Engine Testing

"The gem cannot be polished without friction, nor man perfected without trials."

Chinese proverb

$$CR = \frac{V_s + V_c}{V_c}$$

$$CR = \frac{6}{1}$$

Where:

$CR = compresion\ ratio$

$V_s = swept\ volume \quad [cm^3]$

$V_c = clearance\ volume\ [cm^3]$

The maximum figure is:

$CR = 12\ for\ SI\ engines$

$CR = 16{\sim}23\ for\ CI\ engines$

$$V_s = \frac{\pi}{4}(B)^2 * S$$

Where:

$B = cylinder\ bore\ [cm]$

$S = piston\ stroke\ [cm]$

$$S = 2r$$

$r = crank\ radius\ [cm]$

$$V = V_c + \frac{\pi * B^2}{4}(L + r - Y)$$

$$\theta = \cos^{-1}\left[\frac{Y^2 - L^2 + r^2}{2Yr}\right]$$

$$Y_i = r\cos\theta + \sqrt{L^2 - r^2\sin^2\theta}$$

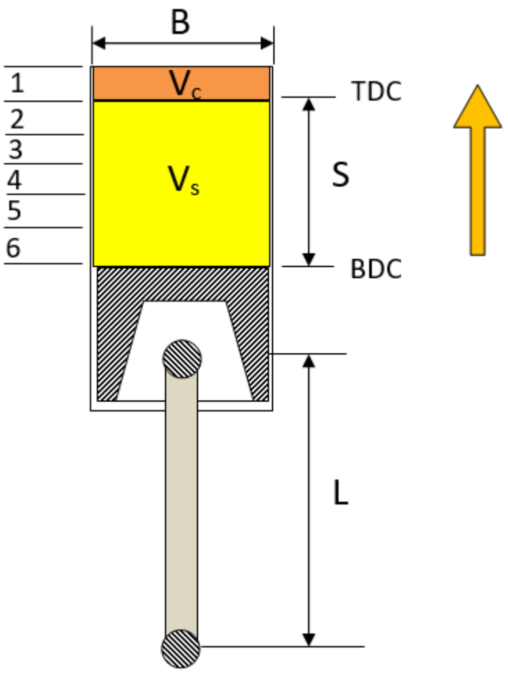


Figure 1 Engine compression ratio

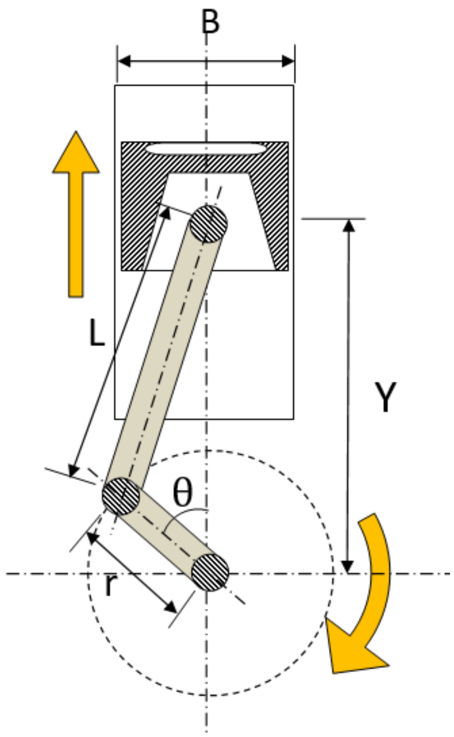


Figure 2 Engine dimensions.

where:

$L = connecting\ rod\ length\ [cm]$

$V = volume\ ahead\ the\ piston\ [cm^3]$

$Y = Distance\ between\ gudgeon\ pin\ and\ crankshaft\ centreline\ [cm]$

$\theta = crank\ angle\ [deg.]$

$$V_d = n * V_s$$

Where:

$V_d = engine\ displacement\ [cm^3]$

$n = Number\ of\ cylinders$

$$v = \frac{2SN}{60}$$

Where:

$v = mean\ piston\ speed\ [cm/s]$

$N = crankshaft\ angular\ velocity\ [rpm]$

Measurement of Torque and Power

The engine torque is measured by the dynamometer. All types of dynamometers operate on the same principle which is to measure a force acting through a distance.

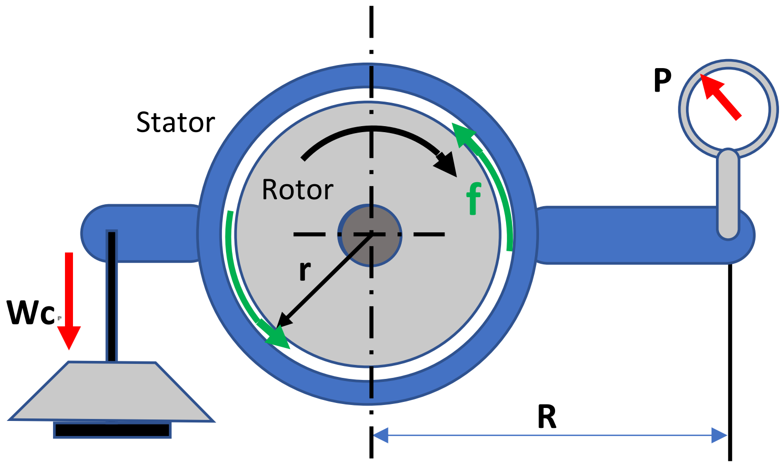


Figure 3 The principle of the brake dynamometer.

The distance (D) moved by a certain point on the rotor periphery in one revolution of the crankshaft is:

$$D = 2\pi r$$

Whatever the coupling matter between the rotor and the stator, (fluidic, magnetic flux, or electric field), a drag force applied by the stator will resist the rotation. The amount of work produced per revolution is:

$$W = f * D$$
$$W = 2\pi r * f$$

Where:

$D\ is\ the\ distance\ travelled\ in\ one\ revolution\ \ [m]$

$r\ \ is\ the\ radius\ of\ rotor\ \ [m]$

$f\ \ is\ the\ friction\ force\ in\ the\ fluid\ layers\ between\ the\ rotor\ and\ stator\ \ [N]$

The amount of torque exerted on the stator by the coupling is:

$$T = f * r$$

This torque could be measured by the scale reading (P) and the distance (R) between scale centerline and the center of rotation as follows:

$$T = P * R$$

$P\ \ is\ the\ pull\ \ \ [N]$

$$P = scale\ reading - tare\ weight$$

Therefore,

$$f * r = P * R$$

And the work is:

$$W = 2\pi * P * R \qquad [J]$$

The brake power of the engine is:

$$BP = \frac{W}{t} \qquad [kW]$$

$$BP = \frac{2\pi}{60} * N * T \qquad [kW]$$

$$1\ hp = 0.746\ kW$$

Eg:

A diesel engine was tested with a dynamometer having a tare weight of 214 N. The arm was 0.61 m in length. At a certain setting of the fuel pump, the engine ran at 1140 rpm and the gross weight on the scale showed 2176 N. Find:

a. The engine torque.
b. The power developed by the engine.

Solution:

$$P = gross\ weight - tare\ weight$$

$$P = 2176 - 214 = 1962\ kg$$

$$T = P * R$$

$$T = 1962 * 0.61 = 1{,}196.82\ \ N.m$$

$$BP = \frac{2\pi}{60} * N * T$$

$$BP = \frac{2\pi}{60} * 1{,}140 * 1{,}196.82 = 142.87\ kW$$

Dynamometers

A. Fluid Dynamometers

There are two types of fluid dynameters as follows:

- The friction fluid dynamometer: the coupling force between the rotor and the stator is produced by the friction between the fluid layers which strongly depend on fluid viscosity and is affected by fluid temperature.

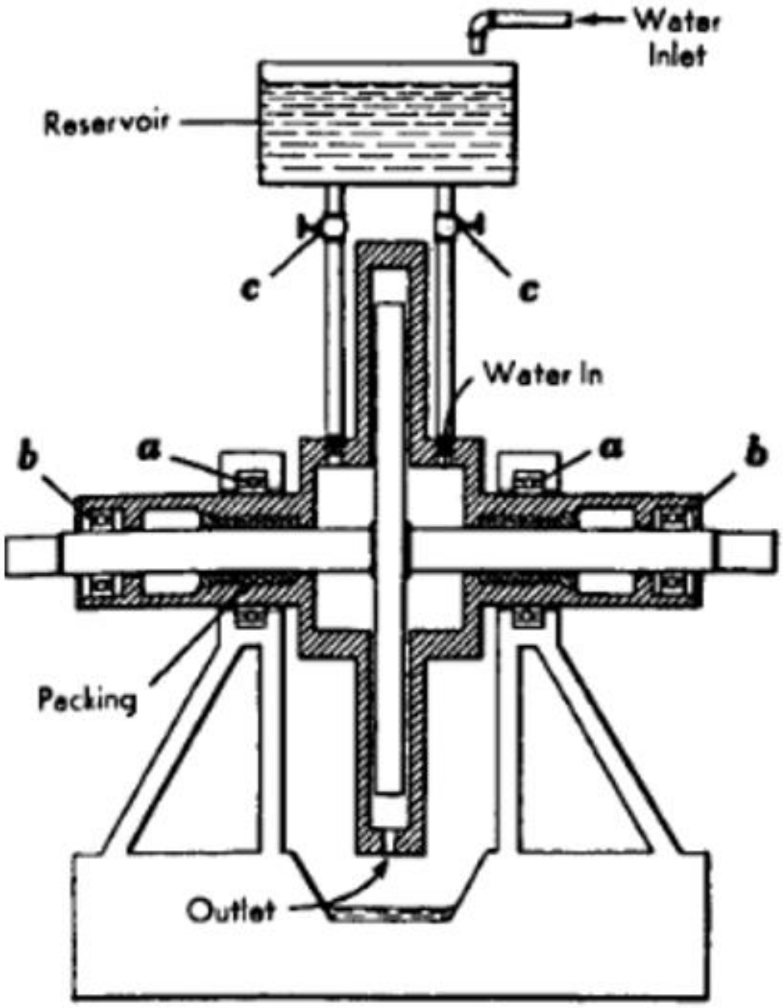


Figure 4 Sectional view of the friction fluid dynamometer.

The stator is cradled by the bearings (a) on the dynameter holding frame. The angular displacement of the stator actuates the strain gage which sends a voltage proportional to the angular displacement to the digital display.

- The agitator fluid dynamometer: the coupling force between the rotor and the stator is produced by the change in momentum of the fluid as it is directed from the rotor vanes to the stator vanes and circulates back to the rotor vanes.

The continuous change in fluid direction results in the change in fluid momentum which will intern produce a force on the stator. The force tries to rotate the stator which is cradled on the dyno frame. That angular motion of the stator is detected by the strain gage and converted to an electric signal as voltage to the digital display.

In both cases the absorbed power varies with the cube of rotor speed.

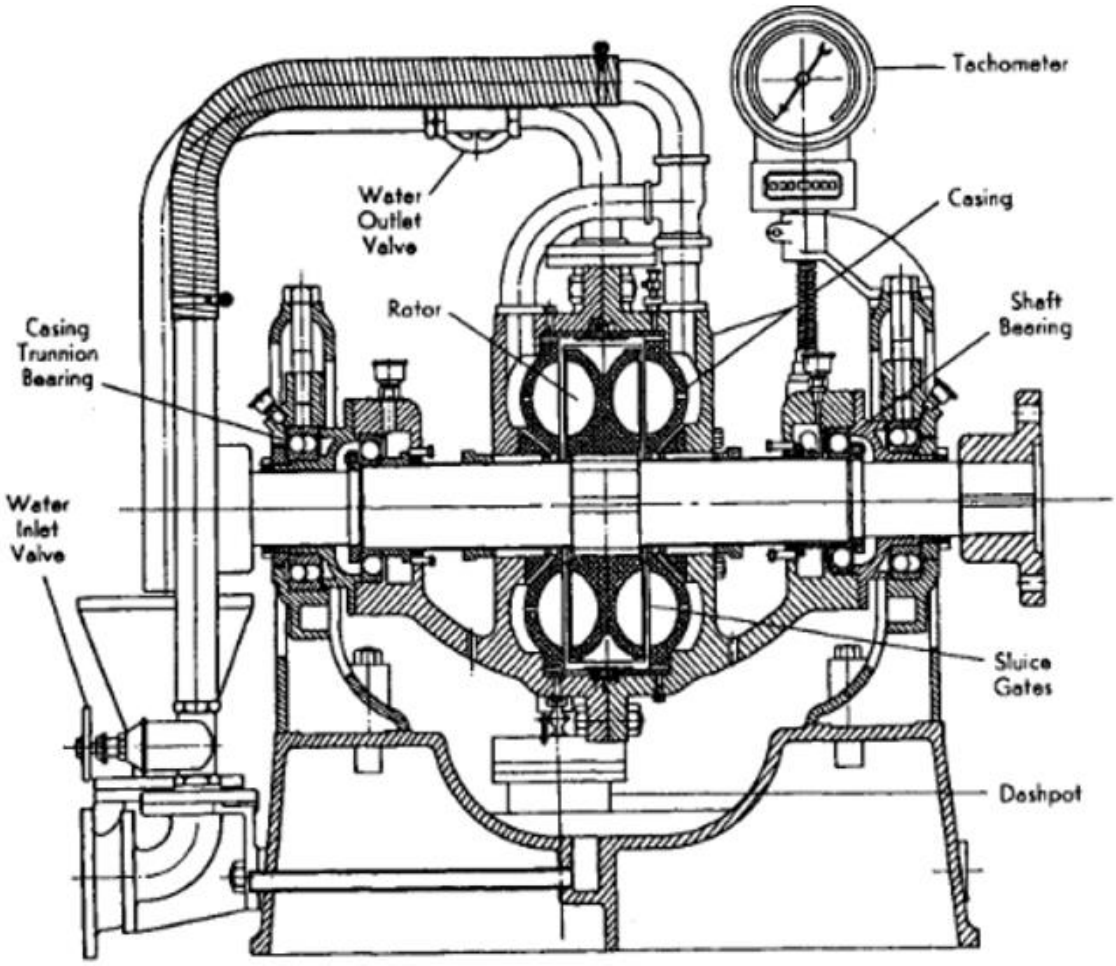


Figure 5 Sectional view in the agitator fluid dynamometer.

Dynamometers measure the torque developed by the engine while tachometers measure the speed of the engine. The final product is the brake power.

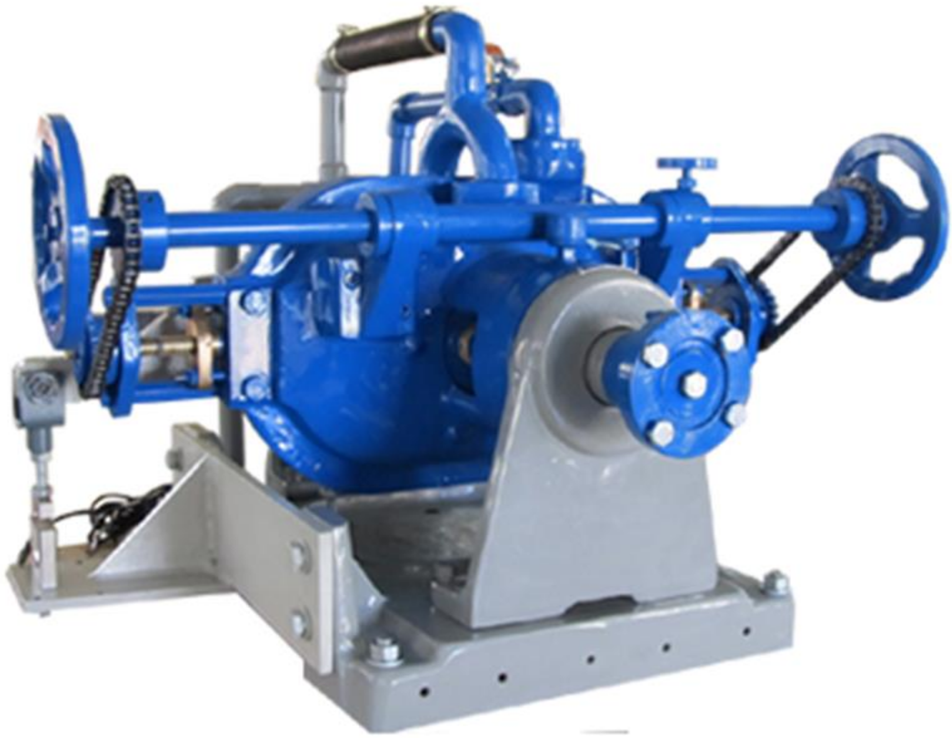


Figure 6 The agitator fluid dynamometer.

Engine Speed Measurement

The tachometer consists of a DC generator and a voltmeter. The higher the speed of rotation the higher the voltage indicated by the voltmeter. This voltage is converted to speed and calibrated by a precise speed pick-up sensor, a signal counter, and a stopwatch.

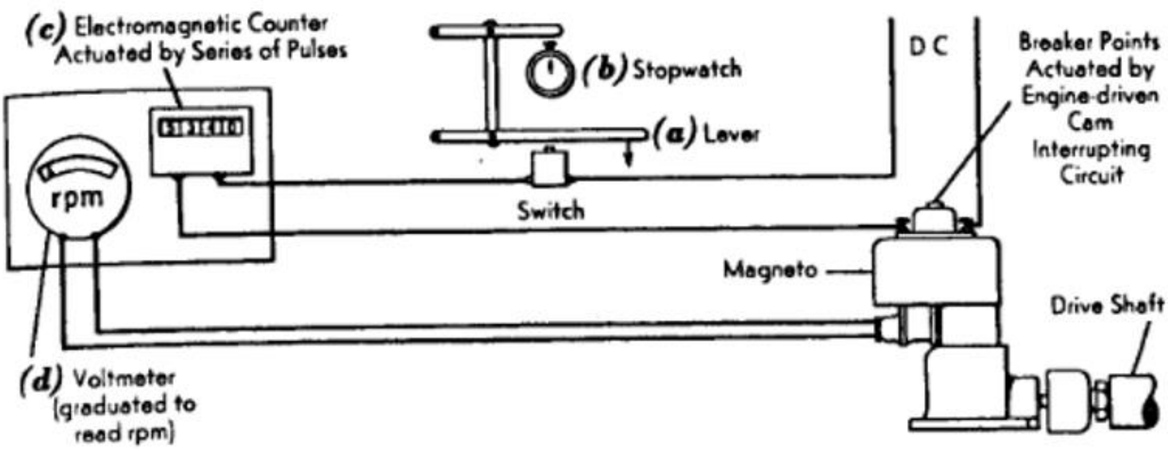


Figure 7 The speed measurement and calibration system.

Fuel Consumption Measurement:

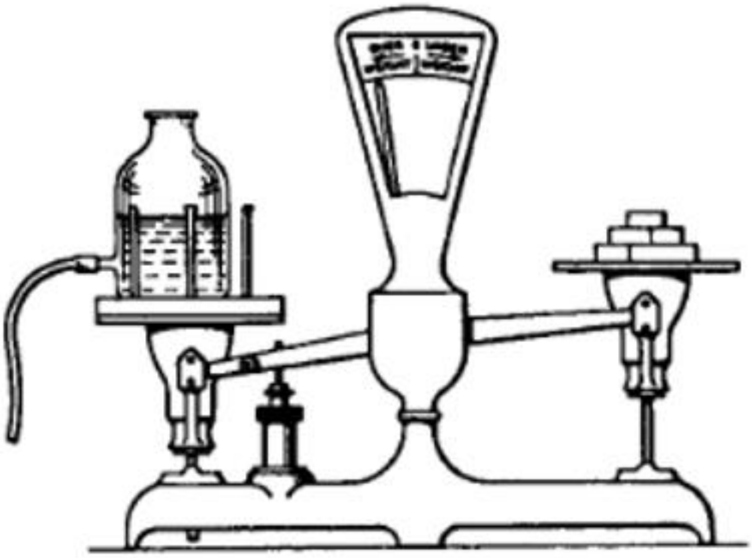


Figure 8 A balance for fuel consumption measurement.

When the engine consumes the fuel from the bottle, this side of the balance will become lighter therefore, when the dial reaches the zero reading, the time to consume a specified mass of fuel could be measured by a stopwatch. This quantity of fuel is called the mass flowrate of fuel.

$$\dot{m}_f = \frac{m_f}{t}$$

Where:

$\dot{m}_f = fuel\ mass\ flowrate\ [g/s]$

$m_f = mass\ of\ fuel\ consumed\ [g]$

$t = time\ elapsed\ for\ fuel\ consumption\ [s]$

The volumetric flowrate of fuel could be measured instantly by a rotameter which is merely a graduated cylinder open on both sides and held vertically. The fuel enters the pipe from the lower side to move upwards. In the pipe there is a conical float which is lifted by the fuel. The higher the fuel flowrate the greater the change in momentum and higher the lift force. The float is carefully selected to match the type of flowing fluid then the rotameter should be calibrated. It is very important to keep the pipe vertical and enable the float to rotate by helical grooves carved on the sides of the float. This will guarantee a central position of the float to avoid friction with the pipe walls.

$$\dot{m}_f = \rho_f * Q_f$$

Where:

$\rho_f = fuel\ density\ [g/cm^3]$

$Q_f = fuel\ volumetric\ flowrate\ [cm^3/s]$

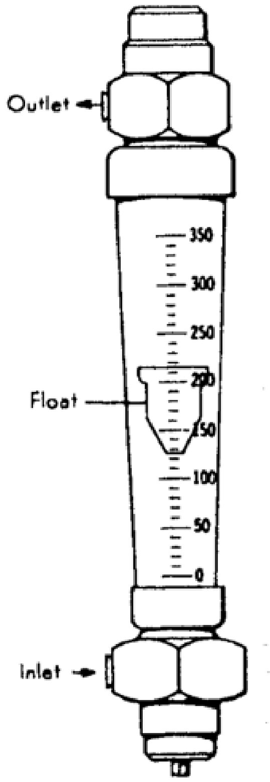
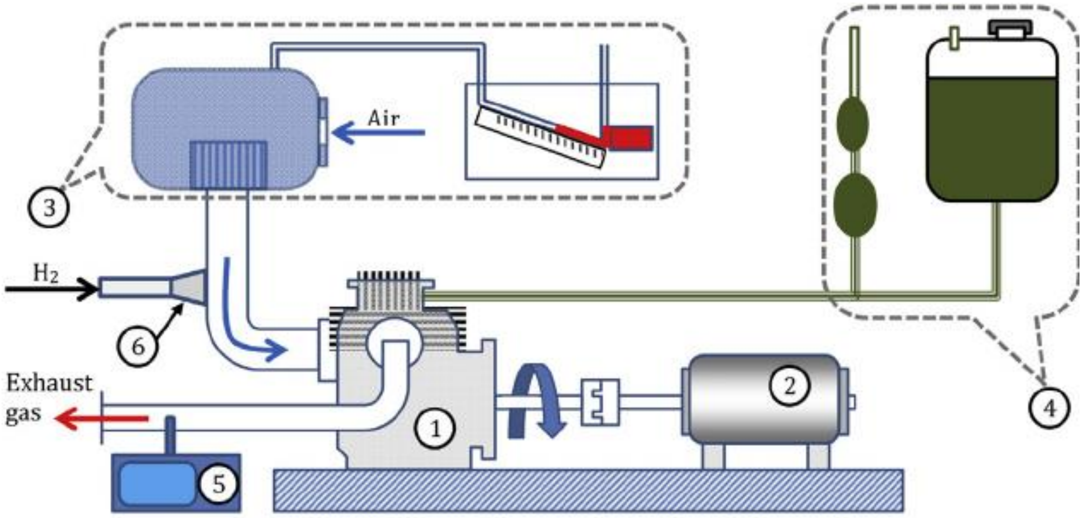


Figure 9 The rotameter for instant flow measurement.

Air Consumption Measurement

The amount of fuel consumed by the engine should be proportional to the amount of fuel consumed at each operational condition and engine demand. The flow of air in the intake manifold is intermittent especially in engines of less than four cylinders. This makes air flow measurement very complicated. Therefore, this intermittent flow could be dampened by passing the air in a large diameter surge tank. An orifice or flow nozzle is fitted to the far end of the container. The volumetric air flow is obtained by measuring the pressure drop across the orifice by an inclined manometer.

For large engines a few orifices are installed in parallel. The orifice should be calibrated to indicate the coefficient of discharge.



1: Ricardo E6 Engine
2: Electrical Dynamometer
3: Air intake metering system
4: Diesel Fuel metering System
5: Gas Analyzer & Opacity meter
6: Hydrogen supply

Figure 10 Engine air and fuel consumption measurement systems.

$$\dot{m}_{a_{act}} = \rho_a C_v C_c A_o \sqrt{2gh}$$

Where:

$\dot{m}_{a_{act}} = actual\ air\ mass\ flowrate\ [kg/s]$

$\rho_a = air\ density\ [kg/m^3]$

$C_v = air\ velocity\ correction\ coefficient$

$C_c = orifice\ area\ correction\ coefficient$

$A_o = orifice\ area\ [m^2]$

$g = gravitational\ acceleration\ [m/s^2]$

$h = manometer\ reading\ [m]$

Engine Performance ratings

These are a group of engine parameters which are used to determine the real performance of the engine, also to compare the performance of different engines regardless to their size and speed.

A. The Brake Mean Effective Pressure

The indicated mean effective pressure (i.m.e.p.) is the algebraic sum of the mean pressures acting on the piston during each stroke over one complete cycle as shown in fig.11. It is calculated using the indicator diagram drawn by the engine indicator. Thus,

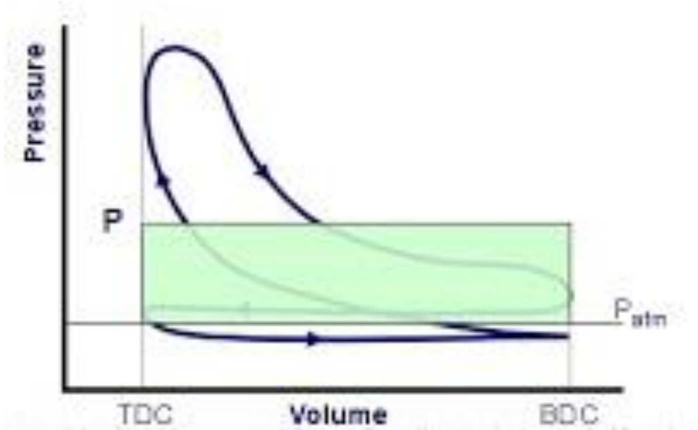


Figure 11 P-V diagram and the mep.

$$imep = \frac{area\ of\ indicator\ diagram}{length\ of\ indicator\ arm * spring\ scale\ index}$$

The brake-mean effective pressure (b.m.e.p.) is the m.e.p. which could have developed power equivalent to the BP. It is a comparative measure of the power capabilities of engines per unit displacement, which operate with the same speed, and forms a basis for the index of performance.

$$bmep = \frac{60 * x * BP}{ASnN}$$

The b.m.e.p. unlike i.m.e.p. cannot be measured directly. Both b.m.e.p. and i.m.e.p of a petrol engine increase with the compression ratio up to the limit of compression fixed by the detonating properties of the fuel used.

The b.m.e.p. indicates how well the engine is using its displacement to produce work (torque).

$$BP = \frac{2\pi NT}{60} = \frac{(bmep)ASnN}{60 * x} \qquad [W]$$

Therefore,

$$bmep = 2\pi x \frac{T}{V_d}$$

This indicates that bmep is a specific torque.

The torque and the b.m.e.p. of a given engine are linearly related. Since,

$$T = \frac{V_d}{2\pi x} * bmep = C * bmep$$

where,

$$C = \frac{V_d}{2\pi x}$$

$C = constant\ for\ a\ given\ engine.$

$x = number\ of\ strokes\ per\ cycle.$

$x = 1\ for\ a\ 2 - stroke\ cycle\ engine.$

$x = 2\ for\ a\ 4 - stroke\ cycle\ engine.$

$N/x = number\ of\ effective\ strokes.$

Thus, when brake torque and b.m.e.p. are plotted against rpm., the shapes of the resulting curves should be similar.

B. Specific Fuel Consumption

The total consumption of fuel by an engine under test conditions in a given duration of time is determined by measuring its volume or weight. The specific fuel consumption (s.f.c) is defined as the total fuel consumption per hour per kW developed [kg/kW.h]. In other words, s.f.c. is the rate of fuel consumption per kW. When IP is used to calculate s.f.c, it is known as indicated specific fuel consumption (i.s.f.c) and when BP is used, it is termed brake specific fuel consumption (b.s.f.c). Thus,

$$isfc = \frac{\dot{m}_f}{IP}$$

$$bsfc = \frac{\dot{m}_f}{BP}$$

The b.s.f.c. shows how well the engine is converting fuel into work. It is more reliable than thermal efficiency because it doesn’t comprise the heating value of the fuel. In practice the exact value of HV is unpredictable in a running engine, it could take any value between the HHV and LHV.

C. Volumetric Efficiency, η_v

It is defined as the ratio of the actual weight of air induced by the engine in the intake stroke to the theoretical weight of air that should have been induced due to piston displacement at a defined intake temperature and pressure. Thus, neglecting the presence of fuel in the mixture,

$$\eta_v = \frac{Volume\ of\ aspirated\ air\ at\ intake\ conditions\ per\ minute}{Engine\ displacement * number\ of\ intake\ strokes\ per\ minute}$$

$$\eta_v = 60 * \frac{\dot{m}_{a_{actual}}}{\rho_a * V_d * \frac{N}{x}}$$

As the volume of petrol present in the mixture is negligible in petrol engines, its presence may be overlooked without getting appreciable errors in η_v. In case of gas engines, the presence of gas displaces a significant part of the volume of air supplied to the engine; hence its influence on η_v cannot be overlooked.

The actual weight of air aspirated under maximum output conditions is always less than the theoretically possible weight due to the following reasons:

- Long and tortuous inlet passage and the presence of the throttle valve in SI engines.
- Insufficient inlet and exhaust valve area.
- Excessive friction of the mixture due to passage through rough surfaces and sudden changes in section of the inlet pipe and the throttle valve.
- Premature heating of the mixture by induction manifold, valves and ports, combustion chamber and cylinder walls, before inlet valve closes.
- Heating of the fresh charge by the residual exhaust gases in clearance volume.
- Excessive back pressures due to exhaust gases.
- Cooling water temperature in cylinder block passages.
- Poor valve design causes insufficient valve lift.
- Incorrect valve timing, i.e., the opening and closing points are incorrect.

Volumetric efficiency is a measure of the breathing ability of an engine. In supercharged engines, the volumetric efficiency is dependent upon the following factors:

- **Engine speed**
  The volumetric efficiency η_v, after attaining a maximum value at a certain speed, falls with any further increase in speed. This generally happens in the case of petrol engines.
- **Compression Ratio**
  η_v tends to decrease with increase in the compression ratio.
- **Mixture Strength.**
  η_v is a minimum for correct and slightly lean mixtures but is a maximum for rich mixtures.
- **Temperature of Inlet Air**
  η_v decreases as the air temperature increases.
- **Temperature of Cooling Water**
  η_v increases slightly with a reduction in the temperature of cooling water.

D. The power-over-weight ratio

It is a ratio between the amount of power (BP) developed by the engine divided by the weight of the engine.

$$\frac{P}{W} = \frac{BP}{W}$$

In automotive practice, the higher the P/GVW the better is the vehicle, however, if the weight of the vehicle is reduced beyond certain limits, the safety of the vehicle might be compromised.

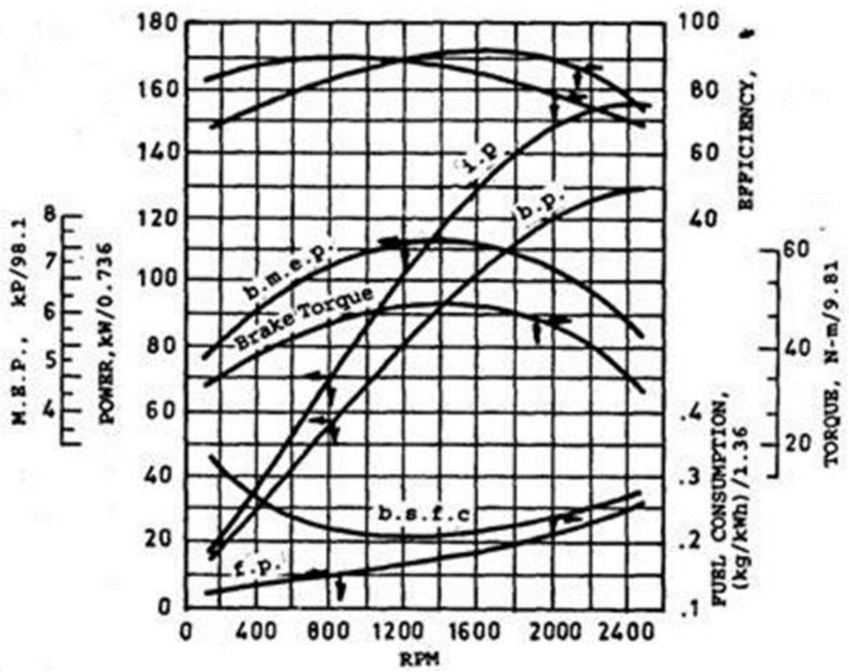


Figure 12 Typical performance graphs of an automotive CI engine.

Example

An eight-cylinder automobile engine of 85.7 mm bore and 82.5 mm stroke with a compression ratio of 7 is tested at 4000 rpm. on a dynamometer which has a 0.5335 m arm. During a 10 minutes test at a dynamometer scale beam reading of 400 N, 4.55 kg of gasoline for which the heating value is 46 MJ/kg are burnt, and air at 294°K and 10 x 10^4 Pa is supplied to the carburetor at the rate of 5.44 kg per min. Find:

a. the BP delivered,
b. the b.m.e.p,
c. the b.s.f.c,
d. the specific air consumption,
e. the brake thermal efficiency,
f. the volumetric efficiency,
g. the air-fuel ratio.

Solution:

a.

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

$$BP = \frac{2\pi * 4000 * 400 * 0.5335}{60 * 1000} = 89.34\ kW$$

b.

$$bmep = \frac{60 * x * BP}{ASnN}$$

$$bmep = \frac{60 * 2 * 89.34 * 1000}{0.0825 * \frac{\pi(0.0857)^2}{4} * 8 * 4000} = 704.36\ kPa$$

c.

$$\dot{m}_f = \frac{4.55}{10} = 0.455\ kg/min$$

$$bsfc = \frac{\dot{m}_f}{BP}$$

$$bsfc = 60 * \frac{0.455}{89.34} = 0.306\ \ kg/kW.h$$

d.

$$bsac = 60 * \frac{\dot{m}_a}{BP}$$

$$bsac = 60 * \frac{5.44}{89.34} = 3.65 \ kg/kW.h$$

e.

$$\eta_{thb} = \frac{BP}{\dot{m}_f * CV}$$

$$\eta_{thb} = \frac{89.34 * 60}{0.455 * 46000} = 25.6 \ \%$$

f. The theoretical air volume at this engine speed is:

$$V_{th} = \frac{ASnN}{x}$$

$$V_{th} = \frac{0.0825 * \pi(0.0857)^2 * 8 * 4000}{4 * 2} = 7.62 \ m^3/min$$

$$P * V_a = \dot{m}_a * R * T$$

$$V_{act} = \frac{5.44 * 287.1 * 294}{10 * 10^4} = 4.56 \ m^3/min$$

$$\eta_v = \frac{V_{act}}{V_{th}}$$

$$\eta_v = \frac{4.56}{7.62} * 100\% = 60\%$$

g.

$$A/F = \frac{\dot{m}_{a_{act}}}{\dot{m}_f}$$

$$A/F = \frac{5.44}{0.455} = 11.96$$

Energy Losses (Heat Balance)

Only part of the energy supplied to the engine is transformed into useful work whereas the rest is either wasted or utilized for heating purposes. The main part of the unutilized heat goes to exhaust gases and to the cooling system. To draw a heat balance chart for an engine, tests should be conducted to give the following information.

- Energy supplied to an engine which is known from the heating value of the fuel consumed.
- Heat converted to useful work.
- Heat carried away by cooling water.
- Heat carried away by exhaust gases.
- Heat unaccounted for (radiation etc.)

It is expected that the heat balance results of CI engine must differ from that of SI engine due to much higher compression and expansion ratios in the former. The higher compression ratio results in lower exhaust gas temperature and lower flame temperature that in turn causes lower heat loss to the cylinder walls in CI engines. The utilization of the fuel's heat energy is also higher in CI engines because of its higher compression ratio.

Although the actual value of heat utilization is dependent upon many factors like compression ratio, engine load, fuel injection quantity, timing etc. Some average figures for heat are presented in the following table:

Table 1 Typical values for fuel energy fractioning in SI and CI engines.

Item	S.I. Engine	C.I. Engine
Heat converted to useful work (IP)	25 to 32%	36 to 45%
Heat carried away by cooling water	33 to 30%	30 to 28%
Heat carried away by exhaust gases	35 to 28%	29 to 20%
Heat unaccounted for	7 to 10%	5 to 7%
Total = Energy supplied	100%	100%

If the shaft work (BP) is considered instead of useful work, the mechanical losses are to be accounted for or are generally included in the cooling water heat.

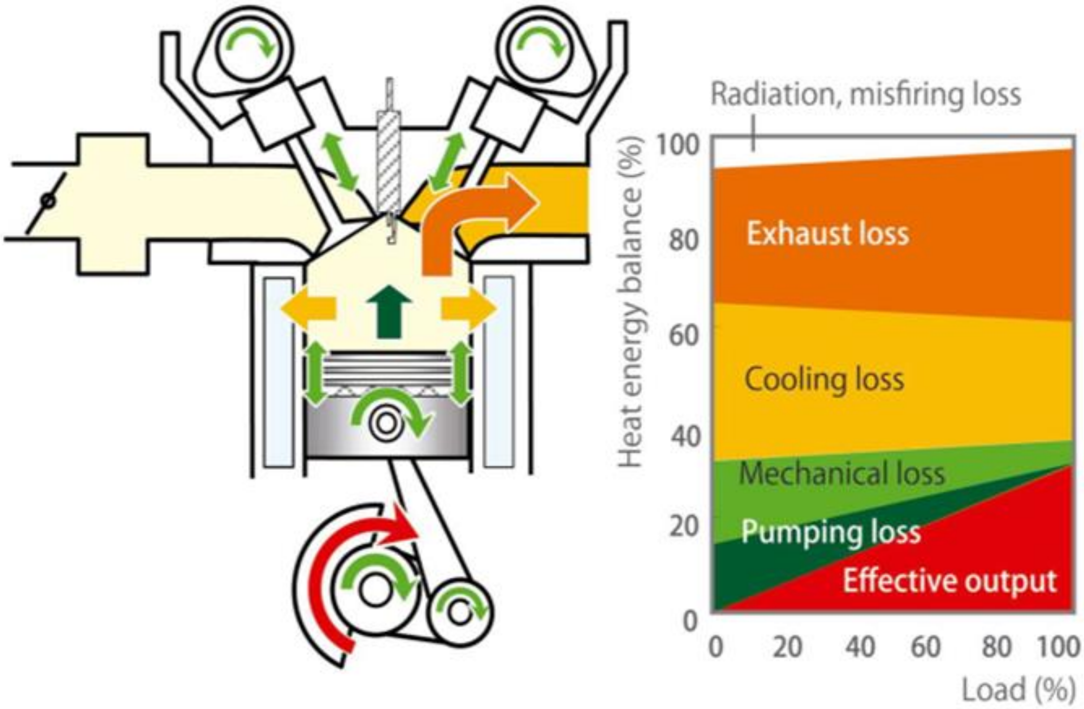


Figure 13 Energy Balance

Factors Affecting Engine Performance

A. Heat Transfer

The heat is exchanged in both directions between the gases and engine cylinder walls and the other parts of the engine being in contact with the gases. During combustion, expansion, exhaust and the later part of the compression, heat transfer takes place from the gases to the walls and from the wall to the cooling water or ambient air. During suction and the earlier part of the compression, heat transfer takes place from the walls to the gases. The heat lost to the walls during the later part of compression is almost equal to or less than the heat received by the gases from the walls during early part of compression. The amount of heat lost during exhaust stroke is unavoidable. The heat lost during combustion and expansion lowers the thermal efficiency of the engine. The factors that affect the heat losses to the walls are as follows:

i Charge combustion duration

A prolonged combustion duration increases heat loss.

ii Combustion temperature

This depends upon the fuel type, compression ratio and the load on the engine. The temperature increases with load and compression ratio. It increases the thermal loss.

iii Engine speed.

The increase of the engine speed decreases the duration of combustion hence decreases the heat loss.

iv Combustion chamber configuration.

Increasing the surface to volume ratio of the combustion chamber decreases the heat loss. However, turbulence and flame propagation also affect the heat transfer to the combustion chamber walls.

v The number of cylinders for a given engine capacity.

The effect of cylinder size is rather complicated. An increase in the cylinder size decreases the surface to volume ratio but increases the flame travel. This increases the combustion duration and hence engine speed is decreased.

vi Ignition timing in S.I. engines and fuel injection timing in C.I. engines.

Proper ignition and injection timings accelerate the combustion process with less after burning and hence less heat loss. The heat flow from the walls to the fresh charge during suction stroke increases the temperature of the charge and hence decreases the density of charge. This decreases the amount of power developed by the engine.

B. Residual Gas

The residual gases left in the compression chamber from the previous cycle dilute the fresh charge by increasing the amount of inert gases in it. This affects the ignition and combustion. The residual gases also lower the volumetric efficiency of the engine and raise the temperature of the charge. They both act to lower the amount of fresh charge induction.

C. Valve Resistance

In the theoretical cycle of four-stroke engines, it is assumed that the exhaust and intake pressures are equal to the atmospheric pressure. But the exhaust pressure is higher, and the suction pressure is lower than atmospheric pressure due to the resistances in exhaust and intake manifolds and valves. The valve resistance affects the volumetric efficiency. The valve resistance causes the pumping losses, which is the negative loop on the P-V diagram. The pumping losses increase with an increase in speed. In two-stroke engines, the power consumption of scavenge and charging pumps corresponds to the pumping losses in four-stroke engines.

D. Valve Timing

In ideal cycles, it is assumed that opening and closing of the intake and exhaust valves take place on dead centers. However, in practice, the exhaust valve closes, and intake valve opens approximately on TDC, but the opening of the exhaust valve and the closing of the intake valve vary considerably from the BDC, depending principally on the desired speed. The net result due to deviations of valve opening and closing other than at dead centers is that the indicator diagram is rounded at the exhaust corner. This reduces the work output by 1 to 2%.

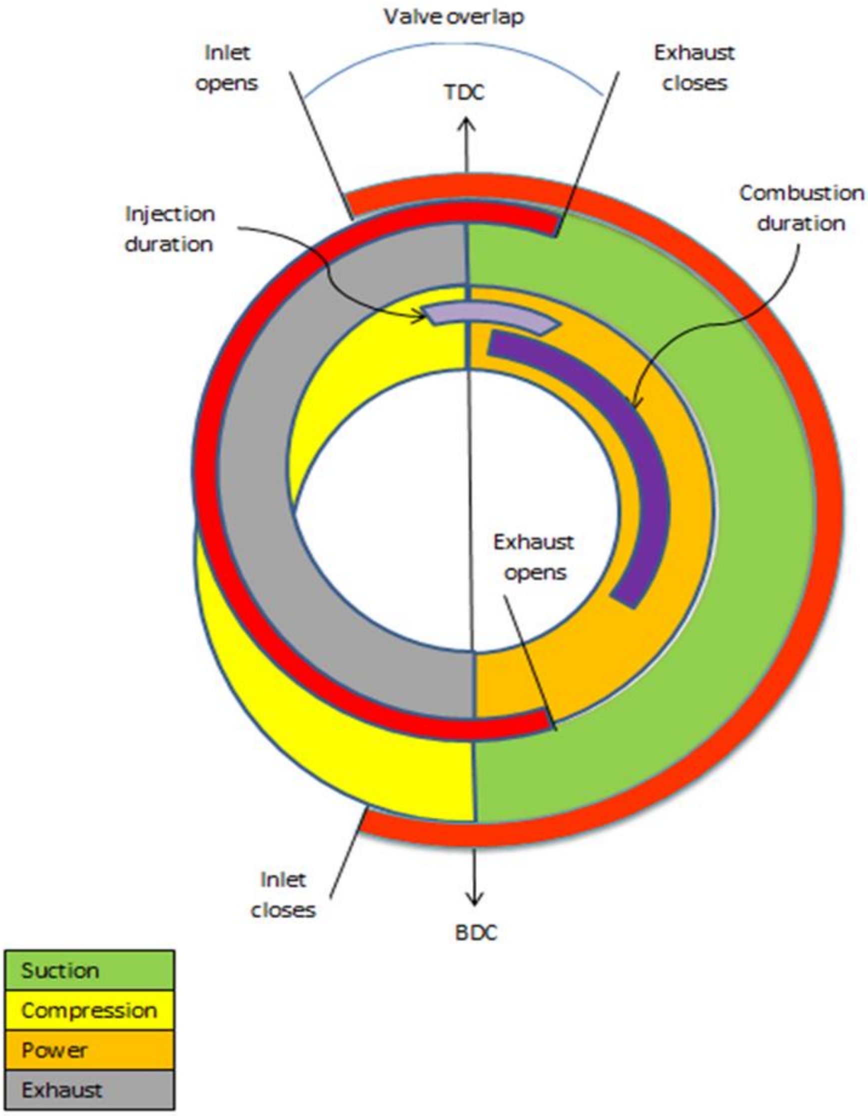


Figure 14 Valve timing, injection timing and combustion during one cycle of a four-stroke engine.

E. Combustion Duration

In ideal cycles, it is assumed that the time of combustion is zero for the constant-volume process and combustion occurs at a rate necessary to maintain constant pressure during the constant pressure process. Actually, the combustion process requires an appreciable amount of time, which depends upon various factors. The increase in combustion time decreases the ideal efficiency by 2 to 3%.

F. Incomplete Combustion

A volumetric analysis of the products of combustion indicates an incomplete combustion of about 2% of the heating value of the fuel. A mixture with excess air tends to reduce this loss to zero; on the other hand, rich mixtures result in considerable unburnt fuel due to oxygen deficiency.

G. Atmospheric Conditions

Air temperature, humidity and barometric pressure affect the air charge. The weight of the air charge was found to be inversely proportional to the square root of the temperature, especially in high-speed automobile engines. Therefore, to obtain the performance at the standard conditions, the following corrections on pressure, temperature and humidity are to be adopted.

i Pressure.

The standard pressure is taken as 760 mm Hg. Adopting correction on observed BP:

$$BP_c = BP * \frac{760}{p}$$

where,

$p \ is \ the \ pressure \ in \ the \ test \ laboratory, [mm \ Hg]$

$BP_c = the \ corrected \ brake \ power \quad [kW]$

ii Temperature.

The standard temperature is taken as 25°C.

$$BP_c = BP * \sqrt{\frac{273 + t}{298}}$$

Where:

$t = the \ temperature \ of \ the \ test \ laboratory, [°C].$

iii Humidity.

The correction for water vapor pressure present in atmosphere is to be made for getting accurate results. The vapor pressure can be obtained by knowing the wet-bulb and dry-bulb temperatures and using the psychometric chart. If p_v is the vapor pressure in the test laboratory in mm of Hg, then the corrected barometric pressure of the test laboratory is $p - p_v$.

Thus, the formula with the above corrections is:

$$BP_c = BP * \sqrt{\frac{273 + t}{298} * \frac{760}{p - p_v}}$$

The effect of change of pressure is to increase or decrease the output power as the level in the barometer rises or falls respectively. The BP varies inversely as the absolute temperature of the intake air increases.

Types of SI Engine Tests

There are two major types of tests for IC engines.

A. The Variable Speed Test

This test is conducted to mobile engines. The test is designed to imitate the real conditions of variable speed and load conditions. The test is subdivided to:

i Full-Load Test:

The engine is started under no-load and left to warm-up. Then the throttle valve is gradually opened however, to maintain the engine speed low, the brake load is also increased simultaneously. As the throttle valve become wide open (WOT) and the load is adjusted so that the engine is not labouring, the engine temperature is watched to stabilise. The time required to collect a specified amount of fuel is recorded. If the spark timing is not adjusted automatically, then it should be optimised to the speed of maximum power. Engine speed, brake load, and the temperatures should be recorded. This is considered as the first run of the test. The second run is accomplished by reducing the brake load while keeping the throttle position unchanged. The spark timing should be optimised again, and the same procedure is repeated. This procedure continues until the entire load is removed.

ii Part-Load Test

The same previous procedure is followed except that the throttle is partially opened. Therefore, at half-load test the throttle is half open and so on.

The SI engine characteristic graphs are presented in fig.14. the following features are perceived from the graphs.

I. Engine torque is strongly dependant on volumetric efficiency and friction losses.
II. Doubling the engine displacement doubles the torque.

III. Brake mean effective pressure (bmep) curve is identical to torque curve. It is independent of engine displacement, but it strongly depends on volumetric efficiency and friction losses.
IV. The speed of maximum torque and mep is nearly half that of the maximum power.
V. The maximum power of the engine depends on engine speed. Doubling the speed of the engine could double the power output. This is achieved by increasing the volumetric efficiency and reducing the friction losses.
VI. The brake specific fuel consumption (bsfc) curve is an inverted mirror of the (bmep) curve. Its minimum value corresponds to speed of maximum torque.
VII. The friction power increases rapidly at high speeds because of the high inertia and fluidic losses at high speeds.

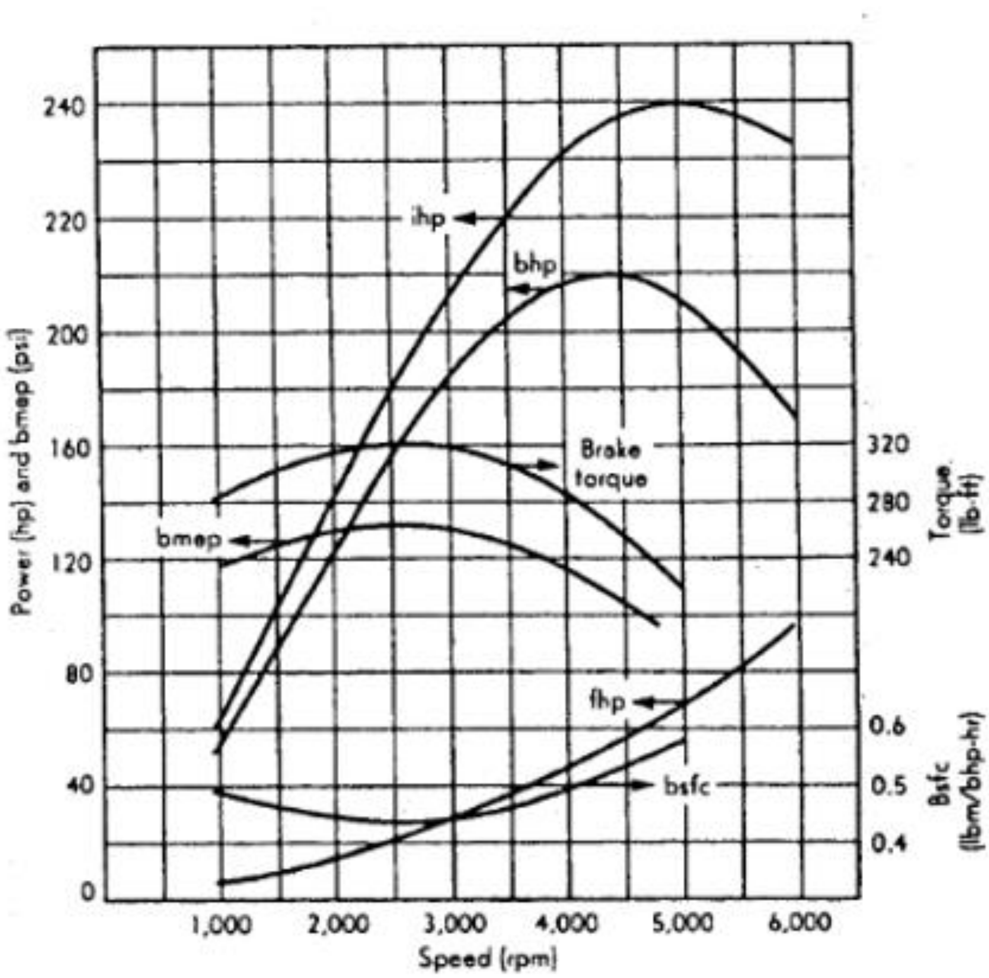


Figure 15 Variable- speed test of automotive SI engine at WOT (CR=9)

B. Constant-Speed Test

I. The engine should warm up while idling until its temperature stabilizes.
II. A certain engine speed is selected at no load condition.
III. The throttle valve is adjusted to attain this particular speed.
IV. A set of engine data is taken which comprises the following parameters:
- Throttle position,
- Load,
- Speed,
- Time to consume a specified amount of fuel.
- The temperatures.

V. As the load increased, the previous engine speed should be maintained by increasing the throttle opening.
VI. This procedure continues until the WOT is reached.
VII. Steps II through VI should be repeated a few times but at different speeds.
VIII. The load should be removed, and the throttle opening reduced simultaneously.
IX. The engine could be switched off safely as the temperature drops to normal operational temperatures.

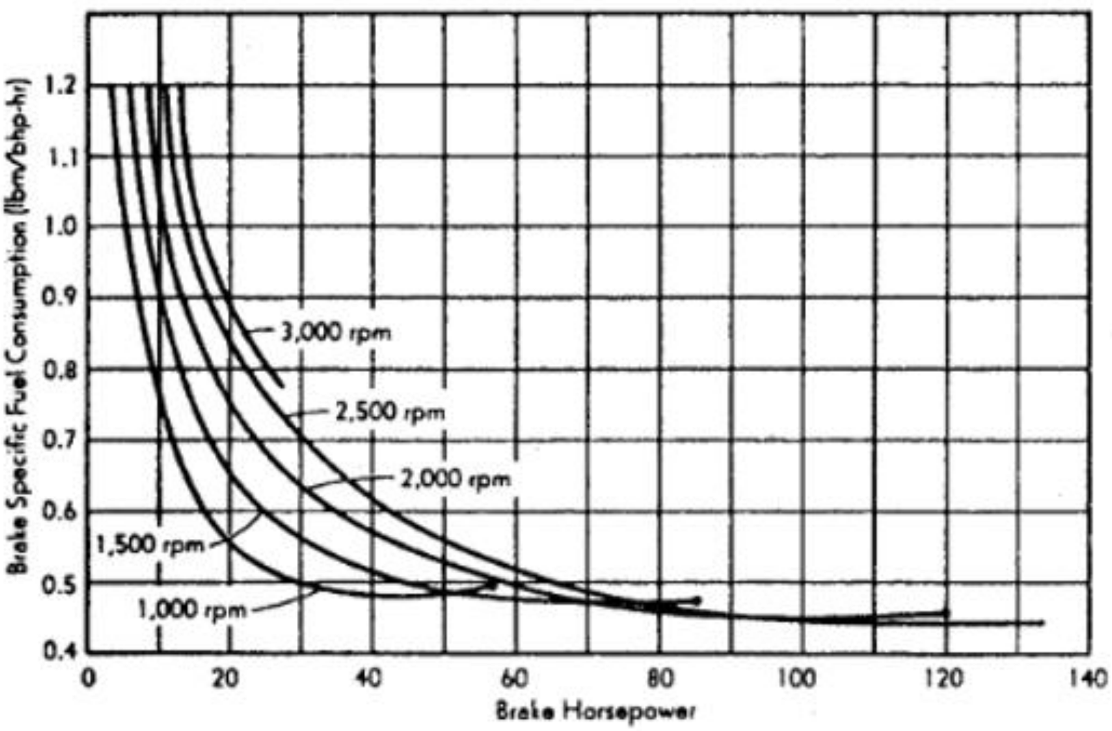


Figure 16 constant speed test of SI engine (CR=9).