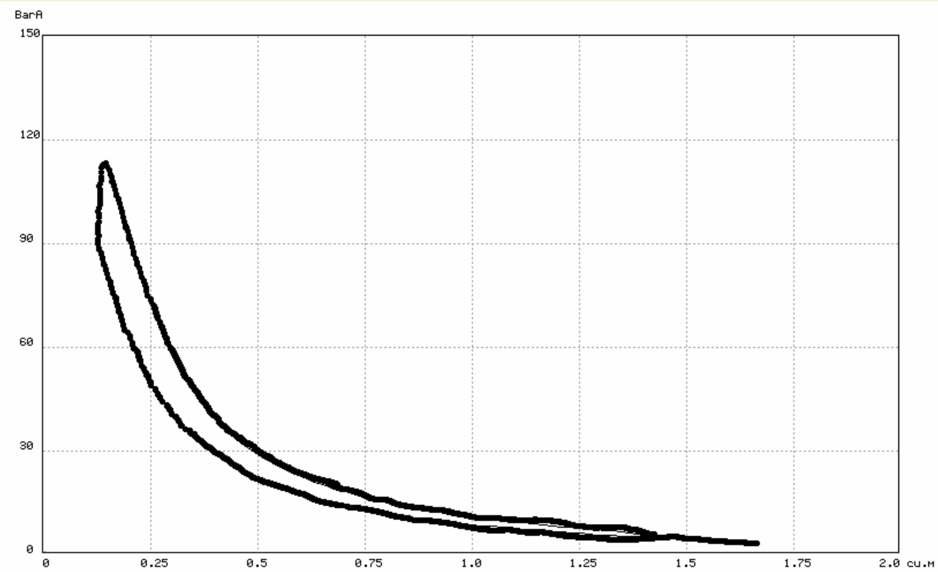
Internal Combustion Engine Performance

Criteria of performance

In section three, we have examined theoretical heat engine cycles and identified their relationship with "real" cycles. However from an operational point of view we are more concerned with an engines power output and fuel consumption, similarly if we are to select an engine for a particular application, the main consideration would be its' power / speed characteristics with important additional factors such as initial capital cost and running cost also being considered.
We may also wish to compare different types of engines or different engines of the same type that use the same thermodynamic cycle. In this case, we must define other performance criteria which can be obtained by measurement during test bed trials and calculation by standard procedures.
The results of these trials may be plotted graphically in the form of performance curves.

Indicated power (ip)

This is defined as the rate of work done by the gas on the piston as determined from an indicator diagram obtained from the engine.
Such a diagram for a two-stroke engine is shown below.



Indicated mean effective pressure P_i

This is defined as the height of a rectangle having the same length and area as the cycle plotted on a PV diagram, multiplied by the indicator spring constant.
In other words, the area of the power card divided by the length of the card and multiplied by the spring constant.

Considering one engine cylinder
The indicated mean effective pressure, "Pi" is given by

$$\text{Indicated mean effectve pressure} = \frac{\text{Diagram area}}{\text{Diagram length}} \times \text{Spring constant}$$

$$Work\ done\ per\ cycle = Force \times distance\ moved = Pressure \times Area \times Engine\ Stroke\ length$$

The actual pressure changes throughout the cycle but the same effect is given by the mean effective pressure acting for the complete stroke.

$$Work\ done\ per\ cycle = Mean\ effective\ Pressure \times Area\ of\ Cylinder \times Stroke$$

$$Power = Work\ per\ unit\ Time$$

$$Power = Pressure \times Area \times Engine\ Stroke\ length \times working\ cycles\ per\ unit\ Time$$

$$Indicated\ Power = P_i \times length \times area \times n$$

The number of cycles per unit time "n" depends on the type of engine;

Four-stroke engines the number of cycles per unit time is half the engine speed (Rev/sec)

Two strokes the number of cycles per unit time is the engine speed (Rev/sec)

Total indicated Power for the Engine
Usually there would be one power card for each cylinder so the total power would be the sum of the individual cylinder powers, however in most questions, only one area is given and the assumption is made that all cylinders produce the same power.

$$Indicated\ Power\ of\ Engine = Indicated\ Power\ of\ one\ cylinder \times Number\ of\ Cylinders$$

Brake power (bp)

This is the measured output of the engine.
The engine is connected to a brake or dynamometer which provides a load so that the engine torque can be measured.
In dynamometers that absorb the engine power the torque is obtained by reading off a net load "F" at a known radius "R", from the axis of rotation, hence the torque "T", is given by force times radius (T=FxR)
In the transmission type of dynamometer the torque "T", transmitted by the driving shaft is measured directly by means of a torsion meter.

The brake Power is then given by

$$Brake\ Power\ of\ Engine = Torque \times angular\ velocity = T\omega$$

$$Brake\ Power\ of\ Engine = 2\pi \times n \times T$$

Friction Power (f p)

The difference between the indicated power and the brake power is that power required to overcome the frictional resistance within the engine and is called the friction power "fp".

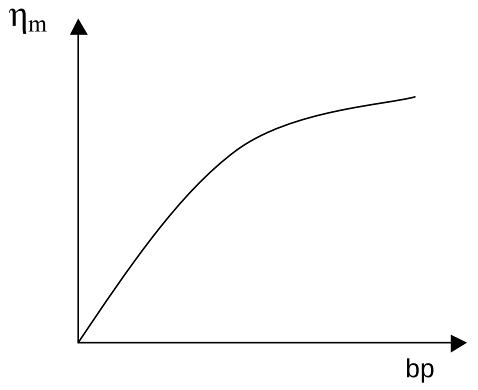
$$Friction\ Power\ of\ Engine = Indicated\ \ power - Brake\ \ power$$

Mechanical Efficiency, η_M

The mechanical efficiency of the engine. is defined as

$Mechanical\ \ Efficiency\ \eta_m = \dfrac{Brake\ \ power}{Indicated\ \ power}$ and usually lies between 80 and 90%.

The fp is very nearly constant at a given engine speed; if the load is decreased giving lower values of bp, then the variation in η_M with bp is as shown below.



At zero bp at the same speed the engine is developing just sufficient power to overcome the frictional resistance.

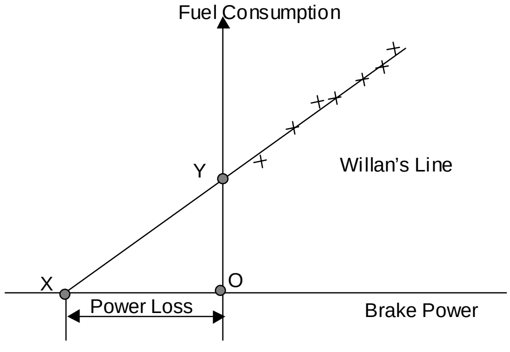
Mechanical efficiency depends on the indicated power and brake power and is found by determining these values on a test bed, several methods can be used each with its advantages and disadvantages, some common methods are;

1. Measurement of the indicated power using power cards and the brake power by measuring the torque as already described.
2. The Morse test: only applicable to multi-cylinder engines.
   With one cylinder cut out its contribution is lost, however the losses due to that cylinder remain the same as when it is firing, by cutting out each cylinder in turn the loss for the engine can be obtained as follows.
   I. The engine is run at the required speed and the torque is measured.
   II. One cylinder is cut out, the speed falls because of the loss of power from one cylinder.
   III. The speed is increased by reducing the load.
   IV. The torque is measured again when the speed has reached its original value.
   V. The procedure II to IV is repeated for each cylinder in turn

$$Brake\ Power\ of\ Engine = \sum_{cylinder\ one}^{final\ cylinder} Indicated\ \ power - cylinder\ loss$$

bp = Brake power with all cylinders
bp_1 = brake power with cylinder one cut out
bp_2 = brake power with cylinder two cut out
bp_3 = brake power with cylinder three cut out and so on
Therefore bp-bp_1 is the indicated power of number one cylinder Ip_1
bp-bp_2 is the indicated power of number two cylinder Ip_2
bp-bp_3 is the indicated power of number three cylinder Ip_3
and so forth for each cylinder.
The total indicated power for the engine is $Ip_1 + Ip_2 + Ip_3$ and so on.

3. Willan's line: this method is used for compression ignition engines only.
   At a constant engine speed the load is reduced in steps and the corresponding brake power and gross fuel consumption readings noted.
   A graph is drawn of fuel consumption against bp and the line is extrapolated back to cut the bp axis at the point X.
   The reading OX is taken as the power loss of the engine at that speed.
   The fuel consumption at zero bp is given by OY; if the relationship between fuel consumption and bp is assumed to be linear, then fuel consumption OY is equivalent to a power loss of OX.



Brake Mean Effective Pressure (bmep),

The brake power of an engine can be obtained accurately and conveniently using a dynamometer

$$Brake\ Power\ of\ Engine = 2\pi \times n \times T$$

$$Mechanical\ \ Efficiency\ \eta_m = \frac{Brake\ \ power}{Indicated\ \ power}$$

$$Brake\ \ Power\ \ of\ \ Engine = \eta_m \times P_{i(per\ cylinder)} \times l \times a \times n \times number\ \ of\ \ cylinders$$

Since η_M and p_i are sometimes difficult to obtain they may be combined and replaced by a brake mean effective pressure, p_b

The bmep may be thought of as that mean effective pressure acting on the pistons which would give the measured bp if the engine were frictionless it is a useful criterion for comparing engine performance.

Combining the two equations for brake power gives;

$$2\pi \times n \times T = \eta_m \times P_{i(per\ cylinder)} \times l \times a \times n \times number\ \ of\ \ cylinders$$

$$2\pi \times n \times T = P_b \times l \times a \times n$$

The engine speed "n" appears on both sides of the equation and cancels out, 2π, length and area, are constants and can be represented by K, therefore

$$P_b = KT$$

$$P_b \propto T$$

This shows that the brake mean effective pressure is directly proportional to the engine torque and is independent of the engine speed.

Thermal Efficiency

The power output of the engine is obtained from the chemical energy of the fuel supplied.

The overall efficiency of the engine is then given by the

Brake Thermal Efficiency "η_{BT}"

$$\eta_{BT} = \frac{Brake\ \ Power}{Energy\ \ Supplied} = \frac{bp}{\dot{m}_f \times Q_{LCV}}$$

where $\dot{m}_f$ is the mass of fuel consumed per unit time, and Q_{LCV} the net or lower calorific value of the fuel.

Indicated Thermal Efficiency, "η_{IT}"
This may also be used as a criteria of performance and is given by

$$\eta_{IT} = \frac{Indicated\ Power}{Energy\ Supplied} = \frac{Ip}{\dot{m}_f \times Q_{LCV}}$$

$$\dot{m}_f \times Q_{LCV} = \frac{Ip}{\eta_{IT}} = \frac{bp}{\eta_{BT}}$$

$$\frac{bp}{Ip} = \frac{\eta_{BT}}{\eta_{IT}} = \eta_m$$

$$\eta_{BT} = \eta_m \times \eta_{IT}$$

Fuel Consumption

Specific Fuel Consumption (sfc)

Is the mass flow rate of fuel consumed per unit power output, and is a criterion of economical power production.
Usually this is based on the brake power with the fuel flow being measured in Kilograms per hour or grams per hour hence we have the Brake specific fuel consumption "bsfc".

$$bsfc = \frac{Mass\ Flow\ Rate\ of\ Fuel}{Brake\ Power} = \frac{\dot{m}_f}{bp} = \frac{kg}{kWh} - or - \frac{g}{kWh}$$

Volumetric efficiency, η_v

This is defined as the ratio of the volume of air drawn into the engine to the total swept volume of the engine, both at the same pressure and temperature.

The power output of an IC engine depends directly upon the amount of air which can be drawn into the cylinder and is generally referred to as the breathing capacity of the engine, thus if a particular engine had a constant thermal efficiency then its output would be in proportion to the amount of air induced.
The volumetric efficiency of an engine is affected by many variables such as compression ratio, valve timing, induction and port design, mixture strength, specific enthalpy of vaporisation of the fuel, heating of the induced charge, cylinder temperature and atmospheric conditions.
The volumetric efficiency of a normally aspirated engine is rarely above 80%, supercharging or turbo-charging is used to improve this.

Mean Piston Speed

This parameter is used as an indication of how highly rated an engine is. In general higher piston speeds may indicate higher levels of stress and wear, however, with developments in lubrication and rubbing surface design a higher piston speed is likely to be no more detrimental than a low piston speed in a well designed engine.
It is given by the following equation where the stroke is in metres and the speed in rev/sec.

$$Mean\ Piston\ Speed = 2 \times Length\ of\ Stroke\ Engine\ Speed$$

Performance characteristics

The testing of IC engines consists of running them at different loads and speeds and taking sufficient measurements for the performance criteria to be calculated.

Energy Balance

An energy balance is sometimes presented for an engine,
The items usually included in an energy balance and expressed as percentages of the energy supplied by the fuel, are:
(a) Brake Power
(b) The heat to cooling water;
(c) The energy of the exhaust referred to inlet conditions, or as obtained by an exhaust calorimeter;
(d) Unaccounted losses obtained by difference which include radiation and convection losses, etc.

The energy balance usually presented is not an accurate account of the energy distribution but is still a useful guide.

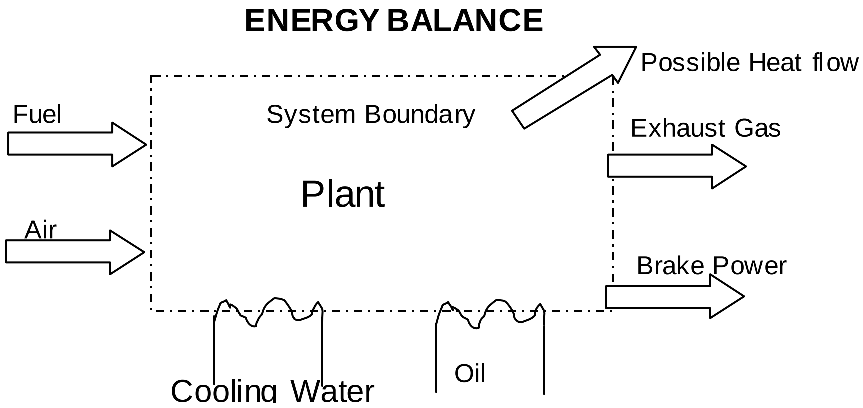
a) The brake power can be accurately measured and the percentage of the input energy to the bp is the most important item in the balance.
b) The heat transferred to the cooling water measured by flow rate and temperature rise of the water, may be used as an indication of how much heal could be usefully obtained if this was used for heating purposes.
c) The energy to exhaust is best obtained by means of an exhaust calorimeter which is simply a heat exchanger in which the exhaust gas is cooled by circulating water, the rate of flow and temperature rise of the water are measured.

Ideally the exhaust gas should be cooled to the temperature of the inlet air, hence the heat taken by the cooling water would be referred to inlet conditions, however to avoid condensation of the steam in the gas, the gas is not usually cooled below about 50°C.
The temperature at which the exhaust gas enters the calorimeter is most likely not that at which it passes through the exhaust valve, and some of the energy to the exhaust will have been taken by the cooling water or lost to the atmosphere.

To obtain the heat in the gas by calculation is not so accurate since the gas is chemically different from the inlet air and the mass flow has increased due to the addition of the fuel. However the error involved is likely to be less than that given by reading a thermometer so the energy in the exhaust gas is given by;

$$Energy\ to\ Exhaust\ Gas = (\dot{m}_a + \dot{m}_f)h_e - \dot{m}_a h_a = Joules$$

m_a and m_f are the mass flow rates of air and fuel
h_e is the specific enthalpy of the exhaust gas (dry exhaust + steam) and h_a is the specific enthalpy of the air at inlet both reckoned from 0°C,
A suitable value for c_p for the dry exhaust gas must be calculated or assumed.



The boundary is drawn around the system to include to the effects considered relevant. The steady flow energy equation is then written for the system.
The components of kinetic and potential energy can be ignored

$$\dot{Q} - \dot{W}_{bp} = (\dot{m}_a + \dot{m}_f)c_{p(gas)}(T_2 - T_0) - \dot{m}_a c_{p(air)}(T_1 - T_0) - \dot{m}_f c_{p(fuel)}(T_1 - T_0) - \dot{m}_f CV + \dot{m}_{cw} c_{p(cw)} \Delta T_{cw} + \dot{m}_{oil} c_{p(oil)} \Delta T_{oil}$$

Where
m_a = mass flow rate of air m_f = mass flow rate of fuel
m_{cw} = mass flow rate of cooling water m_{oil} = mass flow rate of oil

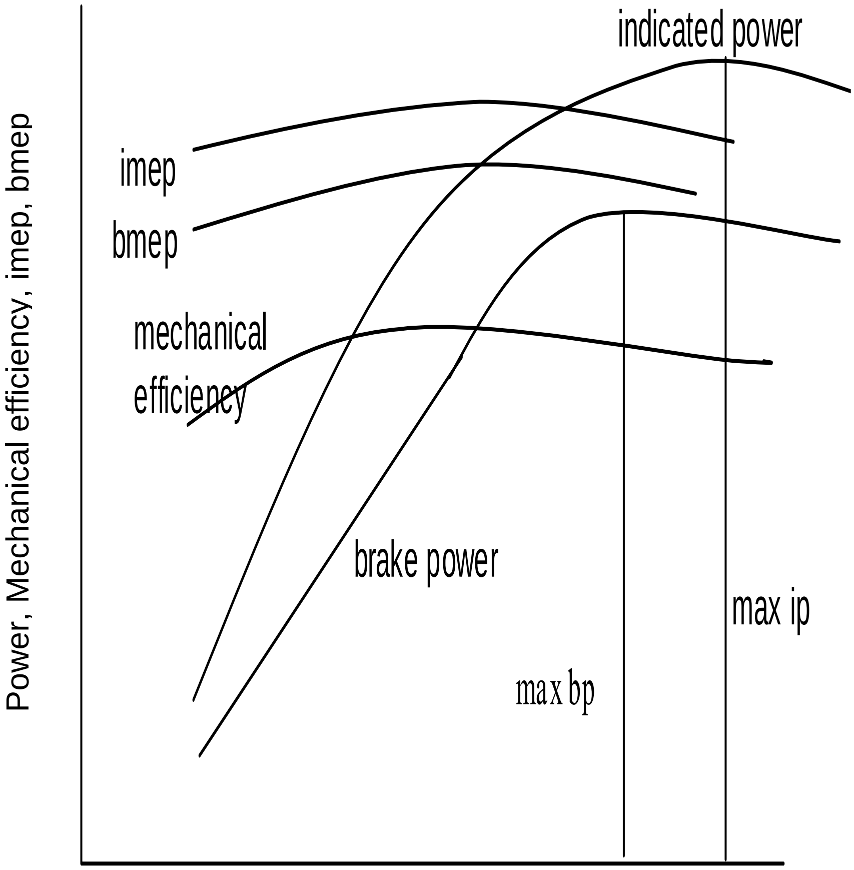
T_0 = datum temperature for enthalpy and calorific values
T_1 = air and fuel inlet temperatures T_2 = exhaust gas temperatures
ΔT_{cw} = temperature rise of cooling water ΔT_{oil} = temperature rise of oil
c_p = specific heat capacity at constant pressure of air, gas, fuel, oil and cooling water.
Q = possible heat transfer across boundary. $W_{(bp)}$= brake power.

Rate of Energy Supply			Rate of Energy Consumption		
	kW	%		kW	%
Enthalpy of Fuel			Brake Power		
Enthalpy of Air			Exhaust Gas Enthalpy		
By Combustion			Transfer to cooling water		
			Transfer to oil		
Total		100	Total		100-a
			Unaccounted loss to radiation by difference		a

If the datum temperature is the air inlet temperature then the only heat supply is that from the fuel, since the fuel temperature is not always given then this limits the heat supply to that from combustion.
For a diesel engine at full load typical -values would be: to bp 35%; to cooling water 20%; to exhaust 35%; to radiation, etc. 10%.
The heat to the jacket water is recoverable and about 18 % of the total energy supplied can be recovered from the exhaust gas.

Power Curves

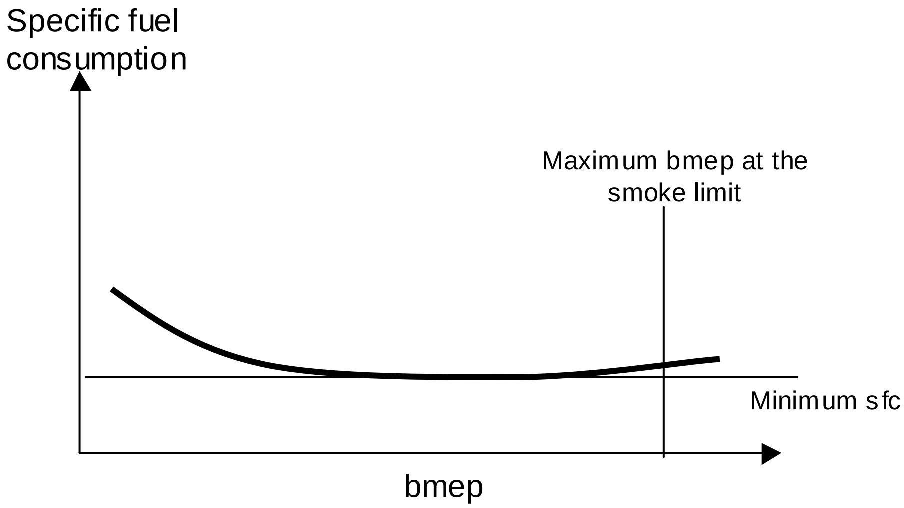
Compression ignition engines are not controlled by throttling but by adjusting the amount of fuel supplied to the engine, and hence are quality governed.
When adjusting the fuel supplied to such an engine the limiting condition is given by the smoke limit, which is the appearance of black smoke in the exhaust.
Although running with a rich mixture enough to produce smoke may give a greater power output, the efficiency under these conditions is low and the engine would soon becomes dirty. The smoke limit occurs at air-fuel ratios of about 16/1.
The engine is tested at different speeds to the smoke limit, which can be observed visually or measured by a smoke meter. The values of torque, bp, fuel consumption, and specific fuel consumption are then plotted against engine speed in revolutions per minute.
The diagram below shows typical engine power characteristics.



Engine speed

Consider the indicated power and brake power curves, as the speed increases from the lower values the two curves deviate since the friction power increases with speed, both curves show maximum values, but they occur at different speeds.
The indicated power tends to fall after the maximum value because of a reduction in the volumetric efficiency with increased speed.
Volumetric efficiency is also influenced by gas temperatures, valve timing, valve mechanism dynamics, and the pressure pulsation patterns in the induction and exhaust manifolds.
This fall in volumetric efficiency also affects the bp curve, which is further decreased by an increase in friction power.
Since the brake power reaches its maximum value at a lower speed than the indicated power the effect of friction power is shown to be predominant.
The brake mean effective pressure curve bmep will follow a similar pattern to the torque curve since bmep is directly proportional to torque as shown previously

A consumption loop for a compression ignition engine is shown below.
This shows a minimum sfc and therefore a maximum brake thermal efficiency at part load (i.e. less than the maximum bmep).
The curve is reasonably flat over a wide range of values of bmep, which shows the advantage of this type of engine compared with a spark ignition engine for part load operation.



Factors Influencing Performance of CI Engines

Effect of Compression Ratio
For combustion to occur at the temperature produced by the compression of the air a compression ratio of 12/1 is required.
The normal range of compression ratios is 13/1 to 17/1 but may be anything up to 25/1.
The efficiency of the cycle increases with higher values of compression ratio and the limit is a mechanical one imposed by the high pressures developed in the cylinder, a factor which adversely affects the power-weight ratio.

Effect of Combustion
The combustible mixture in a CI engine has to be formed at the compression stroke and after injection of the fuel has started.
This leads, to delay periods as the fuel droplets injected have to evaporate and mix with oxygen to give a combustible mixture. The delay period forms the first phase of the combustion process, and is dependent on the nature of the fuel.
The second phase consists of the spread of flame from the initial nucleus to the main body of the charge. There is a rapid increase in pressure during this phase and the rate of pressure rise depends to some extent on the availability of oxygen to the fuel spray, which in turn depends on the turbulence in the cylinder.

The main factor, however, is that of the delay period.
A long delay period means more combustible mixture has had time to form and so more of the charge will be involved in the initial combustion.
As the speed increases the rate of pressure rise in this phase also increases because the delay period is a function of time and at higher speeds more mixture will be formed in the delay period.
The initial rapid combustion can give rise to rough running and a characteristic noise called diesel knock.
During the third phase of combustion the fuel burns as it is injected into the cylinder, which gives more controlled combustion.
One of the main factors in a controlled combustion is the swirl, which is induced by the design of the combustion chamber.

Fuel Quality
The delay period depends on the nature of the fuel, and a fuel with a short delay period is required. This indicated by its cetane number for distillates and the CCAI (calculated carbon aromaticity) index for residual fuels

Air fuel ratio
The air-fuel ratios used in CI engines lie between 20/1-and 15/1.
As these mixtures are much weaker than the stoichiometric proportion then the indicated mean effective pressure will be limited, this also means that for a given fuel consumption the swept volume of the engine will be greater than that of the equivalent spark ignition engine.

Atmosphere
Engine performance is affected by the atmosphere in which they operate and some allowance must be made in figures quoted for variations in pressure, temperature, and relative humidity.
The variations in performance can be represented graphically, but the normal values quoted usually apply up to 30°C and 150 m altitude from sea level for normally aspirated engines.
The reduction in output for every 5 K above 30°C is about 3 %.

Example 1

A four cylinder petrol engine has an output of 52 kW at 2000 rev/min with a brake specific fuel consumption of 0.364 kg/kWh.
A morse test is carried out and the following brake powers were measured 35.61 kW, 35.19 kW, 36.44 kW and 37.07 kW with cylinders 1,2,3,4 cut out respectively.
The calorific value of the fuel is 44.2 MJ/kg.

Calculate the mechanical and brake thermal efficiency of the engine.

Indicated power is the difference between the original brake power and the new brake power

$$\text{Original Brake power} = 52 \text{ kW}$$

$$\text{Indicated power cylinder one} = 52 - 35.61 = 16.39 \text{ kW}$$
$$\text{Indicated power cylinder two} = 52 - 35.19 = 16.81 \text{ kW}$$
$$\text{Indicated power cylinder three} = 52 - 36.44 = 15.56 \text{ kW}$$
$$\text{Indicated power cylinder four} = 52 - 37.07 = 14.93 \text{ kW}$$

Total indicted power is the sum of the individual indicted powers

$$\text{Total indicated power} = 63.69 \text{ kW}$$

The mechanical efficiency is given by $\eta_{\text{mech}} = \dfrac{\text{Brake Power}}{\text{Indicated Power}}$

$$\eta_{\text{mech}} = \frac{52}{63.69} \times 100$$

$$\eta_{\text{mech}} = 81.64\%$$

$$\text{fuel supply} = \text{brake specific fuel consumption} \times \text{brake power}$$
$$\text{fuel supply} = 0.364 \times 52 = 18.424 \frac{\text{kg}}{\text{hour}}$$

$$\text{fuel supply} = \frac{18.424}{3600} = 5.26 \times 10^{-3} \frac{\text{kg}}{\text{second}}$$

$$\text{Heat supply from fuel} = \text{mass flow of fuel} \times \text{calorific value}$$
$$\text{Heat supply from fuel} = 5.26 \times 10^{-3} \times 44.2 \times 10^{6} = 232.5 \ kW$$

$$\eta_{\text{indicated thermal}} = \frac{\text{Indicated Power}}{\text{heat supply from fuel}}$$

$$\eta_{\text{indicated thermal}} = \frac{52}{232.5} = 0.2236$$

$$\eta_{\text{indicated thermal}} = 22.36\%$$

Example 2
A 4 cylinder 4 stroke engine has a bore and stroke of 60 mm and a rated speed of 3000 rev/min.
It was tested at this speed on a brake which has a torque arm of 0.35 m and the following data was obtained, Net brake load 155 N, fuel consumption 6.74 litres/min, fuel density 735 kg/m^3 , fuel net calorific value 44.2 MJ/kg.
A morse test was then carried out and the following brake loads were obtained with units 1,2,3 and cut out in turn, 110 N, 106 N, 104 N, 111 N.
Calculate

a) The full load engine torque
b) The brake thermal efficiency
c) The brake specific fuel consumption
d) The brake mean effective pressure
e) The mechanical efficiency
f) The indicated mean effective pressure.

The full load engine torque is obtained from the equation $\text{Torque} = \text{force} \times \text{radius}$ at the full load values.

$$\text{Torque} = 155 \times 0.35 = 54.25 \text{ Nm}$$

The brake power is obtained from the equation $\text{Brake power} = 2\pi \text{ n T}$ where T is the engine torque and n is the revs per second.

$$\text{Brake power} = 2\pi \times \frac{3000}{60} \times 54.25$$

$$\text{Brake power} = 17.043 \text{ kW}$$

The brake mean effective pressure is obtained from Power = P_{bmep} lan x number of cylinders where the power is the brake power and P_{bmep} is the brake mean effective pressure.

In the equation below $n = \frac{N}{60 \times 2}$ and $A = \frac{\pi \times D^2}{4}$

$$P_{bmep} = \frac{\text{power}}{\text{LAn} \times \text{cylinders}}$$

$$P_{bmep} = \frac{17.043 \times 4 \times 60 \times 2}{0.06 \times \pi \times 0.06^2 \times 3000 \times 4}$$

$$P_{bmep} = 10.04 \text{ bar}$$

The brake thermal efficiency is the brake power divided by the heat supply which is the mass of fuel x the calorific value.

$$\eta_{bt} = \frac{\text{brake power}}{\text{fuel mass flow} \times \text{lower calorific value}}$$

$$\text{fuel mass flow} = \frac{6.74 \times 10^{-3} \times 735}{3600} = 1.376 \times 10^{-3} \ \frac{\text{kg}}{\text{s}}$$

$$\eta_{bt} = \frac{17.043 \times 10^3}{1.376 \times 10^{-3} \times 44.2 \times 10^6}$$

$$\eta_{bt} = 0.28$$

$$\text{brake specific fuel consumption} = \frac{\text{fuel mass flow per hour}}{\text{brake power}}$$

$$\text{brake specific fuel consumption} = \frac{1.376 \times 10^{-3} \times 3600}{17}$$

$$\text{brake specific fuel consumption} = 0.2906 \ \frac{\text{kg}}{\text{kWhr}}$$

The indicated power is obtained by calculating the brake power with each of the cylinders cut out and subtracting this from the full load brake power.

Brake power with cylinder 1 cut out

$$\text{Brake power} = 2\pi \times \frac{3000}{60} \times 110 \times 0.35 = 12.095 \text{ kW}$$

Brake power with cylinder 2 cut out

$$\text{Brake power} = 2\pi \times \frac{3000}{60} \times 106 \times 0.35 = 11.655 \text{ kW}$$

Brake power with cylinder 3 cut out

$$\text{Brake power} = 2\pi \times \frac{3000}{60} \times 104 \times 0.35 = 11.435 \text{ kW}$$

Brake power with cylinder 4 cut out

$$\text{Brake power} = 2\pi \times \frac{3000}{60} \times 111 \times 0.35 = 12.205\,\text{kW}$$

Indicated power is the difference between the original brake power and the new brake power.

$$\text{Original Brake power} = 17.043 \text{ kW}$$

$$\text{Indicated power cylinder one} = 17.043 - 12.095 = 4.948\,\text{kW}$$
$$\text{Indicated power cylinder two} = 17.043 - 11.655 = 5.388\,\text{kW}$$
$$\text{Indicated power cylinder three} = 17.043 - 11.435 = 5.607\,\text{kW}$$
$$\text{Indicated power cylinder four} = 17.043 - 12.205 = 4.838\,\text{kW}$$

Total indicted power is the sum of the individual indicted powers

$$\text{Total indicated power} = 20.781\,\text{kW}$$

The mechanical efficiency is given by $\eta_{mech} = \dfrac{\text{Brake Power}}{\text{Indicated Power}}$

$$\eta_{mech} = \frac{17.043}{20.781} \times 100 = 82\%$$

The indicated mean effective pressure is obtained from Power = P_{imep} lan x number of cylinders where the power is the indicted power and P_{imep} is the indicted mean effective pressure.

$$P_{imep} = \frac{\text{power}}{\text{LAn} \times \text{cylinders}}$$

$$P_{imep} = \frac{20.781 \times 4 \times 60 \times 2}{0.06 \times \pi \times 0.06^2 \times 3000 \times 4}$$

$$P_{imep} = 12.25\,\text{bar}$$

Example 3

A four cylinder four stroke diesel engine has a bore of 212 mm and stroke of 292 mm. At a full load speed of 720 rev/min the brake mean effective pressure is 5.93 bar and the specific brake fuel consumption is 0.226 kg/ kWh.
The mass air fuel ratio as determined by exhaust gas analysis is 25:1.
Atmospheric conditions are 1.01 bar and 15°C.
For air R = 287 J/kgK
Calorific value of the fuel 44.2 MJ/kg.
Calculate the brake thermal efficiency and volumetric efficiency of the engine.

The brake power is obtained from

$$\text{Power} = P_{bmep} \text{ lan x number of cylinders}$$

where P_{bmep} is the brake mean effective pressure.

$$\text{Brake power} = 5.93\times10^5 \times 0.292 \times \frac{\pi\times0.212^2}{4}\times\frac{720}{60}\times4$$

$$\text{Brake power} = 293.386 kW$$

The brake thermal efficiency is the brake power divided by the heat supply which in turn is obtained from the fuel flow and the calorific value.
The fuel flow is obtained from the brake power and brake specific fuel consumption.

$$\text{brake specific fuel consumption} \times \text{brake power} = \text{Fuel mass flow per hour}$$

$$\text{fuel mass flow} = \frac{0.226\times293.386}{3600} = 18.41\times10^{-3}\ \frac{\text{kg}}{\text{s}}$$

$$\eta_{bt} = \frac{\text{brake power}}{\text{fuel mass flow}\times\text{lower calorific value}}$$

$$\eta_{bt} = \frac{293.386\times10^3}{18.41\times10^{-3}\times44.2\times10^6}$$

$$\eta_{bt} = 0.36$$

Therefore the brake thermal efficiency is 36%

The volumetric efficiency is given by the actual air induced at inlet conditions divided by the swept volume.

$$\text{Actual air induced} = \text{Fuel flow} \times \text{air / fuel ratio}$$

$$\text{Actual air induced} = 18.41 \times 10^{-3} \times 25 = 0.46 \frac{\text{kg}}{\text{s}}$$

The volume of air induced at inlet conditions is obtained from pV=mRT

$$\text{Induced volume } V_i = \frac{0.46 \times 287 \times 288}{1.01 \times 10^5} = 0.376 \frac{m^3}{s}$$

$$\text{Swept volume } V_s = \frac{\pi \times 0.212^2}{4} \times 0.292 \times \frac{720}{60} \times 4 = 0.494 \frac{m^3}{s}$$

$$\eta_v = \frac{\text{Induced Volume}}{\text{Swept Volume}} = \frac{0.376}{0.494} = 0.76$$

The volumetric efficiency is therefore 76% which is reasonable for a high speed naturally aspirated engine.

Example 4

The following data refers to a 10 cylinder two stroke single acting marine diesel engine undergoing a shop trial.

Length of stroke	1800 mm	Fuel consumption	3450 kg/hour
Cylinder bore	900 mm	Calorific value of fuel	44 MJ/kg
Speed	120 Rev/min	Exhaust gas flow	112 tonne/hour
Area of indicator card	7.6 cm	Cooling water flow	224 tonne/hour
Spring stiffness	10 bar/cm	Lubricating oil flow	120 tonne/hour
Net brake load	2 MN	Inlet air temperature	30°C
Effective brake radius	580 mm	Exhaust temperature	420°C
Oil inlet temperature	40°C	Cooling water inlet	40°C
Oil outlet temperature	55°C	Cooling water outlet	80°C

Specific heat capacity of cooling water 4.2 kJ/kgK
Specific heat capacity of oil 2.1 kJ/kgK
Specific heat capacity of exhaust gas 1.1 kJ/kgK

Take the area of the indicator card to be the same for all the cylinders.

a) Calculate
   i) The indicated power
   ii) The brake power
   iii) The mechanical efficiency
   iv) The brake thermal efficiency

b) Draw up an energy balance for the plant in terms of the total energy supplied.

$$\text{Indicated mean effectve pressure} = \frac{\text{Diagram area}}{\text{Diagram length}} \times \text{Spring cons tan t}$$

$$Indicated\ mean\ effectve\ pressure = \frac{7.6}{7.6} \times 10 = \frac{cm^2}{cm} \times \frac{bar}{cm} = bar$$

$$Indicated\ mean\ effectve\ pressure = 10\ \text{bar}$$

$$Indicated\ \ Power\ = P_i \times length \times area \times n$$

$$Indicated\ \ power\ per\, cylinder = 10 \times 10^5 \times 1.8 \times \frac{\pi \times 0.9^2}{4} \times \frac{120}{60}$$

$$Indicated\ \ power\ per\, cylinder = 2.29 \times 10^6\ \ \text{W}$$

Since each indicator card has the same area then the total power is that for one cylinder times the number of cylinders.
If the indicator card area was different for each cylinder then the above calculation would be carried out for each cylinder and the result totalled.
In this case only one area has been given and we have been told it is the same for all cylinders.

$$Indicated\ \ power\ \ for\, Engine = 2.29 \times 10^6 \times 10 = 22.9\ \ \text{MW}$$

Brake power is obtained from the torque output

$$Brake\ Power\, of\ \ Engine\ = 2\pi \times n \times T$$

In this case the brake load and radius have been given so we can put torque as force times radius

$$Brake\ Power\, of\ \ Engine\ = 2\pi \times \frac{120}{60} \times 2 \times 10^6 \times 0.58$$

$$Brake\ Power\, of\ \ Engine = 14.577 \times 10^6\ \ \text{W}$$

$$Mechanical\ \ Efficiency\ \eta_m\ = \frac{Brake\ \ power}{Indicated\ \ power}$$

$$Mechanical\ Efficiency\ \eta_m\ = \frac{14.577}{22.9} = 0.636$$

Therefore the mechanical efficiency= 63.6%

Brake thermal Efficiency

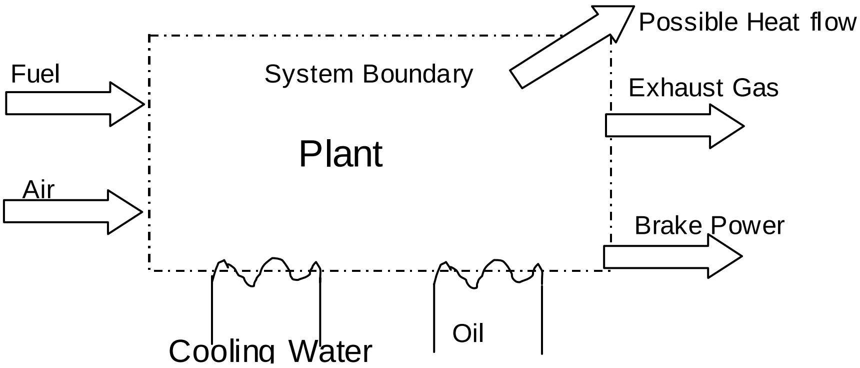
$$\eta_{BT} = \frac{Brake\ \ Power}{Energy\ \ Supplied} = \frac{bp}{\dot{m}_f \times Q_{LCV}}$$

$$\eta_{BT} = \frac{14.577 \times 10^6 \times 3600}{3450 \times 44.2 \times 10^6} = 0.344$$

Remember the 3600 on the top line is to bring the mass flow in kg/hour to kg/sec which makes the units compatible.

Therefore the brake thermal efficiency is 34.4%

ENERGY BALANCE



$$\dot{Q} - \dot{W}_{bp} = (\dot{m}_a + \dot{m}_f)c_{p(gas)}(T_2 - T_0) - \dot{m}_a c_{p(air)}(T_1 - T_0) - \dot{m}_f c_{p(fuel)}(T_1 - T_0) - \dot{m}_f CV + \dot{m}_{cw} c_{p(cw)} \Delta T_{cw} + \dot{m}_{oil} c_{p(oil)} \Delta T_{oil}$$

The above diagram shows the distribution of energy in the system while the steady flow energy equation allows us to calculate the relevant values.

In this case a datum temperature has not been specified so the enthalpy of the air at inlet can be eliminated and the enthalpy of the exhaust gas referred to the air temperature at inlet.
The enthalpy of the fuel at inlet can also be ignored since no information has been given.

$$\text{Heat supply from fuel} = \text{mass flow of fuel} \times \text{calorific value}$$

$$\text{Heat supply from fuel} = \frac{3450}{3600} \times 44.2 \times 10^6 = 42.36\ \ MW$$

Heat in exhaust gas

$$\dot{Q}_{exhaust\ gas} = m_{gas} c_{p(gas)} \left(T_{gas} - T_{air}\right)$$

$$\dot{Q}_{exhaust\ gas} = \frac{112 \times 10^3}{3600} \times 1.1 \times 10^3 (420 - 30)$$

$$\dot{Q}_{exhaust\ gas} = 13.35\,\text{MW}$$

Heat in cooling water

$$\dot{Q}_{Cooling\ water} = m_{water} c_{p(water)} \left(T_{out} - T_{in}\right)$$

$$\dot{Q}_{cooling\ water} = \frac{224 \times 10^3}{3600} \times 4.2 \times 10^3 (80 - 40)$$

$$\dot{Q}_{cooling\ water} = 10.453\,\text{MW}$$

Heat in the oil

$$\dot{Q}_{oil} = m_{oil} c_{p(oil)} \left(T_{out} - T_{in}\right)$$

$$\dot{Q}_{oil} = \frac{120 \times 10^3}{3600} \times 2.1 \times 10^3 (55 - 40)$$

$$\dot{Q}_{oil} = 1.05\,\text{MW}$$

The energy balance is best shown in a table.
The total energy supplied is from the fuel, the energy distribution has then been calculated and shown as a percentage of the total supplied.

Rate of Energy Supply			Rate of Energy Consumption		
	MW	%		MW	%
By Combustion	42.36	100	Brake Power	14.58	34.42
			Exhaust Gas Enthalpy	13.34	31.5
			Transfer to cooling water	10.45	24.67
			Transfer to oil	1.05	2.48
Total		100	Total	39.42	93.07
			Unaccounted loss to radiation by difference	2.94	6.93