245}{287 \times 308} = 0.48 \text{ kg} \ \textbf{Ans.}$$

20.23. Sources of Air into the Condenser

The following are the main sources through which the air may enter into the condenser :

1. The dissolved air in the feed water enters into the boiler, which in turn enters into the condenser with the exhaust steam.
2. The air leaks into the condenser, through various joints, due to high vacuum pressure in the condenser.
3. In case of jet condensers, dissolved air with the injection water enters into the condenser.

20.24. Effects of Air Leakage

Following are the effects of air leakage on the performance of condensing plants :

1. It reduces the vacuum pressure in the condenser.
2. Since air is a poor heat conductor, particularly at low densities, it reduces the rate of heat transmission.
3. It requires a larger air pump. Moreover, an increased power is required to drive the pump.

20.25. Air Pump

The main function of an air pump is to maintain a vacuum in the condenser as nearly as possible, corresponding to exhaust steam temperature. This is done by removing uncondensable air from the condenser. Another common, but not the essential function of the pump, is to remove both air and condensate from the condenser.

The air pump, which extracts both the condensate and air, is called a *wet air pump*. But a pump which extracts only moist air is known as *dry air pump*. The air pumps may be of reciprocating type or rotary type. But here we shall discuss only a reciprocating type air pump (or Edward's air pump) which is commonly used.

20.26. Edward's Air Pump

It is a wet air pump of the reciprocating type. The special features of Edward's air pump is the absence of suction valve and bucket valve, which are necessary in the ordinary reciprocating type air pumps.

The Edward's air pump consists of delivery or head valves, as shown in Fig. 20.10. These valves are placed in the cover which is on the top of the pump barrel lever. The reciprocating piston of the pump is flat on its upper surface and conical at the bottom as shown. The pump lever has a ring of ports around its lower end for the whole circumference. This communicates with the condenser.

When the piston is at the top of the barrel, the condensate and air from the condenser is collected in the conical portion of the lower part of the barrel, through the ports. On the downward stroke of the reciprocating piston , the vacuum is produced above it, since the head valves are closed and sealed by water. The piston uncovers the ports. When it moves downwards, the mixture of condensate, vapour and air rushes into the space above the piston. This mixture is compressed, when the piston goes to the top and raises the pressure slightly above the atmospheric pressure. The head valves are now open, which allow the mixture to pass on the top of the cover. The condensate flows over the weir to the hot well, which is at atmospheric pressure. A relief valve is placed in the base of the cylinder to release the pressure.

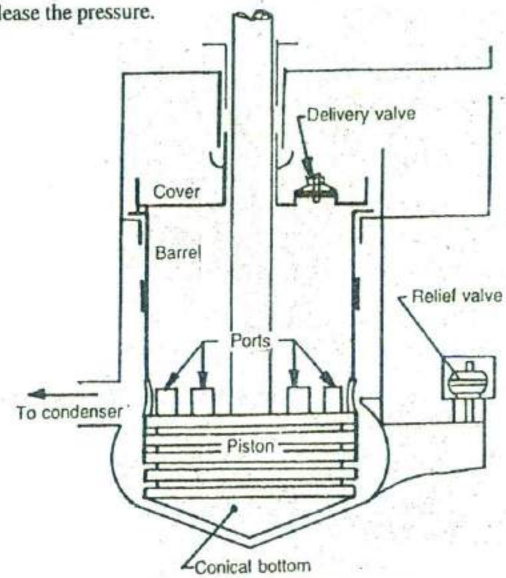


Fig. 20.10. Edward's air pump.

20.27. Cooling Towers

The cooling towers are used in many applications in engineering. The main applications are in power plants and refrigeration plants. Its function is to cool the hot water from the condenser by exposing it to the atmospheric air, so that the cold water may be used again for circulation. The cooling towers are used in steam power plants where there is a limited supply of cooling water. It is placed at a certain height (at about 9 metres from the ground level). The hot water falls down in radial sprays from a height and the atmospheric air enters from the base of the tower. The partial evaporation of water takes place which reduces the temperature of circulating water. This cooled water is collected in the pond at the base of the tower and pumped into the condenser.

Following are some factors which affect the cooling of water in a cooling tower :

1. Size and height of cooling tower,
2. Arrangement of plates in cooling tower,
3. Velocity of air entering the cooling tower,
4. Temperature of air,
5. Humidity of air, and
6. Accessibility of air to all parts of the cooling tower.

20.28. Type of Cooling Towers

The cooling towers may be classified as follows :

1. *According to the type of draught.* The cooling towers, according to the type of draught are (*a*) Natural draught cooling towers, (*b*) Forced draught cooling towers, and (*c*) Induced draught cooling towers.

In a *natural draught cooling tower,* as shown in Fig. 20.11, the circulation of air is produced by the pressure difference of air inside and outside the cooling tower.

In a *forced draught cooling tower*, as shown in Fig. 20.12, the circulation of air is produced by means of fans placed at the base of the tower.

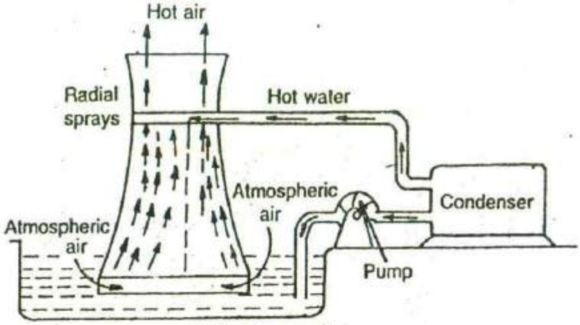


Fig. 20.11. Natural draught cooling tower.

In an *induced draught cooling tower*, as shown in Fig. 20.13, the circulation of air is provided by means of fans placed at the top of the tower.

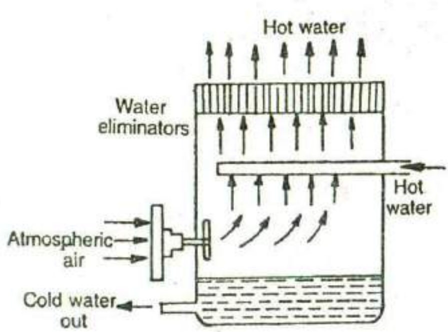
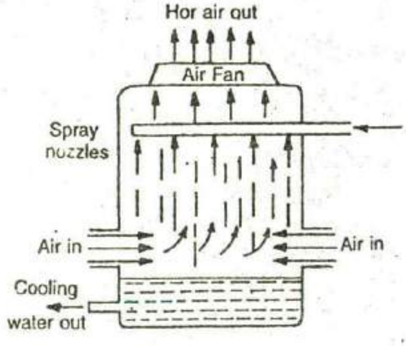


Fig. 20.12. Forced draught cooling tower.



20.13. Induced draught cooling tower.

2. *According to the material used.* The cooling towers, according to the material used are (*a*) Timber cooling towers, (*b*) Concrete (ferro-concrete, multideck concrete hyperbolic) cooling towers, and (*c*) Steel duct type cooling towers.

The timber cooling towers are rarely used due to the following disadvantages :

(*i*) Short life, (*ii*) High maintenance charges, (*iii*) Limited cooling capacity, (*iv*) Rots easily due to exposure to sun, wind, water etc., (*v*) Design generally does not facilitate proper circulation of air.

The concrete cooling towers has the following advantages :

(*i*) Large capacity, (*ii*) Improved draught and air circulation, (*iii*) Increased stability under pressure, (*iv*) Low maintenance.

The duct type cooling towers are rarely used in case of modern power plants due to their small capacity.

EXERCISES

1. The vacuum in a condenser is 665 mm of Hg and the barometer reading is 762 mm of Hg. The temperature of steam at inlet to the condenser is 35° C. Calculate the mass of air entering the condenser per unit volume of the condenser. **[Ans. 0.082 kg/m^3]**

2. The barometer stands at 760 mm and condenser vacuum is at 710 mm and temperature is 30° C. Calculate the mass of air per kg of uncondensed steam. **[Ans.** 0.913 kg]

3. A vacuum gauge fitted to a condenser reads 660 mm of Hg, when the barometer reads 750 mm of Hg. Find the corrected vacuum referred to standard barometer of 760 mm of Hg. **[Ans.** 670 mm of Hg]

4. Calculate the vacuum efficiency of a condenser from the following data : Vacuum at steam inlet to condenser = 710 mm of Hg ; Barometer reading = 760 mm of Hg ; Hot well temperature = 32° C. **[Ans.** 98%]

5. The vacuum gauge on a condenser reads 655 mm of Hg at a barometric pressure of 760 mm of Hg. Steam condenses at 48° C. Find the ratio of mass of water vapour (steam) to air to be dealt by dry air pump. If the air leakage into the condenser were reduced by 50%, find the resulting alteration in vacuum in mm of Hg. **[Ans.** 2.48; 665.6 mm of Hg]

6. A steam condenser has separate air and condensate pumps. The entry to the air pump section is screened. Steam enters the condenser at 38° C and the condensate is removed at 37° C. The air removed has a temperature of 36° C. If the quantity of air infiltration from various sources is 5 kg/h, determine the volume of air handled by the air pump per hour. Compare this with the quantity that would have to be dealt with by using a combined air and condensate pump. Neglect the pressure due to air at the entry of steam and assume uniform pressure in the condenser. **[Ans.** 652 m^3/h ; 1271 m^3/h]

7. The vacuum in the shell of a condenser is 710 mm of Hg and atmospheric pressure is 760 mm of Hg. The temperature at inlet and outlet of the air cooling section are 35° C and 30° C respectively. Calculate for a leakage of 0.5 kg of air per hour : 1. The volume of air entering the cooling section per hour, and 2. The mass of steam condensed per hour in the section. Take R = 294 J/kg K. **[Ans.** 42.9 m^3/h ; 1.152 kg/h]

8. A 110 kW steam engine has a steam consumption of 9.5 kg per kWh. The back pressure of the engine which is approximately the same as the condenser pressure is 0.15 bar. The temperature of condensate is 35° C. The cooling water temperature at inlet and outlet are 18° C and 34° C respectively. Estimate the quantity of cooling water required per hour if the steam exhausted to the condenser is dry **[Ans.** 38140 kg/h]

9. A steam turbine uses 45 000 kg of steam per hour which it exhausts at a dryness fraction 0.9 into a condenser fitted with water extraction and air pumps. With the barometer at 760 mm of mercury, the vacuum at the air pump suction is 716.8 mm and the temperature 32° C. The air leakage is estimated at 1 kg per 1000 kg of steam condensed. Estimate the capacity of the air pump in m^3/min and the quantity of circulating water required in kg/min if the temperature rise is 15° C. **[Ans.** 66.3 m^3/min ; 25 990 kg/min]

10. A surface condenser receives exhaust steam at 0.14 bar from an engine developing 130 kW. The circulating water enters the condenser at 15° C and leaves at 40° C. The final temperature of the condensed steam is 50° C. If the engine consumes 12.25 kg of steam per kW hour, determine the quality of steam entering the condenser, if the mass of circulating water per hour is 31 600 kg. **[Ans.** 0.87]

11. The vacuum in a condenser dealing with 8100 kg of steam per hour is found to be 710 mm of Hg when the barometer reads 750 mm of Hg. The temperature in the condenser is 20° C. The air leakage amounts to 8.1 kg per 1000 kg of steam. Determine the capacity of a suitable dry air pump in m^3 per minute required for the condenser. Take volumetric efficiency of pump as 0.85. **[Ans.** 36.3 m^3/min]

12. The exhaust steam having a dryness fraction of 0.84 enters a surface condenser where the vacuum is 695 mm of Hg and is condensed to water at 35.8° C. The temperature of the hot well is 32.6° C. The circulating water enters the condenser at 15° C and leaves at 35° C. The barometric pressure is 756 mm of Hg. Calculate : 1. The mass of circulating water required per kg of steam, and 2. The mass of air extracted per m^3 of condenser volume. **[Ans.** 24.3 kg/kg of steam ; 0.0364 kg/m^3]

13. A surface condenser receives 15 150 kg of steam per hour after the steam does work in the turbine. Steam at 10 bar and 250° C enters the turbine. The vacuum in the condenser is maintained at 650 mm of Hg and the barometer reads 752 mm of Hg. The rise in temperature of cooling water is limited to 15° C. The temperature of condensate leaving the condenser is 35° C. What is the amount of cooling water required per hour ?

[Ans. 485 470 kg/h]

[**Hint** : First of all, find the dryness fraction of steam entering the condenser by equating the entropy of steam entering the turbine to the entropy of steam leaving the turbine or entering the condenser]

14. A turbine consumes 14 000 kg of steam per hour while developing 2500 kW. Steam is supplied at 30 bar and 300° C. The exhaust from the turbine is condensed in a condenser at a vacuum of 725 mm of Hg and the barometer reads 758 mm of Hg. The condensate is removed from the condenser at a temperature of 28° C. The temperature of cooling water increases from 7° C to 27° C. Assuming no radiation losses, find : 1. The

dryness fraction of steam entering the condenser, and 2. The mass of circulating water per hour.

[Ans. 0.965 ; 391 870 kg/h]

15. The following observations refer to a surface condenser :

Mean temperature of condensation = 34.9° C ; Temperature of hot well = 29.7° C ; Condenser vacuum = 701 mm of Hg ; Barometer = 763 mm of Hg ; Mass of cooling water = 45 500 kg/h ; Inlet temperature of cooling water = 16.5° C ; Outlet temperature of cooling water = 30.6° C ; Mass of condensate = 1180 kg/h.

Find : 1. the mass of air present per m^3 of condenser volume ; 2. the state of steam entering the condenser ; and 3. the vacuum efficiency. [**Ans.** 0.03 kg/m^3 ; 0.935 ; 97.24%]

QUESTIONS

1. What are functions of the condenser in a steam plant ?
2. Describe the principle requirements of a steam condensing plant.
3. Explain the principles of operation of different types of jet condensers. Describe with a sketch a low level jet condenser of the counter flow type.
4. Describe with a neat sketch the working of a surface condenser.
5. Compare the merits and demerits of surface condenser over jet condenser.
6. State Dalton's law of partial pressures.
7. What do you understand by the term vacuum efficiency of a condensing plant ? On what factors does this efficiency depend ?
8. Prove with the help of an example that the vacuum efficiency decreases with the increase in barometric pressure.
9. What are the various sources of air leakage into a steam condenser ? How does it affect the performance of the condensing plant ?
10. Explain the construction and working of Edward's air pump.
11. What part is played by a cooling tower ? What are the different types of cooling towers ? Mention advantage and disadvantage of each type.

OBJECTIVE TYPE QUESTIONS

1. A condenser in a steam power plant

   (*a*) increases expansion ratio of steam (*b*) reduces back pressure of steam

   (*c*) reduces temperature of exhaust steam (*d*) all of these

2. The temperature of condensate is ...... on leaving the condenser than that of circulating water at inlet.

   (*a*) higher (*b*) lower

3. A condenser where circulating water flows through tubes which are surrounded by steam, is known as

   (*a*) surface condenser (*b*) jet condenser

   (*c*) barometric condenser (*d*) evaporative condenser

4. The ratio of actual vacuum to the ideal vacuum in a condenser is called

   (*a*) condenser efficiency (*b*) vacuum efficiency

   (*c*) boiler efficiency (*d*) nozzle efficiency

5. The actual vacuum in a condenser is equal to

   (*a*) barometric pressure + actual pressure (*b*) barometric pressure – actual pressure

   (*c*) gauge pressure + atmospheric pressure (*d*) gauge pressure – atmospheric pressure

ANSWERS

1. (*d*) 2. (*a*) 3. (*a*) 4. (*b*) 5. (*b*)