

**24AG301- APPLIED THERMODYNAMICS AND THERMAL
SYSTEMS IN AGRICULTURAL ENGINEERING**

**UNIT 2 INTERNAL COMBUSTION ENGINES FOR
AGRICULTURE**

HANDLED BY

MS SWETHA S



(iv) Mean effective pressure, p_m :

The mean effective pressure is given by

$$p_m = \frac{W \text{ (work done)}}{V_s \text{ (swept volume)}} = \frac{Q_s - Q_r}{(V_1 - V_2)}$$
$$= \frac{(210 - 106) \times 10^3}{(0.45 - 0.081) \times 10^5}$$
$$= \mathbf{2.818 \text{ bar. (Ans.)}}$$

(v) Power developed, P :

Power developed,

$$P = \text{Work done per second}$$
$$= \text{Work done per cycle}$$
$$\quad \times \text{number of cycles per second}$$
$$= (210 - 106) \times (210/60)$$
$$= \mathbf{364 \text{ kW. (Ans.)}}$$

Diesel cycle

Diesel engines (four-stroke cycle). As is the case of Otto four-stroke; this cycle too is completed in four-strokes as follows. (Refer to Fig.).

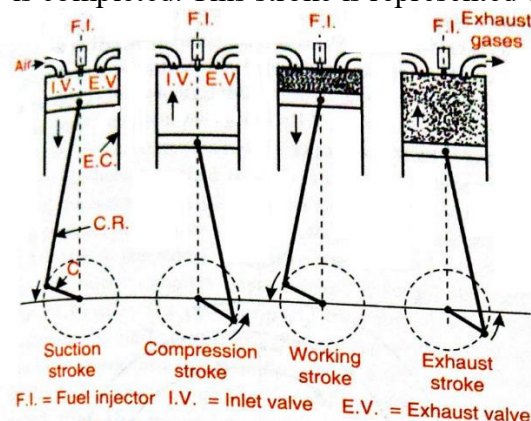
1. Suction stroke. With the movement of the piston from T.D.C. to B.D.C. during this stroke, the inlet valve opens and the air at atmospheric pressure is drawn inside the engine cylinder; the exhaust valve however remains closed. This operation is represented by the line 5-1 (Fig.).

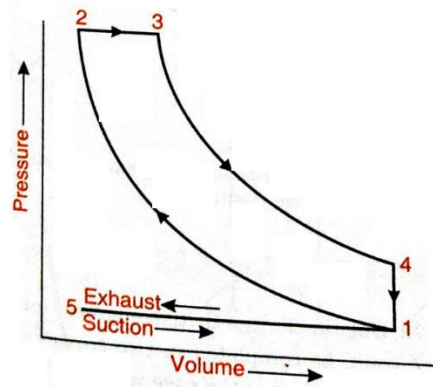
2. Compression stroke. The air drawn at atmos-

Atmospheric pressure during the suction stroke is compressed to high pressure and temperature (to the value of 35 bar and 600°C respectively) as the piston moves from B.D.C. to T.D.C. This operation is represented by 1-2 (Fig.). Both the inlet and exhaust valves do not open during any part of this stroke.

3. Expansion or working stroke. As the piston starts moving from T.D.C. a metered quantity of fuel is injected into the hot compressed air in fine sprays by the Fuel injector and it (fuel) starts burning at constant pressure shown by the line 2-3. At the point 3 fuel supply is cut off. The fuel is injected at the end of compression stroke but in actual practice the ignition of the fuel starts before the end of the compression stroke. The hot gases of the cylinder expand adiabatically to point 4, thus doing work on the piston. The expansion is shown by 3-4.

4. Exhaust stroke. The piston moves from the B.D.C. to T.D.C. and the exhaust gases escape to the atmosphere through the exhaust valve. When the piston reaches the T.D.C. the exhaust valve closes and the cycle is completed. This stroke is represented by the line 1-5 (Fig.)

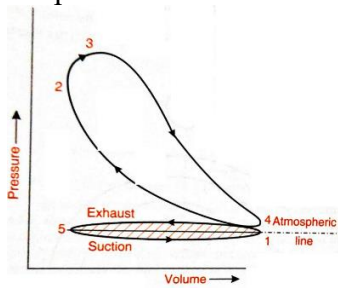




Theoretical p - V diagram of a four-stroke Diesel cycle.

The actual indicator diagram for a four-stroke Diesel cycle engine. It may be noted that line 5-1 is below the atmospheric pressure line. This is due to the fact that owing to the restricted area of the inlet passages the entering air can't cope with the speed of the piston. The exhaust line 4-5 is slightly above the atmospheric line, This is because of the restricted exhaust passages which do not allow the exhaust gases to leave the engine cylinder quickly.

The loop of area 4-5-1 is called negative loop; it gives the pumping loss due to admission of air and removal of exhaust gases. The area 1-2-3-4 is the total or gross work obtained from the piston and net work can be obtained by subtracting area 4-5-1 from area 1-2-3-4.



Actual p - V diagram of four-stroke Diesel cycle.

This cycle comprises of the following *operations*

- (i) 1–2.....*Adiabatic compression.*
- (ii) 2–3.....*Addition of heat at constant pressure.*
- (iii) 3–4.....*Adiabatic expansion.*
- (iv) 4–1.....*Rejection of heat at constant volume.*

Point 1 represents that the cylinder is full of air. Let p_1 , V_1 and T_1 be the corresponding pressure, volume and absolute temperature. The piston then compresses the air adiabatically (*i.e.*, $pV^\gamma = \text{constant}$) till the values become p_2 , V_2 and T_2 respectively (at the end of the stroke) at point 2. Heat is then added from a hot body at a constant pressure. During this addition of heat let volume increases from V_2 to V_3 and temperature T_2 to T_3 , corresponding to point 3.

This point (3) is called the ***point of cut-off***. The air then expands adiabatically to the conditions p_4 , V_4 and T_4 respectively corresponding to point 4. Finally, the air rejects the heat to the cold body at constant volume till the point 1 where it returns to its original state.

$$\begin{aligned}
 \eta_{\text{diesel}} &= \frac{\text{Work done}}{\text{Heat supplied}} \\
 &= \frac{c_p(T_3 - T_2) - c_v(T_4 - T_1)}{c_p(T_3 - T_2)} \\
 &= 1 - \frac{(T_4 - T_1)}{\gamma(T_3 - T_2)} \quad \dots(i)
 \end{aligned}$$

$$\left[\because \frac{c_p}{c_v} = \gamma \right]$$

Let compression ratio, $r = \frac{v_1}{v_2}$, and cut-off ratio,

$$\rho = \frac{v_3}{v_2} \text{ i.e., } \frac{\text{Volume at cut-off}}{\text{Clearance volume}}$$

$$\eta_{\text{diesel}} = 1 - \frac{1}{\gamma(r)^\gamma - 1} \left[\frac{\rho^\gamma - 1}{\rho - 1} \right]$$

Net work

$$= \frac{p_1 v_1 r^{\gamma-1} [\gamma(\rho - 1) - r^{1-\gamma} (\rho^\gamma - 1)]}{(\gamma - 1)}$$

Mean Effective pressure

$$p_m = \frac{p_1(r)^\gamma [\gamma(\rho - 1) - r^{1-\gamma} (\rho^\gamma - 1)]}{(\gamma - 1)(r - 1)}$$

Question

The stroke and cylinder diameter of a compression ignition engine are 250 mm and 150 mm respectively. If the clearance volume is 0.0004 m³ and fuel injection takes place at constant pressure for 5 per cent of the stroke determine the efficiency of the engine. Assume the engine working on the diesel cycle

$$\text{Diameter of cylinder, } D = 150 \text{ mm} = 0.15 \text{ m}$$

$$\text{Clearance volume, } V_2 = 0.0004 \text{ m}^3$$

$$\begin{aligned}
 \text{Swept volume, } V_s &= \pi/4 D^2 L = \pi/4 \times 0.15^2 \times 0.25 \\
 &= 0.004418 \text{ m}^3
 \end{aligned}$$

Total cylinder volume

$$= \text{Swept volume} + \text{clearance volume}$$

$$= 0.004418 + 0.0004 = 0.004818 \text{ m}^3$$

$$\text{Volume at point of cut-off, } V_3 = V_2 + \frac{5}{100} V_s$$

$$= 0.0004 + \frac{5}{100} \times 0.004418 = 0.000621 \text{ m}^3$$

$$\text{Cut-off ratio, } \rho = \frac{V_3}{V_2} = \frac{0.000621}{0.0004} = 1.55$$

Compression ratio,

$$r = \frac{V_1}{V_2} = \frac{V_s + V_2}{V_2} = \frac{0.004418 + 0.0004}{0.0004} = 12.04$$

Hence,

$$\begin{aligned} \eta_{\text{diesel}} &= 1 - \frac{1}{\gamma(r)^{\gamma-1}} \left[\frac{\rho^{\gamma} - 1}{\rho - 1} \right] \\ &= 1 - \frac{1}{1.4 \times (12.04)^{1.4-1}} \left[\frac{(1.55)^{1.4} - 1}{1.55 - 1} \right] \\ &= 1 - 0.264 \times 1.54 = \mathbf{0.593 \text{ or } 59.3\%} \quad (\text{Ans.}) \end{aligned}$$

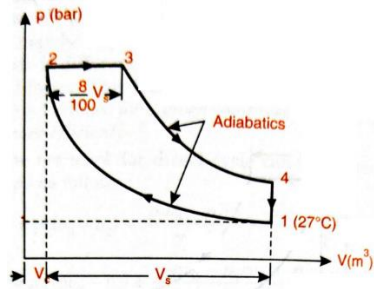
Question

An engine with 200 mm cylinder diameter and 300 mm stroke works on theoretical Diesel cycle. The initial pressure and temperature of air used are 1 bar and 27°C. The cut-off is 8% of the stroke. Determine :

- (i) Pressures and temperatures at all salient points.*
- (ii) Theoretical air standard efficiency.*
- (iii) Mean effective pressure.*
- (iv) Power of the engine if the working cycles per minute are 380.*

Assume that compression ratio is 15 and working fluid is air.

Consider all conditions to be ideal.



Cylinder diameter, $D = 200 \text{ mm or } 0.2 \text{ m}$

Stroke length, $L = 300 \text{ mm or } 0.3 \text{ m}$

Initial pressure, $p_1 = 1.0 \text{ bar}$

Initial temperature, $T_1 = 27 + 273 = 300 \text{ K}$

Cut-off $= \frac{8}{100} V_s = 0.08 V_s$

(i) Pressures and temperatures at salient points :

Now, stroke volume,

$$V_s = \pi/4 D^2 L = \pi/4 \times 0.2^2 \times 0.3 = 0.00942 \text{ m}^3$$

$$V_1 = V_s + V_c = V_s + \frac{V_s}{r-1} \quad \left[\because V_c = \frac{V_s}{r-1} \right]$$

$$= V_s \left(1 + \frac{1}{r-1} \right) = \frac{r}{r-1} \times V_s$$

$$\therefore, V_1 = \frac{15}{15-1} \times V_s = \frac{15}{14} \times 0.00942$$

$$= 0.0101 \text{ m}^3. \quad (\text{Ans.})$$

Mass of the air in the cylinder can be calculated by using the gas equation,

$$p_1 V_1 = m R T_1$$

$$m = \frac{p_1 V_1}{R T_1} = \frac{1 \times 10^5 \times 0.0101}{287 \times 300} = 0.0117 \text{ kg/cycle}$$

For the *adiabatic (or isentropic) process 1-2*

$$p_1 V_1^\gamma = p_2 V_2^\gamma \quad \text{or} \quad \frac{p_2}{p_1} = \left(\frac{V_1}{V_2} \right)^\gamma = (r)^\gamma$$

$$p_2 = p_1 \cdot (r)^\gamma = 1 \times (15)^{1.4} \\ = 44.31 \text{ bar.} \quad (\text{Ans.})$$

$$\text{Also, } \frac{T_2}{T_1} = \left(\frac{V_1}{V_2} \right)^{\gamma-1} = (r)^{\gamma-1} = (15)^{1.4-1} = 2.954$$

$$T_2 = T_1 \times 2.954 = 300 \times 2.954$$

$$= 886.2 \text{ K. (Ans.)}$$

$$V_2 = V_c = \frac{V_s}{r-1} = \frac{0.00942}{15-1}$$

$$= 0.0006728 \text{ m}^3. \text{ (Ans.)}$$

$$p_2 = p_3 = 44.31 \text{ bar. (Ans.)}$$

$$\% \text{ cut-off ratio} = \frac{\rho-1}{r-1}$$

$$\frac{8}{100} = \frac{\rho-1}{15-1}$$

$$\rho = 0.08 \times 14 + 1 = 2.12$$

$$V_3 = \rho V_2 = 2.12 \times 0.0006728$$

$$= 0.001426 \text{ m}^3. \text{ (Ans.)}$$

$$\left[\begin{array}{l} V_3 \text{ can also be calculated as follows:} \\ V_3 = 0.08V_s + V_c = 0.08 \times 0.00942 + 0.0006728 \\ = 0.001426 \text{ m}^3 \end{array} \right]$$

For the constant pressure process 2-3,

$$\frac{V_3}{T_3} = \frac{V_2}{T_2}$$

$$T_3 = T_2 \times \frac{V_3}{V_2} = 886.2 \times \frac{0.001426}{0.0006728}$$

$$= 1878.3 \text{ K. (Ans.)}$$

isentropic process 3-4,

$$p_3 V_3^\gamma = p_4 V_4^\gamma$$

$$p_4 = p_3 \times \left(\frac{V_3}{V_4} \right)^\gamma = p_3 \times \frac{1}{(7.07)^{1.4}}$$

$$= \frac{44.31}{(7.07)^{1.4}} = 2.866 \text{ bar. (Ans.)}$$

$$\left[\begin{aligned} \because \frac{V_4}{V_3} &= \frac{V_4}{V_2} \times \frac{V_2}{V_3} = \frac{V_1}{V_2} \times \frac{V_2}{V_3} \\ &= \frac{r}{\rho}, \quad \because V_4 = V_1 = \frac{15}{2.12} = 7.07 \end{aligned} \right]$$

$$\frac{T_4}{T_3} = \left(\frac{V_3}{V_4} \right)^{\gamma-1} = \left(\frac{1}{7.07} \right)^{1.4-1} = 0.457$$

$$T_4 = T_3 \times 0.457 = 1878.3 \times 0.457$$

$$= \mathbf{858.38 \text{ K. (Ans.)}}$$

$$V_4 = V_1 = \mathbf{0.0101 \text{ m}^3. (Ans.)}$$

(ii) Theoretical air standard efficiency :

$$\begin{aligned} \eta_{\text{diesel}} &= 1 - \frac{1}{\gamma(r)^{\gamma-1}} \left[\frac{\rho^{\gamma} - 1}{\rho - 1} \right] \\ &= 1 - \frac{1}{1.4 (15)^{1.4-1}} \left[\frac{(2.12)^{1.4} - 1}{2.12 - 1} \right] \\ &= 1 - 0.2418 \times 1.663 \\ &= \mathbf{0.598 \text{ or } 59.8\%. (Ans.)} \end{aligned}$$

(iii) Mean effective pressure, p_m :

Mean effective pressure of Diesel cycle is given by

$$\begin{aligned} p_m &= \frac{p_1(r)^{\gamma} [\gamma(\rho - 1) - r^{1-\gamma} (\rho^{\gamma} - 1)]}{(\gamma - 1)(r - 1)} \\ &= \frac{1 \times (15)^{1.4} [1.4(2.12 - 1) - (15)^{1-1.4} (2.12^{1.4} - 1)]}{(1.4 - 1)(15 - 1)} \\ &= \frac{44.31 [1.568 - 0.338 \times 1.863]}{0.4 \times 14} \\ &= \mathbf{7.424 \text{ bar. (Ans.)}} \end{aligned}$$

(iv) Power of the engine, P :

Work done per cycle

$$= p_m V_s = \frac{7.424 \times 10^5 \times 0.00942}{10^3} = \mathbf{6.99 \text{ kJ/cycle}}$$

Work done per second = Work done per cycle × no. of cycles per second

$$= 6.99 \times 380/60 = 44.27 \text{ kJ/s} = 44.27 \text{ kW}$$

Hence power of the engine = 44.27 kW. (Ans.)

PERFORMANCE PARAMETERS

Engine performance is an indication of the degree of success of the engine performs its assigned task, i.e. the conversion of the chemical energy contained in the fuel into the useful mechanical work.

The performance of an engine is evaluated on the basis of the following : (a) Specific Fuel Consumption. (b) Brake Mean Effective Pressure. (c) Specific Power Output. (d) Specific Weight. (e) Exhaust Smoke and Other Emissions.

For the evaluation of an engine performance few more parameters are chosen and the effect of various operating conditions, design concepts and modifications on these parameters are studied. The basic performance parameters are the following : (a) Power and Mechanical Efficiency. (b) Mean Effective Pressure and Torque. (c) Specific Output. (d) Volumetric Efficiency. (e) Fuel-air Ratio. (f) Specific Fuel Consumption. (g) Thermal Efficiency and Heat Balance. (h) Exhaust Smoke and Other Emissions. (i) Specific Weight.

Power and Mechanical Efficiency

The main purpose of running an engine is to obtain mechanical power. • Power is defined as the rate of doing work and is equal to the product of force and linear velocity or the product of torque and angular velocity.

The power developed by an engine and measured at the output shaft is called the brake power (bp) and is given by,

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$$bp = \frac{2\pi NT}{60}$$

where, T is torque in N-m and N is the rotational speed in revolutions per minute.

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Indicated Power

It is the power developed in the cylinder and thus, forms the basis of evaluation of combustion efficiency or the heat release in the cylinder.

If, p_{mi} = Indicated mean effective pressure, bar.

A = Area of piston, m^2 ,

L = Length of stroke, m,

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

$k = \frac{1}{2}$ for 4-stroke engine,

= 1 for 2-stroke engine.

Work done per second = Work done per stroke
 $\times$ number of working stroke per second

$$= p_{mi} \times L \times A \times 10^5 \times \frac{N}{60} \times k \text{ Nm/s or J/s}$$

$$= \frac{p_{mi} \times LANk \times 10^5}{60 \times 1000} \text{ kW}$$

Indicated power,

$$\text{I.P.} = \frac{p_{mi} LANk \times 10}{6} \text{ kW}$$

If n is the number of cylinders, then

$$\text{I.P.} = \frac{np_{mi} LANk \times 10}{6} \text{ kW}$$

TEST CONDUCTED ON ENGINES

1. MORSE TEST

This test is only applicable to multi-cylinder engines. The engine is run at the required speed and the torque is measured. One cylinder is cut out, by shorting the plug if an I.C. engine is under test, or by disconnecting an injector if a C.I. engine is under test. The speed falls because of the loss of power with one cylinder cut out, but is restored by reducing the load. The torque is measured again when the speed has reached its original value. If the values of I.P. of the cylinders are denoted by I_1, I_2, I_3 and I_4 and (considering a four-cylinder engine), and the power losses in each cylinder are denoted by

$$B = (I_1 - L_1) + (I_2 - L_2) + (I_3 - L_3) + (I_4 - L_4)$$

If number 1 cylinder is cut out, then contribution of I_1 is lost; Now B_1 is

$$B_1 = (I_2 - L_2) + (I_3 - L_3) + (I_4 - L_4)$$

Subtracting eqn. (ii) from eqn. (i), we get

$$B - B_1 = I_1$$

Similarly,

$$B - B_2 = I_2 \text{ when cylinder number 2 is cut out,}$$

$$\text{and } B - B_3 = I_3 \text{ when cylinder number 3 is cut out,}$$

$$\text{and } B - B_4 = I_4 \text{ when cylinder number 4 is cut out}$$

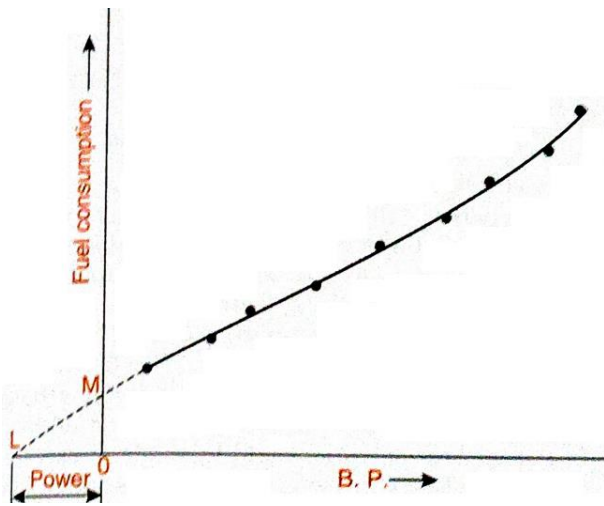
Then, for the engine,

$$I = I_1 + I_2 + I_3 + I_4$$

Test are

2. Willan's line method

At a constant engine speed the load is reduced in increments and the corresponding B.P. and gross fuel consumption readings are taken. A graph is then drawn of fuel consumption against B.P. as in Figure below. The graph drawn is called the Willan's line (analogous to Willan's line for a steam engine), and is extrapolated back to cut the B.P. axis at the point L. The reading OL is taken as the power loss of the engine at that speed. The fuel consumption at zero B.P. is given by OM; and if the relationship between fuel consumption and B.P. is assumed to be linear, then a fuel consumption OM is equivalent to a power loss of OL.



The main drawback of this method is the long distance to be extrapolated from data measured between 5 and 40% load towards the zero line of fuel in put.

- The directional margin of error is rather wide because of the graph which may not be a straight line many times.
- The changing slope along the curve indicates part efficiencies of increments of fuel. The pronounced change in the slope of this line near full load reflects the limiting influence of the air-fuel ratio and of the quality of combustion.
- Similarly, there is a slight curvature at light loads. This is perhaps due to difficulty in injecting accurately and consistently very small quantities of fuel per cycle.
 - Therefore, it is essential that great care should be taken at light loads to establish the true nature of the curve.
- The Willan's line for a swirl-chamber CI engine is straighter than that for a direct injection type engine.
- The accuracy obtained in this method is good and compares favorably with other methods if extrapolation is carefully done.

3. Motoring test

In this test the engine is first run upto the desired speed by its own power and allowed to remain under the given speed and load conditions for sometime so that oil, water and engine component temperatures reach stable conditions. The power of the engine during this period is absorbed by a dynamometer (usually of electrical type). The fuel supply is then cut off and by suitable electric switching devices the dynamometer is converted to run as a motor to drive or 'motor' the engine at the same speed at which it was previously running. The power supply to the motor is measured which is a measure of F.P. of the engine. This method, though determines the fp at temperature conditions very near to the actual operating temperatures at the test speed and load, does, not give the true losses occurring under firing conditions due to the following reasons.

4. Difference between I.P. and B.P.

The method of finding the F.P. by finding the difference between I.P. as obtained from an indicator diagram, and B.P. as obtained by a dynamometer is the ideal method. However, due to difficulties in obtaining accurate indicator diagrams, especially at high engine speeds, this method is usually only used in research laboratories and its use at commercial level is very limited.

The difference between the ip and bp is the indication of the power lost in the mechanical components of the engine (due to friction) and forms the basis of mechanical efficiency; which is defined as follows :

$$\text{Mechanical efficiency} = \frac{bp}{ip}$$

The difference between ip and bp is called friction power (fp).

$$fp = ip - bp$$

$$\therefore \text{Mechanical efficiency} = \frac{bp}{(bp + fp)}$$

Mean Effective Pressure and Torque

Mean effective pressure is defined as a hypothetical/average pressure which is assumed to be acting on the piston throughout the power stroke. Therefore,

$$P_m = \frac{ip \times 60}{LANk}$$

where, P_m = Mean effective pressure, N/m^2 ,

I_p = Indicated power, Watt,

L = Length of the stroke, m,

A = Area of the piston, m^2 ,

N = Rotational speed of the engine, rpm (It is $N/2$ for four stroke engine),
and

k = Number of cylinders.

Thus, the torque and the mean effective pressure are related by the engine size. A large engine produces more torque for the same mean effective pressure.

Specific Output

Specific output of an engine is defined as the brake power (output) per unit of piston displacement and is given by,

$$\text{Specific output} = \frac{bp}{A \times L}$$

$$= \text{Constant} \times bmep \times rpm$$

Volumetric Efficiency

Volumetric efficiency of an engine is an indication of the measure of the degree to which the engine fills its swept volume. It is defined as the ratio of the mass of air inducted into the engine cylinder during the suction stroke to the mass of the air corresponding to the swept volume of the engine at atmospheric pressure and temperature.

Volumetric efficiency, η_v

$$= \frac{\text{Mass of charge actually sucked in}}{\text{Mass of charge corresponding to the cylinder intake } P \text{ and } T \text{ conditions}}$$

Fuel-Air Ratio (F/A)

IC Engine Testing Fuel-air ratio (F/A) is the ratio of the mass of fuel to the mass of air in the fuel-air mixture. Air-fuel ratio (A/F) is reciprocal of fuel-air ratio. Fuel-air ratio of the mixture affects the combustion phenomenon in that it determines the flame propagation velocity, the heat release in the combustion chamber, the maximum temperature and the completeness of combustion.

$$\text{Relative fuel-air ratio, } F_R = \frac{\text{Actual fuel - Air ratio}}{\text{Stoichiometric fuel - Air ratio}}$$

Brake Specific Fuel Consumption

Specific fuel consumption is defined as the amount of fuel consumed for each unit of brake power developed per hour. It is a clear indication of the efficiency with which the engine develops power from fuel

$$\text{Brake specific fuel consumption (bsfc)} = \frac{\text{Actual fuel} - \text{Air ratio}}{\text{Stoichiometric fuel} - \text{Air ratio}} \cdot \cdot$$

Thermal Efficiency and Heat Balance

Thermal efficiency of an engine is defined as the ratio of the output to that of the chemical energy input in the form of fuel supply. It may be based on brake or indicated output. It is the true indication of the efficiency with which the chemical energy of fuel (input) is converted into mechanical work.

$$\text{Brake thermal efficiency} = \frac{bp}{m_f \times C_v}$$

where, C_v = Calorific value of fuel, kJ/kg, and

m_f = Mass of fuel supplied, kg/sec.

ENGINE PERFORMANCE CURVES

The following parameters are of interest (to ineers):

Torque

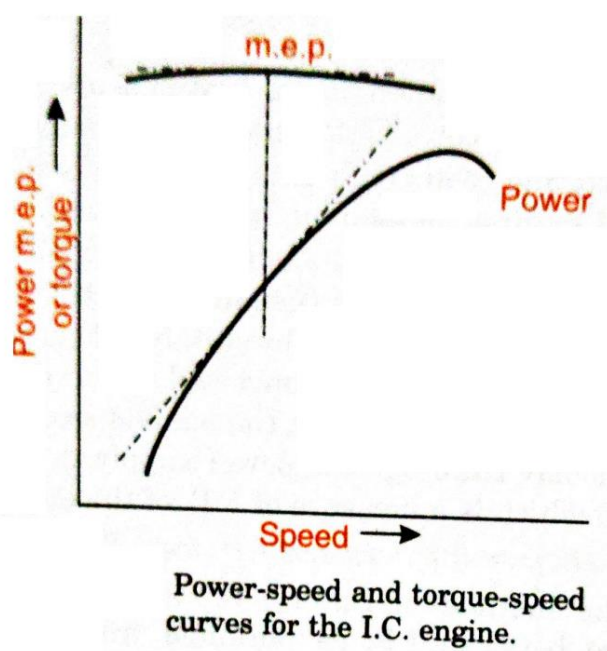
Power

Specific fuel consumption and its inverse

I.C. engine efficiency

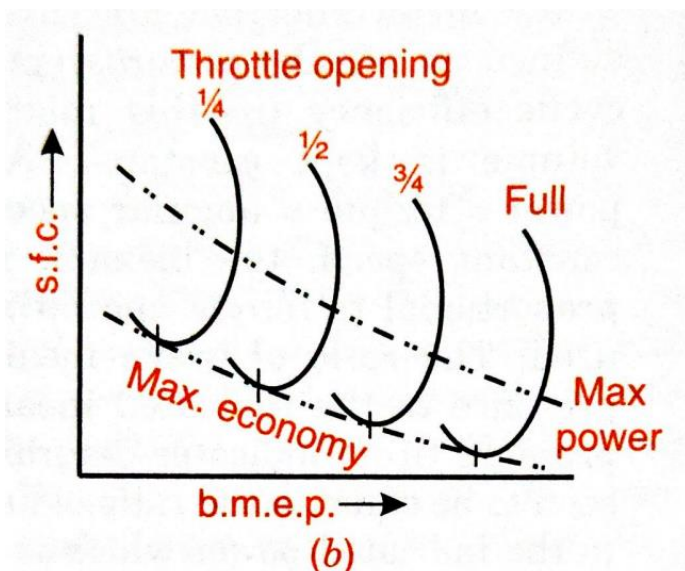
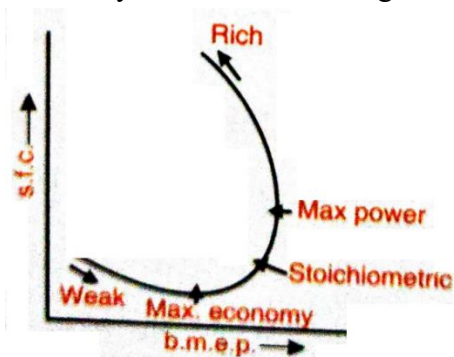
Brake mean effective pressure; this is defined as: Brake power/swept volume x cyclic efficiency (in this relation swept volume is kept constant). Also brake power = torque x angular speed, thus at constant speed, the b.e.m.p. is directly proportional to torque and either may be used. The ratio of brake mean effective pressure to the indicated mean effective pressure (from indicator diagram) may be seen to be equal to the ratio of brake power to the indicated power which is defined as mechanical efficiency. Thus indicator diagram and the output torque may be connected with a suitable allowance for mechanical efficiency.

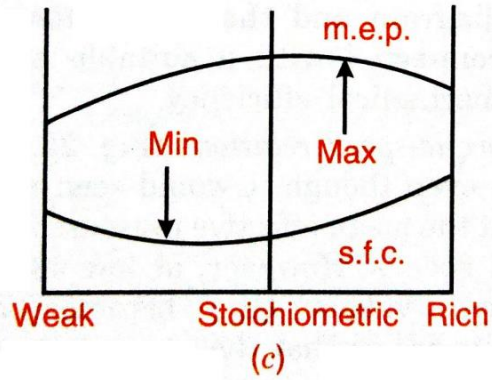
The torque-speed relations (Fig. below) exhibit a curve even though it would seem reasonable to expect the mean effective pressure to be constant at all speeds. However, at low speeds leakage through valves etc., becomes of greater significance so that the m.e.p. falls and at high speeds the volumetric efficiency causes the induced mass to fall with a parallel fall in m.e.p. The power curves are the product of the torque curves with speed.



Specific fuel consumption relations:

The curves are plotted for constant throttle opening, constant speed and constant ignition setting. The only variable is the air-fuel ratio. The effect of ignition setting or speed may be ascertained by producing a new family of curves. An alternative method of plotting these parameters is to use the air-fuel ratio as the abscissa. Here it can be seen that maximum economy occurs with a slightly weak mixture. This means that there is excess air and combustion is complete. Maximum power occurs with a slightly rich mixture when all the available oxygen is used. The I.C. engine efficiency is the inverse of the specific fuel consumption with the constant calorific value as a factor. Thus the curves of specific fuel consumption (s.f.c.) also represent efficiency. The maximum value of brake I.C. engine efficiency for S.I. and C.I. engines are of the order 35% and 40% respectively.

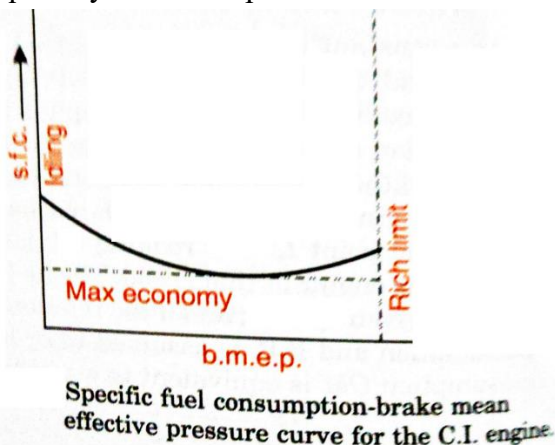




Specific fuel consumption-brake mean effective pressure curves for the S.I. engine.

C.I. engine.

The flat curve of Fig. below illustrates that at part load the compression ignition engine is more economical than the spark ignition engine. This is the benefit of quality control rather than quantity control of power.



Other performance curves

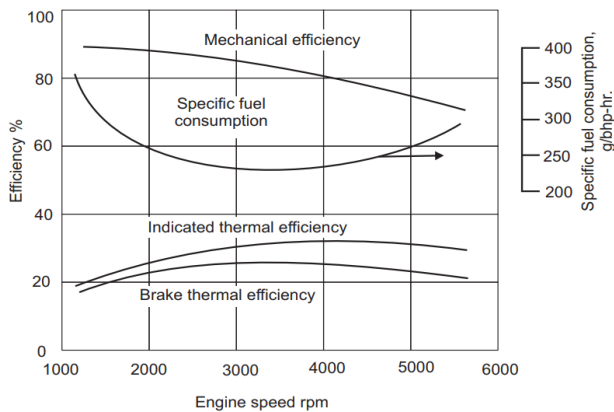
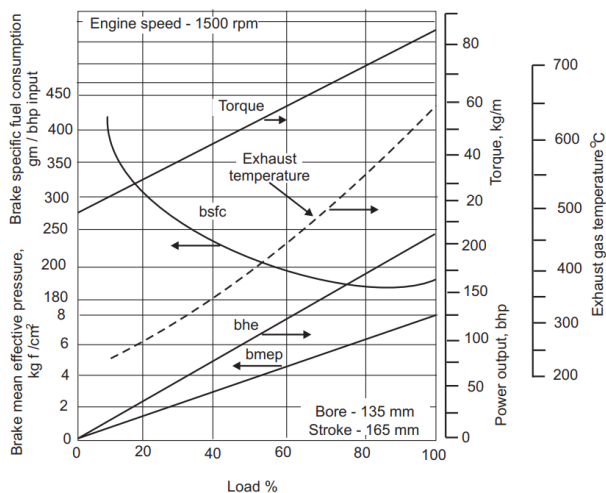


Figure 7.12 : Efficiency and Specific Fuel Consumption Vs.



: Performance Curves of a Six Cylinder Four-stroke Cycle Automotive Type CI Engine at Constant Speed

QUESTIONS

The power output of an I.C. engine is measured by a rope brake dynamometer. The diameter of the brake pulley is 700 mm and the rope diameter is 25 mm. The load on the light side of the rope is 50 kg mass and spring balance reads 50 N. The engine running at 900 r.p.m. consumes fuel of calorific value of 44000 kJ/kg, at a rate of 4 kg/h.

Assume $g = 9.81 \text{ m/s}^2$. Calculate :

- Brake specific fuel consumption
- Brake thermal efficiency.

Solution. Given : $D_b = 700 \text{ mm} = 0.7 \text{ m}$,
 $d = 25 \text{ mm} = 0.025 \text{ m}$,
 $W = 50 \text{ kg}$, $S = 50 \text{ N}$; $N = 900 \text{ r.p.m.}$;
 $C = 44000 \text{ kJ/kg}$, $m_f = 4 \text{ kg/h}$

(i) Brake specific fuel consumption, b.s.f.c

Brake power,

$$\begin{aligned} \text{B.P} &= \frac{(W - S) \pi (D_b + d) \times N}{60 \times 1000} \text{ kW} \\ &= \frac{(50 \times 9.81 - 50) \times \pi (0.7 + 0.025) \times 900}{60 \times 1000} \\ &= 15.05 \text{ kW} \end{aligned}$$

Brake specific fuel consumption,

$$\begin{aligned} \text{b.s.f.c.} &= \frac{m_f (\text{kg/h})}{\text{B.P. (kW)}} \\ &= \frac{4}{15.05} = 0.266 \text{ kg/kWh. (Ans.)} \end{aligned}$$

(ii) Brake thermal efficiency, $\eta_{th(B)}$:

$$\begin{aligned} \eta_{th(B)} &= \frac{\text{B.P.}}{\dot{m}_f \times C} \quad (\text{where } \dot{m}_f = \text{fuel used, in kg/s}) \\ &= \frac{15.05}{(4 / 3600) \times 44000} \\ &= 0.3078 \text{ or } 30.78\%. \quad (\text{Ans.}) \end{aligned}$$

QUESTION

A four cylinder four-stroke S.I. engine has a compression ratio of 8 and bore of 100 mm, with stroke equal to the bore. The volumetric efficiency of each cylinder is equal to 75%. The engine operates at a speed of 4800 r.p.m. with an air-fuel ratio 15.

Given that the calorific value of fuel = 42 MJ/kg, atmospheric density = 1.12 kg/m³, mean effective pressure in the cylinder = 10 bar and mechanical efficiency of the engine = 80%, determine the indicated thermal efficiency and the brake power.

Solution. Given : Number of cylinders, $n = 4$; $k = \frac{1}{2}$ (engine being four-stroke) ; $r = 8$; $D = 100 \text{ mm} = 0.1 \text{ m}$; $L = D = 0.1 \text{ m}$; $\eta_{\text{vol}} = 75\%$; $N = 4800 \text{ r.p.m.}$; Air-fuel ratio = 15 ; $C = 42 \text{ MJ/kg}$; $\rho = 1.12 \text{ kg/m}^3$; $p_{mi} = 10 \text{ bar}$, $\eta_{\text{mech}} = 80\%$.

Indicated thermal efficiency, $\eta_{\text{th. (I)}}$:

Indicated power,

$$\begin{aligned} \text{I.P.} &= \frac{n p_{mi} L A N k \times 10}{6} \\ &= \frac{4 \times 10 \times 0.1 \times \left(\frac{\pi}{4} \times 0.1^2 \right) \times 4800 \times \frac{1}{2} \times 10}{6} \text{ kW} \\ &= 125.66 \text{ kW} \end{aligned}$$

$$\begin{aligned} \text{Air consumption} &= n \times \frac{\pi}{4} D^2 \times L \times \frac{N}{2} \times \eta_{\text{vol.}} \\ &= 4 \times \frac{\pi}{4} \times 0.1^2 \times 0.1 \times \frac{4800}{2} \times 0.75 \\ &= 5.655 \text{ m}^3/\text{min} = 0.09425 \text{ m}^3/\text{s} \end{aligned}$$

$$\text{Mass flow of air, } \dot{m}_a = 0.09425 \times 1.12 = 0.1056 \text{ kg/s}$$

$$\begin{aligned} \text{Fuel consumption, } \dot{m}_f &= \frac{0.1056}{\text{Air-fuel ratio}} = \frac{0.1056}{15} \\ &= 0.00704 \text{ kg/s} \end{aligned}$$

$$\begin{aligned}\therefore \eta_{th(I)} &= \frac{I.P.}{\dot{m}_f \times C} \\ &= \frac{125.66 \times 10^3}{0.00704 \times 42 \times 10^6} \\ &= \mathbf{0.425 \text{ or } 42.5\%} \quad (\text{Ans.})\end{aligned}$$

Brake power, B.P. :

$$\begin{aligned}\text{Brake power} &= \text{Indicated power} \times \eta_{mech.} \\ &= 125.66 \times 0.8 = \mathbf{100.53 \text{ kW.}}\end{aligned}$$

QUESTION

A single-cylinder four-stroke diesel engine running at 1800 r.p.m. has a bore of 85 mm and a stroke of 110 mm. It takes 0.56 kg of air per minute and develops a brake power output of 6 kW while the air-fuel ratio is 20 : 1. The calorific value of the fuel used is 42550 kJ/kg, and the ambient air density is 1.18 kg/m³. Calculate :

- (i) The volumetric efficiency, and
- (ii) Brake specific fuel consumption.

Solution. Given : $N = 1800 \text{ r.p.m.}$; $D = 85 \text{ mm} = 0.085 \text{ m}$;

$$L = 110 \text{ mm} = 0.11 \text{ m} ;$$

$$\text{Air flow rate, } m = 0.56 \text{ kg/min.} ; \text{ B.P.} = 6 \text{ kW} ;$$

$$\text{Air-fuel ratio} = 20 : 1 ;$$

$$C = 42550 \text{ kJ/kg} ; \rho = 1.18 \text{ kg/m}^3.$$

(i) The volumetric efficiency, $\eta_{vol.}$:

$$\begin{aligned}\text{Volume displacement} &= \frac{\pi}{4} D^2 \times L \times \frac{N}{2} \\ &= \frac{\pi}{4} \times (0.085)^2 \times 0.11 \times \frac{1800}{2} \\ &= 0.5617 \text{ m}^3/\text{min}\end{aligned}$$

$$\text{Mass of air} = 0.5617 \times 1.18 = 0.663 \text{ kg/min}$$

$\therefore$ Volumetric efficiency

$$= \frac{0.56}{0.663} = \mathbf{0.845 \text{ or } 84.5\%} \quad (\text{Ans.})$$

(ii) Brake specific fuel consumption (b.s.f.c.)

Fuel consumption

$$= \frac{0.56}{\text{Air-fuel ratio}} = \frac{0.56}{20} = 0.028 \text{ kg/min.}$$

$\therefore$ Brake specific fuel consumption

$$\begin{aligned}&= \frac{\text{Fuel used / hour}}{\text{B.P.}} \text{ kg/kWh} \\ &= \frac{0.028 \times 60}{6} = \mathbf{0.28 \text{ kg/kWh.}} \quad (\text{Ans.})\end{aligned}$$

QUESTION

Following data refer to a four-stroke double-acting diesel engine having cylinder diameter 200 mm and piston stroke 350 mm.

m.e.p. on cover side	= 6.5 bar
m.e.p. on crank side	= 7 bar
Speed	= 420 r.p.m.
Diameter of piston rod	= 20 mm
Dead load on the brake	= 1370 N
Spring balance reading	= 145 N
Brake wheel diameter	= 1.2 m
Brake rope diameter	= 20 mm

Calculate the mechanical efficiency of the engine.

Solution. Given : $P_{mi(cover)} = 6.5$ bar,
 $P_{mi(crank)} = 7$ bar, $D = 0.2$ m,
 $L = 0.35$ m,

$N = 420$ r.p.m., $d_{rod} = 20$ mm = 0.02 m,
 $W = 1370$ N, $S = 145$ N,

$D_b = 1.2$ m, $d = 0.02$ m, $k = \frac{1}{2}$... 4-stroke cycle engine

Mechanical efficiency ; $\eta_{mech.}$:

Area of cylinder on cover end side,

$$A_{cover} = \pi/4 D^2 = (\pi/4) \times (0.2)^2 = 0.03141 \text{ m}^2$$

Effective area of cylinder on crank end side,

$$\begin{aligned} A_{crank} &= \pi/4 (D^2 - d_{rod}^2) \\ &= \pi/4 (0.2^2 - 0.02^2) = 0.0311 \text{ m}^2 \end{aligned}$$

Indicated power on cover end side,

$$\begin{aligned} \text{I.P.}_{(\text{cover})} &= \frac{p_{mi(\text{cover})} \times L A N k \times 10}{6} \\ &= \frac{6.5 \times 0.35 \times 0.03141 \times 420 \times \frac{1}{2} \times 10}{6} \\ &= 25 \text{ kW} \end{aligned}$$

Indicated power on crank end side,

$$\begin{aligned} \text{I.P.}_{(\text{crank})} &= \frac{p_{mi(\text{crank})} \times L A N k \times 10}{6} \\ &= \frac{7 \times 0.35 \times 0.0311 \times 420 \times \frac{1}{2} \times 10}{6} \\ &= 26.67 \text{ kW} \end{aligned}$$

$$\text{Total I.P.} = 25 + 26.67 = 51.67 \text{ kW}$$

Now, brake power,

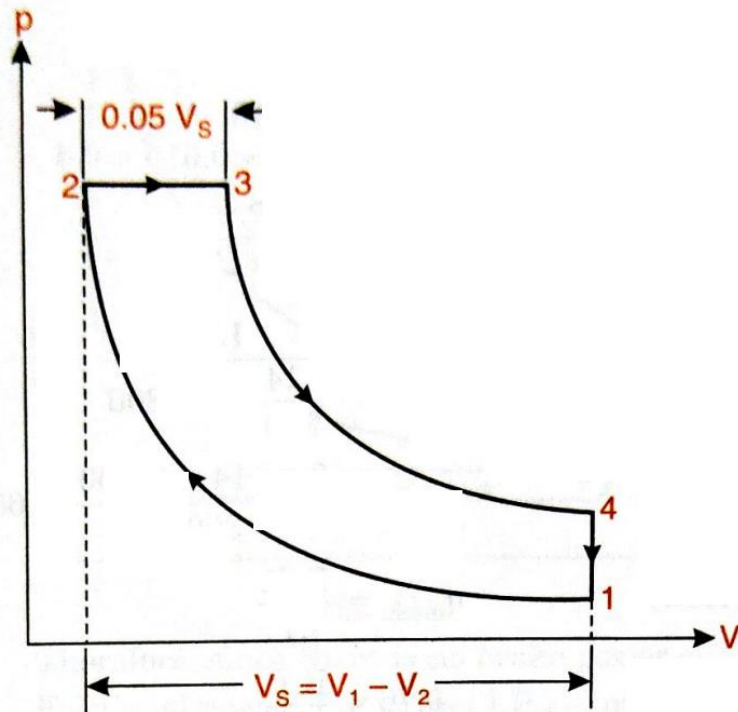
$$\begin{aligned} \text{B.P.} &= \frac{(W - S) \pi (D_b + d) N}{60 \times 1000} \\ &= \frac{(1370 - 145) \pi (1.2 + 0.02) \times 420}{60 \times 1000} \text{ kW} \\ &= 32.86 \text{ kW} \end{aligned}$$

Mechanical efficiency,

$$\eta_{\text{mech.}} = \frac{\text{B.P.}}{\text{I.P.}} = \frac{32.86}{51.67} = 0.6359 = \mathbf{63.59\%} \quad (\text{Ans.})$$

QUESTION

A 2-cylinder C.I. engine with a compression ratio 13 : 1 and cylinder dimensions of 200 mm × 250 mm works on two-stroke cycle and consumes 14 kg/h of fuel while running at 300 r.p.m. The relative and mechanical efficiencies of engine are 65% and 76% respectively. The fuel injection is effected upto 5% of stroke. If the calorific value of the fuel used is given as 41800 kJ/kg, calculate the mean effective pressure developed.



Diameter of cylinder, $D = 200 \text{ mm} = 0.2 \text{ m}$
 Stroke length, $L = 250 \text{ mm} = 0.25 \text{ m}$
 Number of cylinders, $n = 2$
 Compression ratio, $r = 13$
 Fuel consumption $= 14 \text{ kg/h}$
 Engine speed, $N = 300 \text{ r.p.m.}$
 Relative efficiency, $\eta_{\text{relative}} = 65\%$
 Mechanical efficiency, $\eta_{\text{mech.}} = 76\%$
 Cut-off $= 5\% \text{ of stroke}$
 Calorific value of fuel, $C = 41800 \text{ kJ/kg}$
 $k = 1$ for two-stroke cycle engine

Cut-off ratio, $\rho = \frac{V_3}{V_2}$

$$V_3 - V_2 = 0.05V_s = 0.05(V_1 - V_2)$$

$$V_3 - V_2 = 0.05 (13V_2 - V_2)$$

$$\left[\because \frac{V_1}{V_2} = r = 13 \right]$$

$$V_3 - V_2 = 0.05 \times 12V_2 = 0.6V_2.$$

$$\frac{V_3}{V_2} = \rho = 1.6$$

$$\begin{aligned}
 \eta_{\text{air-standard}} &= 1 - \frac{1}{\gamma(r)^{\gamma-1}} \left[\frac{\rho^\gamma - 1}{\rho - 1} \right] \\
 &= 1 - \frac{1}{1.4(14)^{1.4-1}} \left[\frac{1.6^{1.4} - 1}{1.6 - 1} \right] \\
 &= 1 - 0.248 \times 1.55 \\
 &= 0.615 \text{ or } 61.5\%
 \end{aligned}$$

Also,

$$\eta_{\text{relative}} = \frac{\eta_{\text{thermal}}}{\eta_{\text{air-standard}}}$$

$$0.65 = \frac{\eta_{\text{thermal}}}{0.615}$$

$$\eta_{\text{thermal}} = 0.65 \times 0.615 = 0.4$$

But,

$$\eta_{\text{thermal (I)}} = \frac{\text{I.P.}}{\dot{m}_f \times C}$$

$$0.4 = \frac{\text{I.P.}}{\frac{14}{3600} \times 41800}$$

$$\text{I.P.} = \frac{0.4 \times 14 \times 41800}{3600} = 65 \text{ kW}$$

$$\eta_{\text{mech.}} = \frac{\text{B.P.}}{\text{I.P.}}$$

$$0.76 = \frac{\text{B.P.}}{65}$$

$$\text{B.P.} = 0.76 \times 65 = 49.4 \text{ kW}$$

where, p_{mi} = Indicated mean effective pressure

$$65 = \frac{2 \times p_{mi} \times 0.25 \times \pi / 4 \times 0.2^2 \times 300 \times 1 \times 10}{6}$$

$$\begin{aligned}
 \therefore p_{mi} &= \frac{65 \times 6 \times 4}{2 \times 0.25 \times \pi \times 0.2^2 \times 300 \times 10} \\
 &= 8.27 \text{ bar. (Ans.)}
 \end{aligned}$$

and brake mean effective pressure,

$$p_{mb} = 0.76 \times 8.27 = 6.28 \text{ bar. (Ans.)}$$

Real-Life Engine Optimization Examples

1. Turbocharging in Tractors

- **Example:** John Deere, Mahindra, New Holland tractors (40–75 hp).
- **Optimization:** Turbochargers force more air into the cylinder, improving **volumetric efficiency**.
- **Benefit:** 10–15% more power output without increasing engine size, better pulling ability for heavy implements like rotavators and cultivators.

2. Direct Injection (DI) Diesel Engines

- **Example:** Power tillers (like Kubota, VST Shakti) and tractors (Sonalika, Escorts).
- **Optimization:** Fuel is injected **directly into the combustion chamber** rather than the pre-chamber.
- **Benefit:** Improves **fuel efficiency by 8–12%**, reduces specific fuel consumption, and allows smoother operation at low speeds (ideal for puddling in rice fields).

3. Electronic Fuel Injection (EFI) in Modern Tractors

- **Example:** Massey Ferguson "NEXT GEN" tractors, John Deere 5E series.
- **Optimization:** Replaces mechanical pumps with electronically controlled injection.
- **Benefit:**
 - Optimized injection timing under varying load conditions.
 - Reduced **smoke and NOx emissions**.
 - Lower fuel consumption (important for long-duration field operations).

4. Engine-Governor Optimization

- **Example:** Power tillers (Greaves Cotton, Honda).
- **Optimization:** Centrifugal governors adjust fuel supply based on load.
- **Benefit:** Maintains **constant speed** during operations like ploughing or threshing, preventing engine stalling and saving fuel.

5. Multi-Speed PTO Optimization in Tractors

- **Example:** Swaraj, Mahindra, and New Holland tractors.
- **Optimization:** Engines are tuned to operate at **economy PTO rpm** (e.g., 540E at 1700 rpm instead of 540 at 2200 rpm).
- **Benefit:**
 - Saves **10–15% fuel** when operating implements like sprayers, threshers, and pumps.
 - Reduces engine wear by running at lower speeds.

6. Low-HP High-Torque Engines in Tillers

- **Example:** VST Shakti 130 DI tiller, Yanmar tillers.
- **Optimization:** Engines designed for **high torque at low rpm**.
- **Benefit:** Provides sufficient draft power for wetland puddling and dryland tillage without frequent gear changes, improving **work output per liter of fuel**.

7. Hybrid/Energy-Saving Designs

- **Example:** Escorts and TAFE developing hybrid tractor prototypes.
- **Optimization:** Engines coupled with **electric motors** for partial load conditions.
- **Benefit:** Lower diesel consumption during light-load operations (transport, spraying).

8. Cooling System Optimization

- **Example:** Kubota power tillers and Eicher tractors in hot climates.

- **Optimization:** Improved water cooling and oil cooling to maintain **engine temperature between 80–95°C**.
- **Benefit:** Prevents overheating during long ploughing sessions, improving **engine life**.

9. Alternate Fuel Engines

- **Example:** Kirloskar and Mahindra dual-fuel tractors (diesel + biogas/CNG).
- **Optimization:** Small diesel pilot injection with biogas/CNG as the main fuel.
- **Benefit:** Up to **25–30% reduction in diesel consumption**, suitable for farms with biogas plants.