

Powertrain Technology

AIMS

- To allow the student to acquire basic knowledge of a range of powertrain systems and associated technologies.
- To assess their impact and function on overall vehicle design, underpinned by the engineering science fundamentals.

LEARNING OUTCOMES

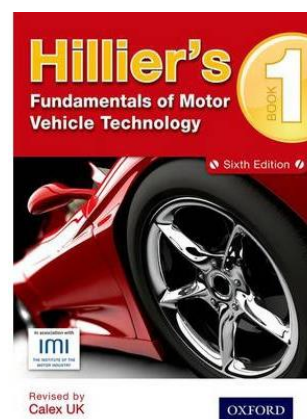
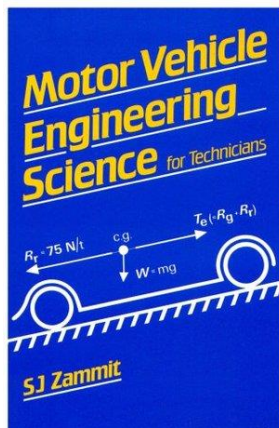
Upon the successful completion of this module, you should be able to demonstrate the ability to:

1. Show a detailed knowledge and understanding of the underlying principles of engine and vehicle technologies and practices for analysis powertrain systems.
2. Apply subject specific skills with respect to providing solutions using engineering principles applied to an automotive problem and the interpretation of experimental data to assess validity of solutions proposed.
3. Show cognitive skills with respect to simplification and modelling of real automotive problems, the application of fundamental mechanical engineering principles and the analysis of problems and making recommendations on findings.
4. Demonstrate transferable skills in problem formulation and decision making.

Assessment

Two pieces of coursework

1. Engine dynamometer testing and data analysis
2. Chassis dynamometer and data analysis.



Semester 1

Vehicle Emissions

The burning of hydrocarbons (Petrol or Diesel) creates several by-products, what is the content of exhaust gases from an internal combustion engine?.....

[illegible]

Emissions legislation

Emissions data history is fairly recent (compared with how long we have had vehicles), since emissions regulations have only been mandatory since 1982, and then were not up-dated until 1992, with the introduction of the 'Euro' regulations by the European Union.

Below are two tables, Tables 1 and 2, containing the data for both Petrol and Diesel internal combustion engines emissions from 1982 up to the present Euro vi standard.

Year of Regulation	CO (g/km)	HC & NOx (g/km)	HC (g/km)	NOx (g/km)
1982	20.7	5.8	-	-
1992 (Euro 1)	2.72	0.97	-	-
1996 (Euro 2)	2.2	0.5	-	-
2000 (Euro 3)	2.3	0.35	0.2	0.15
2005 (Euro 4)	1.0	0.18	0.1	0.08
2009 (Euro 5)	1.0	0.16	0.1	0.06
2014 (Euro 6)	1.0	0.16	0.1	0.06

Table 1: Petrol Emissions

Data are shown graphically in figure 1 and illustrate the reduction of emissions due to developments in legislation.

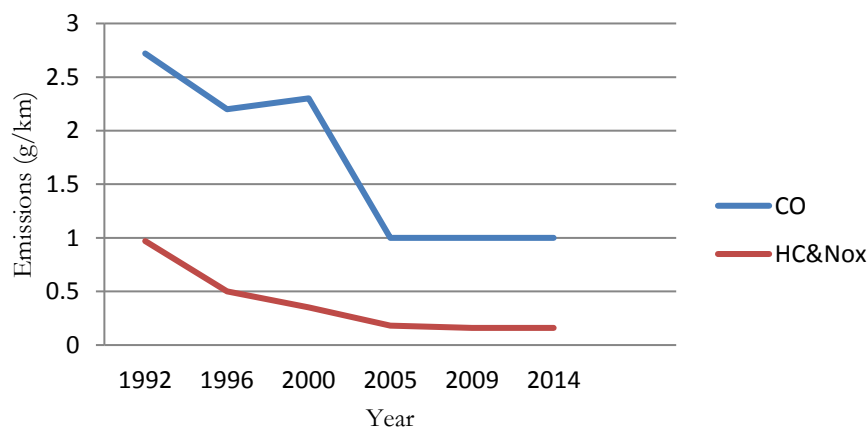


Figure 1: Petrol Emission Regulations Over Time

Year of Regulation.	CO (g/km)	HC & NOx (g/km)	HC (g/km)	NOx (g/km)	Particulate Matter (PM) (g/km)
1982	20.7	5.8	-	-	-
1992 (Euro 1)	2.72	0.97	-	-	0.14
1996 (Euro 2)	1.0	0.7	-	0.8	0.08
2000 (Euro 3)	0.64	0.56	0.5	0.5	0.05
2005 (Euro 4)	0.5	0.3	0.25	0.25	0.025
2009 (Euro 5)	0.5	0.23	-	0.18	0.005
2014 (Euro 6)	0.5	0.17	-	0.08	0.005

Table 2: Diesel Emissions

(www.europa.eu.int, 2004)

Data are shown graphically in figure 2.

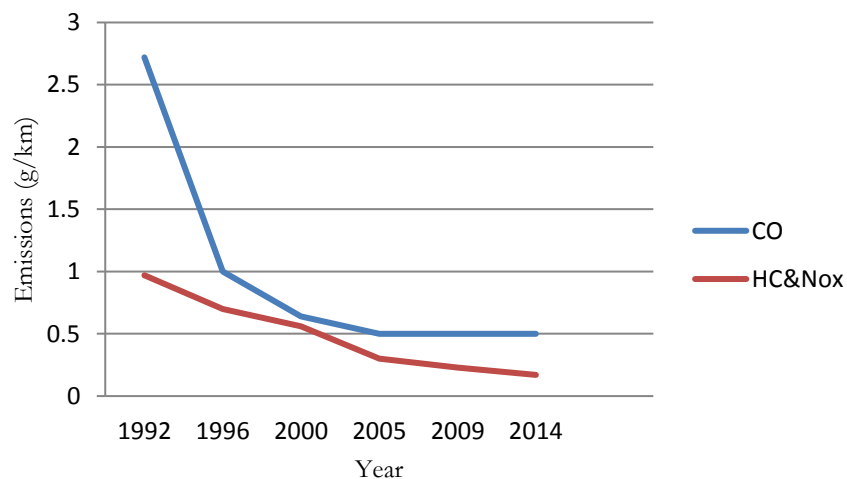


Figure 2: Diesel Emission Regulations Over Time

Both Figure 1 and Figure 2 illustrate the reduction of emissions required since the mandatory regulations were introduced. Each manufacturer has had to conform to the legislation with every

vehicle in their range if the car is to be sold in the E.U. If the car is of a higher emission output then the owner pays a percentage of tax based on the vehicle's emissions, a ruling that will be in place until April 2017. (Haymarket Motoring, 2004)

Prior to 2003, the main method for measuring tailpipe emissions was to take an instantaneous value, this measurement being performed within 40 seconds of the engine being started. It is within this period that a vehicle emits the highest volume of unwanted emissions (Blix 2004).

Since 2003, test cycles have been devised to synthetically generate a typical driving curve, including an urban traffic cycle and a suburban section. This idea has been developed and legislated on by the E.U. as the ECE/EC test cycle and is currently required by law in the European countries.

Figures 3 and 4 below show the ECE/EC Euro v test cycle, over which emissions measurements are taken. Figure 3 concerns petrol powered passenger cars and light commercial vehicles, whilst Figure 4 concerns diesel engined equivalents and heavier commercial vehicles. These figures also show how the regulations relate to vehicles of specific weights.

The petrol cycle is 1220secs (20 minutes) long; it covers a distance of 11 km; averages a speed of 32.5km/h and reaches a top speed of 120km/h. The diesel cycle lasts for 1180secs, and also has an average speed of 32.5km/h and a top speed of 120km/h (Bosch, 2000).

Looking at the first 850 second of the petrol emissions cycle (Figure 3) the chart shows a series of peaks increasing from around 15km/h then 35km/h and finally 50km/h between each peak the vehicle comes back to rest. This series of peaks is repeated four times. This part of the drive cycle simulates urban driving, with a 'start stop' pattern that a driver might experience whilst driving through town, travelling through congestion and traffic management e.g. traffic light systems. From 850 seconds until the end of the test, the cycle accelerates up to 70km/h maintains this speed for a short period, drops to 50km/h for a short period, and then accelerates up to 120km/h before returning to rest. This section of the cycle represents a suburban driving pattern, where the driver is now making progress on a more open highway e.g. dual carriageway and experiencing less congestion.

The diesel cycle (Figure 4) mimics the petrol cycle very closely but simulates slightly higher top speeds in the urban driving section of the test.

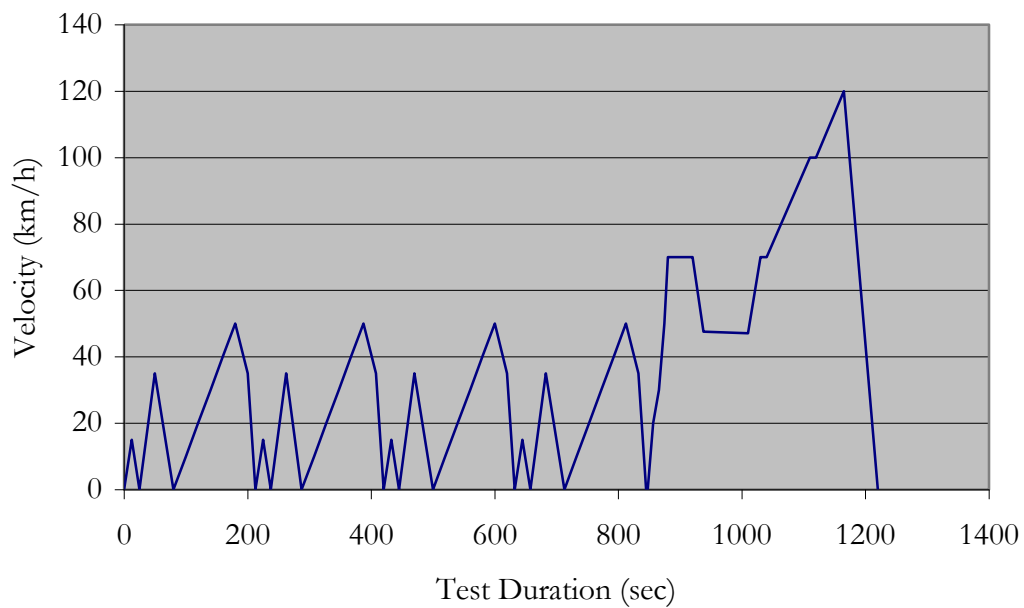


Figure 3: Petrol Emissions Test Cycle

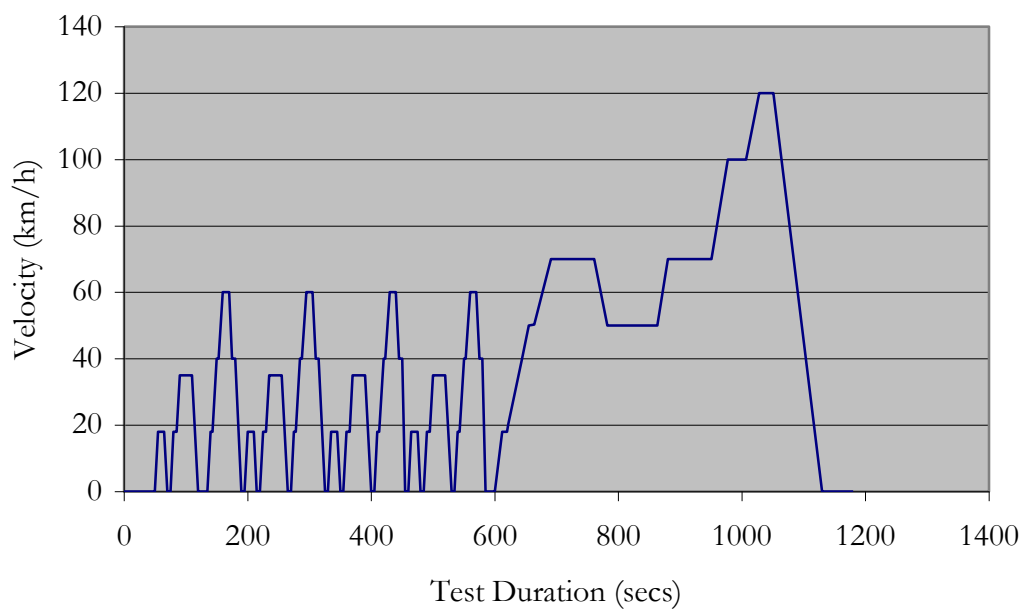


Figure 4: Diesel Emissions Test Cycle

Criticism

Inability to represent real-life driving: The NEDC was conceived at a time when European vehicles were lighter and less powerful. The test offers a stylised driving speed pattern with low rates of acceleration, constant speed cruises, and many idling events, however the transient accelerations are much steeper and more dynamic in practice,^[6] in part caused by the power surplus of modern engines, as the 0–100 km/h average time decreased from 14 seconds in 1981 to 9 seconds in 2007. Thus, drivers fail to achieve the certified values in practice.

Cycle beating: For the emission standards to deliver real emission reductions it is crucial to use a test cycle that reflects real-world driving style. However, the fixed speeds, gear shift points and accelerations of the NEDC offer possibilities for manufacturers to engage in what was called 'cycle beating' to optimise engine emission performance to the corresponding operating points of the test cycle, while emissions from typical driving conditions would be much higher than expected, undermining the standards and public health. In one particular instance, research from two German technology institutes found that for diesel cars no 'real' NO_x reductions have been achieved after 13 years of stricter standards.

The Volkswagen emissions scandal, devalued the EDC after VAG programmed engine management software to detect test conditions, and alter the emission control when under these conditions. So although the cars would pass the test, during actual driving conditions, the cars emitted up to 40 times more emissions.

Alkane Derived Fuels

If these fuels are burned with enough oxygen the only emissions are CO₂ and H₂O:

- $2C_8H_{18} + 25O_2 = 16CO_2 + 18H_2O$
(Petrol + Oxygen = Carbon Dioxide + Water)
- $2C_{12}H_{26} + 37O_2 = 24CO_2 + 26H_2O$
(Diesel + Oxygen = Carbon Dioxide + Water)

Water vapour and carbon dioxide are not harmful to the environment directly, but are greenhouse gases. The production of CO₂ in particular needs to be lowered in an attempt to reduce the observed increase in CO₂ levels in the atmosphere. (Kiehl, J.T, and Trenberth, K, 1997).

If combustion is not complete, i.e. there is not enough oxygen present during burning, then other chemicals are produced, namely carbon monoxide, hydrocarbons, and Nitrous Oxides. These chemicals are harmful to plants and animals as described earlier and hence it is important to keep production of these chemicals to a minimum. There are different degrees of incomplete combustion; the less oxygen there is present whilst combustion is taking place, the worse the resulting emissions (Nelson, 1978).

Below are the variations of incomplete combustion for both petrol and diesel.

Petrol:

- $C_8H_{18} + 12O_2 = 7CO_2 + CO + 9H_2O$
(Petrol + Oxygen = Carbon Dioxide + Carbon Monoxide + Water)
- $C_8H_{18} + 10O_2 = 6CO_2 + CO + 7H_2O + CH_4$
(Petrol + Oxygen = Carbon Dioxide + Carbon Monoxide + Water + Hydro Carbons)
- $C_8H_{18} + 7O_2 = 7CO + 7H_2O + CH_4$
(Petrol + Oxygen = Carbon Monoxide + Water + Hydro Carbons)

Diesel:

- $C_{12}H_{26} + 18O_2 = 11CO_2 + CO + 13H_2O$
(Diesel + Oxygen = Carbon Dioxide + Carbon Monoxide + Water)
- $C_{12}H_{26} + 11O_2 = 10CO_2 + CO + 11H_2O + CH_4$
(Diesel + Oxygen = Carbon Dioxide + Carbon Monoxide + Water + Hydro Carbons)
- $C_{12}H_{26} + 11O_2 = 11CO + 11H_2O + CH_4$
(Diesel + Oxygen = Carbon Monoxide + Water + Hydro Carbons)

Nitrogen oxides are also produced as a result of lack of oxygen during combustion. NO and NO₂ are produced from the nitrogen present in air, combined with oxygen during combustion.

Another toxin produced during Diesel combustion is sulphur dioxide, SO₂ (described in chapter 1) that is present in the fuel and comes out as an emission as a result of the combustion process.

All alkanes produce the same types of emission but in different quantities depending on the amount of oxygen present during combustion. Hence alkanes are problematic as fuels for internal combustion engines as there will always be undesirable bi-products of the combustion process, even if combustion is stoichiometric.

Engine Performance Testing and Analysis**Calculation of the Engine Capacity.**

Engines come in varying sizes; the larger the engine the more fuel it can burn in one engine cycle. The size of the engine is known as its capacity and is measured in either litres or cc.

Calculation of the capacity of the engine can be performed using a small amount of data from the engine measurements.

The area of your bore/cylinder can be calculated using the simple formula for the area of a circle:

$$\text{Area of a circle} = \frac{\pi d^2}{4}$$

Where d is the diameter of the cylinder bore, and can be measured by hand.

The distance the piston travels up and down the bore is called the stroke and is represented by the letter 'l', this can also be measured by hand.

So the engine capacity is calculated using the following formula:

$$\text{Engine Capacity} = \frac{\pi d^2}{4} \times l \times 4$$

Example

Given that a four cylinder engine has a cylinder diameter of 83mm and a stroke of 88mm, determine its capacity.

.....

.....

.....

.....

.....

.....

.....

.....

.....

This figure is not entirely accurate as there is a portion scooped out of the cylinder head known as the combustion chamber that contributes to this capacity but can only be obtained by measuring.

Engine Performance Testing and Analysis

Engine Testing

Use of an engine dynamometer to measure standard engine variables such as torque, engine speed, time for fuel consumption, ignition timing, manifold pressure and indicated power.

Calculation and plotting of engine variables such as engine power, engine torque, SFC, thermal efficiency and volumetric efficiency against engine speed.

Calculation of BMEP and IMEP, from engine readings and hence calculation of mechanical efficiency.

Recommended Reading:

- Stone R-Introduction to Internal Combustion Engines (Macmillan)
- Heisler Heinz-Vehicle and Engine Technology (Edward Arnold)
- Zammit SJ-Motor Vehicle Science for Technicians (Longman)

Engine Dynamometer

A dynamometer, is a device for measuring force, moment of force (torque), or power. the power produced by an engine, motor or other rotating prime mover can be calculated by simultaneously measuring torque and rotational speed (rpm). A dynamometer can also be used to determine the torque and power required to operate a driven machine such as a pump. In that case, a motoring or driving dynamometer is used. A dynamometer that is designed to be driven is called an absorption or passive dynamometer. A dynamometer that can either drive or absorb is called a universal or active dynamometer. (setra.com, 2019)

Engine Testing: Important Formulae

So given knowledge of the Four Stroke Cycle the internal work done by expanding gas each engine power stroke is given by:

$$W.D = \frac{MEPLAn}{2} \dots\dots\dots \text{Equation 1}$$

Where:

W.D is the Work done (J)

MEP is the mean effective pressure (Pa)

L is the engine stroke (m)

A is the piston area (m²)

n is the number of cylinders

2 is the idle stroke (every other stroke with four strokes)

What is the equation for 'Work done'?

$W.D = \text{Force} \times \text{Distance}$

So in terms of engines:

$W.D \text{ per engine revolution} = 2\pi RW$

Where $2\pi R$ is the distance in a circular motion (circumference) and W is the load on the engine i.e. the force.

So in engine terms (circular motion) the formula becomes:

$T = WR$

Where T is the Torque produced by the engine (Nm)

W is the engine load (N)

R is the radius at which the load acts (m)

So now **Work done per engine revolution $= 2\pi T$ Equation 2**

So by Equating equations 1 and 2

$$2\pi T = \frac{(MEP)LAN}{2}$$

$$T = \frac{(MEP)LAN}{4\pi} \dots\dots\dots \text{Equation 3}$$

Power = Work done per second so from Equation 2

$$P = \frac{2\pi NT}{60} \dots\dots\dots \text{Equation 4}$$

Where P is the power developed at the engine (W)

and N is the engine speed (r/m)

So substituting for T from Equation 3

$$P = \frac{(MEP)LANn}{60 \times 2} \dots\dots\dots \text{Equation 5}$$

Indicated Mean Effective Pressure (IMEP)

This is the mean of the pressure actually available within the cylinders to do work throughout the stroke. It is a measure of the effectiveness of the combustion process and allows different engines to be compared.

It can be found by calculation after finding indicated power by the Morse test method or by calculation. Typical value 1200kPa

The Morse Test

Another way of measuring the indicated power of a multi cylinder internal combustion engine is by means of the Morse Test.

- In this method an engine coupled to a dynamometer is run up to a specific rpm and the brake horse power is measured for all the cylinders running.
- Following this measurement in turn each cylinder is 'cut-out'.
- In the case of a spark ignition engine this is done by short circuiting the spark to each cylinder in turn.
- In the case of a compression ignition engine, the interrupting the fuel supply to the cylinder.
- The loss of a cylinder will cause the power to drop and hence the load on the dyno' must be reduced to restore the engine to its original rpm.
- The brake horse power is then measured again with each cylinder cut-out and the difference between the two powers is the equivalent to the indicated power of the cut out cylinder.
- This procedure repeated for each of the engine cylinders will eventually give the indicated power for the whole engine.
- This must be performed relatively quickly to avoid any changes to the running caused by the cooling of the cut out cylinder.

So with all cylinders firing:

Brake Power = Indicated Power – Friction Power.

For a four cylinder engine

I.e. Power out = ($Ip_1 + Ip_2 + Ip_3 + Ip_4$) – Friction Power.....(1)

With one cylinder out:

Power out = ($Ip_2 + Ip_3 + Ip_4$) – Friction Power.....(2)

So $(1) - (2) = I_{p1}$

And is repeated for the remaining cylinders to obtain the total indicated power.

Also

$I_p = (\text{total bp} \times \text{No of cylinders}) - (\text{the sum of the bp with each of the cylinders out})$

Brake Mean Effective Pressure (BMEP)

This is the equivalent of the mean of the pressure available throughout the stroke to do mechanical work after all frictional losses have been taken into account. It is proportional to the engine torque and enables the performance of different engines to be compared irrespective of their capacity. It can be found by calculation after finding the brake power at the flywheel.

Typical value 900kPa

Typical Peak Pressure Values:

SI Engine: 7000kPa

Diesel: 14000kPa

F1 Turbo: 28000kPa

Specific Fuel Consumption (SFC)

SFC is measured in kilograms of fuel used per kilowatt hour (kg/kWh) and is an important measure of an engine's efficiency in producing power. It enables the designer to determine the best operating conditions for the engine as well as estimating the running costs of the vehicle. Although it is rarely quoted for passenger cars it is widely used by truck engineers where the cost of carrying a given load is critical.

Indicated SFC = $\frac{M'}{IP}$

Brake SFC = $\frac{M'}{BP}$

IP

BP

Where M' is the mass of fuel used per hour

IP is the indicated power (kW)

BP is the brake power (kW)

To obtain SFC the time taken in seconds to use a known quantity of fuel (cc) is recorded.

$$SFC = \frac{V \times 0.75 \times 3600}{t \times 1000 \times P}$$

Where:

1000 converts fuel volume in cc to litres,

0.75 is the Specific Gravity of the fuel and converts litres of fuel to kg,

3600 converts fuel used per second to fuel used per hour,

V is the volume of fuel being measured (cc),

t is the time taken to use the fuel (s).

Typical value 0.3-0.5kg/kWh

For electronic fuel injection SFC can also be found from the injection pulse width.

Mechanical Efficiency (η_m)

This is a measure of the frictional losses in an engine. The higher the efficiency the lower frictional losses.

$$\eta_m = \frac{BP}{IP} \quad \text{also} \quad \eta_m = \frac{BMEP}{IMEP}$$

And hence

$$FP = IP - BP$$

Where FP is frictional power

IP is indicated power

BP is brake power

Brake Thermal Efficiency (η_t)

Brake thermal efficiency is an arbitrary, overall measure of the ability of an engine to convert fuel energy into mechanical energy. Engines do not convert all of their heat energy supplied by the fuel into useful work, approximately 1/3 of the heat value of fuel is converted into mechanical work leaving the remaining 2/3 to be lost to exhaust gas, cooling system and radiation to the surrounding area.

Thermal efficiency is dependent on the following factors:

- i. Mixture strength
- ii. Compression ratio

- iii. Engine speed
- iv. Throttle position
- v. Valve timing
- vi. Ignition timing
- vii. Fuel type
- viii. Dimensions of the engine
- ix. Combustion chamber design
- x. Inlet charge temperature
- xi. Cylinder temperature

η_t of an engine is a ratio: $\frac{\text{heat} - \text{energy} - \text{equivalent} - \text{to} - W.D / \text{sec}}{\text{heat} - \text{energy} - \text{supplied} - \text{from} - \text{fuel} / \text{sec}}$

But for the purpose of calculating the η_t from measurements taken whilst dyno' testing:

$$\eta_t = \frac{3600 \times 1000}{SFC \times CV}$$

Where SFC is the specific fuel consumption (kg/kWh)

and CV is the calorific value of fuel (kJ/kg)

Typical value 20%-30% for an SI engine

Volumetric Efficiency η_v

This is a measure of the effectiveness of the induction and exhaust system in filling the cylinder with charge i.e a volumetric efficiency of 100% would indicate that during each cycle the cylinder is completely filled with air at standard temperature and pressure (STP). It can also be explained as the ratio of actual charge admitted into the cylinder at STP to the volume swept by the piston:

$$\eta_v = \frac{\text{airflow into engine} / \text{sec}}{\text{swept volume} / \text{sec}}$$

The η_v rarely exceeds 80% for a n/a engine.

Factors influencing it can be:

- i. Engine speed
- ii. Compression ratio
- iii. Inlet charge temperature
- iv. Mixture strength
- v. Inlet and exhaust design
- vi. Valve area
- vii. Valve timing
- viii. Throttle opening
- ix. Cylinder temperatures.

Power Correction for Varying Atmospheric Conditions

To enable tests to be carried out under different atmospheric conditions to be compared a correction factor is used to modify the actual power figures obtained. Although this is an accepted European practice (DIN Standard 70020) it does not have any absolute validity as some engines may not respond to atmospheric changes in this way.

Engine power output is affected by air charge density (hence why your car might feel quicker on a cool crisp morning compared with a warm humid afternoon) and by the flow conditions into the engine. The flow conditions are affected by the Mach number (speed of charge flow relative to the speed of sound)

The power correction factor used is given by:

Power Correction Factor = Air density factor x Mach number factor

Which, when simplified gives a factor of:

$$\frac{P_s}{P} = \frac{1.01325}{p_a} \sqrt{\frac{T_a}{293}}$$

Where:

P_s is the corrected power

P is the measured power

p_a is the ambient pressure

T_a is the ambient temperature

1.01325 is standard pressure in bar

293 is standard temperature in Kelvin

Example Sheet 1: Engine Performance**Example 1.**

A four stroke cycle engine develops a Torque of 220Nm at an engine speed of 5200rpm.

Calculate the power.

[119.8Kw]

Example 2.

A four stroke cycle engine produces 80 kW of power at 6200rpm, Calculate:

(i) The torque developed by the engine.

(ii) The load on the engine supplied by a dynamometer with a torque arm of radius 0.25m.

[123.22Nm, 492.87N]

Example 3.

Determine the Indicated power of a four stroke four cylinder engine of the following:

Bore = 0.06m

Engine speed = 3200rpm

Stroke = 0.088m

MEP = 600kN/m².

[15.77kW]

Example Sheet 2: Engine Performance

1. An engine rotating at a speed of 3000rpm lifts a dynamometer load of 87.5N. If the length of the torque arm is 0.4m, calculate the torque and brake power of the engine (*35Nm, 11kW*)
2. Given the following engine dynamometer readings, calculate the brake power in kW: Load = 84N, Length of Torque Arm = 0.5m, Engine Speed = 100 rev/sec (*26.4kW*)
3. During an engine test a single cylinder ran at 2500rpm, against a load of 75N on a dynamometer with a torque arm radius of 0.35m. The IMEP was 860 kPa. Given that the bore is 75mm and the stroke is 100mm, determine the Indicated Power, Brake Power, Frictional Power and Mechanical Efficiency (*7.92kW, 6.87kW, 1.05kW, 86.7%*)
4. An engine on test at 2800rpm overcomes a resistance of 165.5N acting at a radius of 0.355m while consuming fuel at the rate of 6.9 litres/hour. If 1 litre/hour is the equivalent of 12kW determine the mechanical efficiency. (*20.5%*)
5. A Morse Test on a 6 cylinder petrol engine gave the following results when the speed was maintained at 2000rpm.

Cylinder Firing	Not Firing	None	1	2	3	4	5	6
Brake Power (kW)		40.65	32.44	31.32	32.44	32.80	32.10	31.70

Determine the indicated power, Frictional Power and the Mechanical Efficiency. (*51.1kW, 10.45kW, 79.55%*)

6. An engine dyno test gives a power measurement of 43.2kW. However, the temperature in the test cell is 32 degrees Celsius, and the ambient pressure is 1.011bar. What is the corrected power according to the DIN standard? (*44.2kW*)

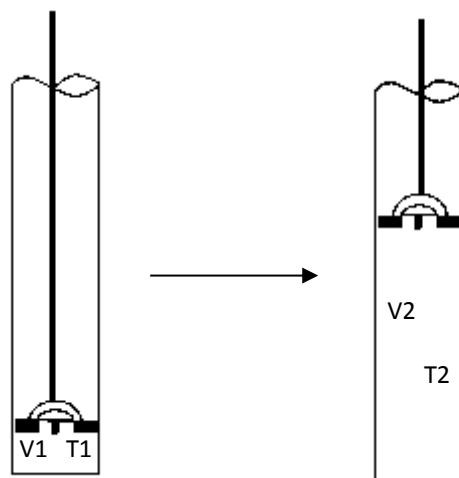
Gas Laws Lesson
Engineering Science

Charles Law.

'At constant pressure, the volume of a given mass of an ideal gas increases or decreases by the same factor as its temperature on the absolute temperature scale.'

E.g. If, P, the pressure remains constant for every °C rise the temp the volume expands.
Charles' Law describes the direct relationship of temperature and volume of a gas.

So imagine if you double the temperature of the air in a plastic bottle of a certain mass of gas, in simple terms, the particles become excited and move about more, effectively taking up more double volume.



Therefore: $V_1/T_1 = V_2/T_2$ where the pressure of the gas is constant, and

V_1 = Initial Volume (m^3)

V_2 = Final Volume (m^3)

T_1 = Initial Temperature (K)

T_2 = Final Temperature (K)

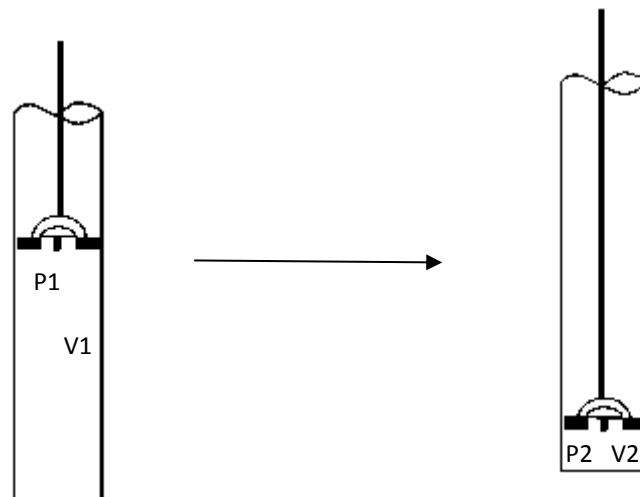
To convert °C to K add 273.

Boyle's Law.

'The Volume of a gas of fixed mass, is inversely proportional to the pressure, provided the temperature remains constant.'

E.g. If, T, the temperature remains constant and P, the pressure increases then V the volume filled by the gas decreases.

So imagine a volume of gas in a plastic bottle, if a pressure is applied to the bottle then the volume taken up by the gas gets less and the bottle will contract.



Therefore: $P_1V_1 = P_2V_2$ where the temperature of the gas is constant, and

V_1 = Initial Volume (m^3)

V_2 = Final Volume (m^3)

P_1 = Initial Pressure (Pa)

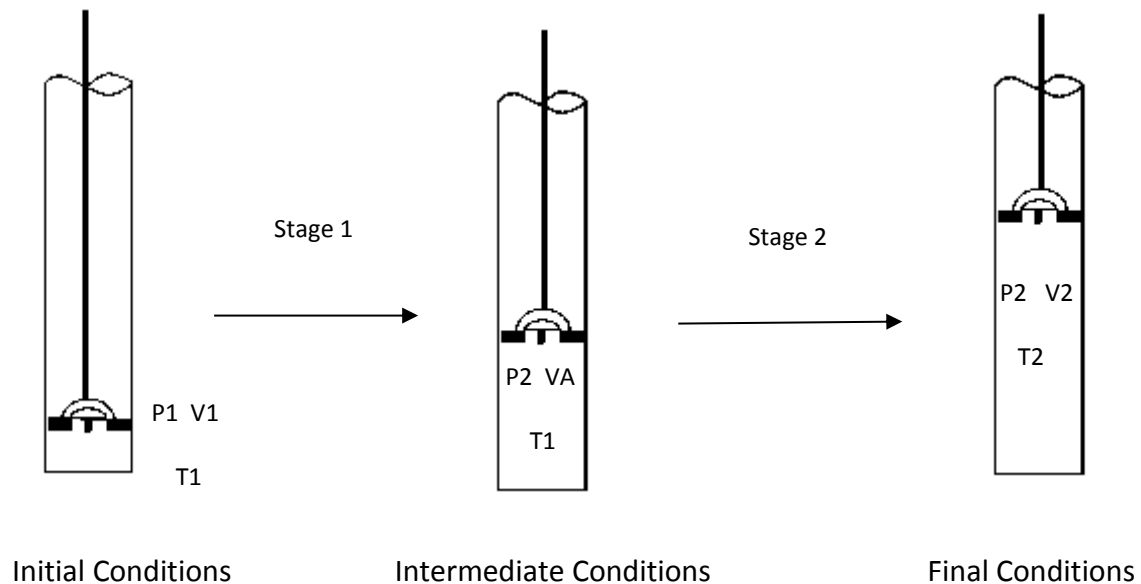
P_2 = Final Pressure (Pa)

But we can combine Boyle's and Charles's Laws to get the Gas equation.

Imagine changes according to both Laws in two stages...

Stage 1 where T is constant.

Stage 2 where P is constant.



So from Stage 1: $P_1V_1 = P_2V_A$

$$V_A = (P_1V_1)/P_2 \quad \text{Equation 1.}$$

And from Stage 2: $V_A/T_1 = V_2/T_2$

$$V_A = (V_2T_1)/T_2 \quad \text{Equation 2.}$$

Equate 1 and 2.

$$(P_1V_1)/P_2 = (V_2T_1)/T_2$$

OR

$$(P_1V_1)/T_1 = (P_2V_2)/T_2$$

Gas Law Examples

1. A given mass of gas occupies a volume of 200 litres at a temperature of 27°C. What would the volume of this mass of gas be at:

- a. 90°C,
- b. -30°C, under the same pressure conditions?

N.B: 1 Liter is 0.001m³

[0.242m³, 0.162m³]

2. A quantity of gas at a temperature of 15°C is heated at constant pressure until the volume is doubled. What will be the final temperature of the gas? [576K]

3. The compression ratio of an engine is 9 to 1, the pressure at the commencement of the compression stroke is 110kN/m² absolute and the temperature 115°C. Calculate the absolute pressure at the end of the compression stroke if the temperature has then risen to 180°C. [1155.9kN/m²]

4. At the commencement of the compression a cylinder contains 480cm³ of gas at a pressure of 96kN/m² absolute and temperature 100°C. At the end of the compression the volume is 80cm³ and the pressure 725kN/m² absolute. What will be the temperature at the end of compression? N.B 1cm³=1x10⁻⁶m³ [469.5K]

5. When the pressure in a car tyre was checked at a temperature of 12°C, the tyre gauge showed a reading of 1.75bar. What would be the tyre gauge reading when the temperature had increased to 38°C, assuming the volume of air in the tyre to be constant? (take atmospheric pressure to be 1.013 bar. 1bar=1x10⁵N/m²) [2.003bar]

6.

a) A compression ignition engine has a bore of 14cm and a stroke 12.2cm. The clearance volume is 128cm³. Calculate the compression ratio of the engine.

b) In the above engine, the air pressure at the commencement of the compression stroke is 100kN/m² and the temperature is 35°C. Calculate the temperature at the end of the compression stroke when the pressure is 4000kN/m². [15.7:1, 786.1K]

7. A compression ignition engine has a compression ratio of 14:1. The pressure of the air at the intake is 1.013bar, temperature 18°C and 2525cm³ per cylinder. If the pressure at the end of the compression is 38.5bar, calculate the temperature of the air at the end of compression. [789.98K]
8. A quantity of gas occupies a volume of 0.056m³ at and pressure of 14 bars and a temperature of 27°C. It is allowed to expand until the pressure has fallen to 8.4bars and the temperature to 15°C. What will the new volume be? If the temperature then increases to 38°C without a change in volume, what will be the new pressure? [0.0896 m³, 9.07bar]
9. A quantity of air in a cylinder is compressed to one fifth of its volume. The pressure is initially 105kN/m² and the temperature rises during compression from 15°C to 120°C. What is the pressure of the compressed air? If this air is allowed to expand at constant temperature to its original volume, what will be the new pressure? [716.4kN/m², 143.28kN/m²]
10. In the compression stroke a diesel engine, 3 litres of air at pressure 93kN/m² and a temperature of 45°C are compressed to a pressure of 2900kN/m² and a temperature of 690°C. The compression follows the combine's laws of Boyle and Charles. Calculate the volume of air, in millilitres at the end of the compression. [291.3ml]

Pistons and Rings

The piston is an area where material choice is critical. Smaller engine pistons are usually made from aluminium whereas larger slower rpm engines tend to have cast iron pistons. Pistons undergo three main types of force, a frictional force caused by the effect of the piston rings on the wall of the cylinder (this is usually minimized through design of the rings and lubrication), an inertial force that is a result of the mass of piston and connecting rod assembly and the force of the explosion of combustion on the crown of the piston hence forcing the assembly down. The inertial forces can be minimized by reducing the mass of the piston and connecting rod, since the connecting rod takes the brunt of the compressing inertial forces there is very little can be done to reduce its weight and any radical material removal is done at the cost of strength often resulting in catastrophic failure. Pistons however have much scope for reducing the weight of the assembly since the force of combustion is spread over the area of the piston crown and are not concentrated. There are two ways of reducing piston weight, firstly by reducing the amount of piston skirt and secondly by manufacturing it from lighter materials. The piston skirts function is largely to take the brunt of the inertial side loading and also as some cooling effect, transmitting the heat away from the piston crown, this material can be reduced but not done away with totally.

Material choice is the best weight saving option, aluminium being the lightest but not always suitable for larger engines. Modern composites technology is progressing rapidly and there is much speculation of the possibilities of carbon pistons in the internal combustion engine. But when applying this technology to piston design the following has to be considered:

- Low thermal expansion.
- High strength at high temperatures.
- Lower strength than cast iron.
- Lower modulus of elasticity.
- Deforms easily under pressure loading.

The first two criteria suggest that the carbon piston would significantly reduce the inefficiencies caused during engine warm up (as with the aluminium pistons) since there is little thermal expansion the piston will hardly change size during heating, also once up to temperature the carbons strength will be unaffected by the heat. Unfortunately the last three criteria count against the carbon piston, its low strength and modulus of elasticity mean that its inability to undergo repeated compression and extension however small would result in premature failure, and its tendency to deform under pressure means that it would be no good for diesel fuelled (compression ignition) vehicles.

(SAE SP-1071, 1995)

There are also weaknesses in the construction of the carbon piston:

1. The boss to under the crown blend.
2. The upper most face of the gudgeon pin hole.

These regions are areas of difficulty where fabrication is concerned and are also areas of stress concentration. If construction of these regions were of a poor quality then a carbon piston would fail when put under the conditions of an internal combustion engine. (SAE SP-1071, 1995)

With this in mind the continued use of aluminium pistons would be more appropriate until further perfection of the carbon equivalent is perfected.

Piston rings have three main roles:

1. To seal the combustion chamber from the rest of the crank casing.
2. To transfer heat away from the piston to the cylinder walls.
3. To control the flow of oil.

The piston system accounts for approximately 40% of the total frictional force of an engine and approximately 70% of the frictional force of the piston system is that of the piston rings. Since this is the case, engine builders are continually striving to reduce the friction caused by the piston rings by improving their design. There are two types of ring used in the piston assembly, a compression or scraper ring made from cast iron and an oil control ring made from mild steel. These rings can be prone to wear since they are always in contact with the cylinder walls, ring wear causes damage to the surface of the cylinder and the seal between the combustion chamber and the crank casing to be compromised, resulting in inefficient and less effective combustion and allowing oil to flow into the combustion chamber being burnt during the process increasing the toxicity of exhaust gases.

The process of nitriding can improve ring wear. Gas nitrided or silicon carbide rings feature a number of superior characteristics as well as high wear resistance, reduced attack on cylinder bore and reducing oil ring groove wear by ion nitriding.

Furthermore, partial nitriding with ion nitriding is considered to be effective to maintain strength and avoid deformation of the three thin components of the oil control ring, which are required for high performance engines.

There are usually two compression rings and one oil control ring used in engines, but recent research suggests that having only one of each ring produces less friction throughout the whole engine speed range and improve fuel consumption and acceleration. Also the use of a 'double tapered step joint'

in the scraper ring instead of the usual plain taper dramatically reduces oil consumption (see figure 1 below).(SAE SP-1071, 1995)

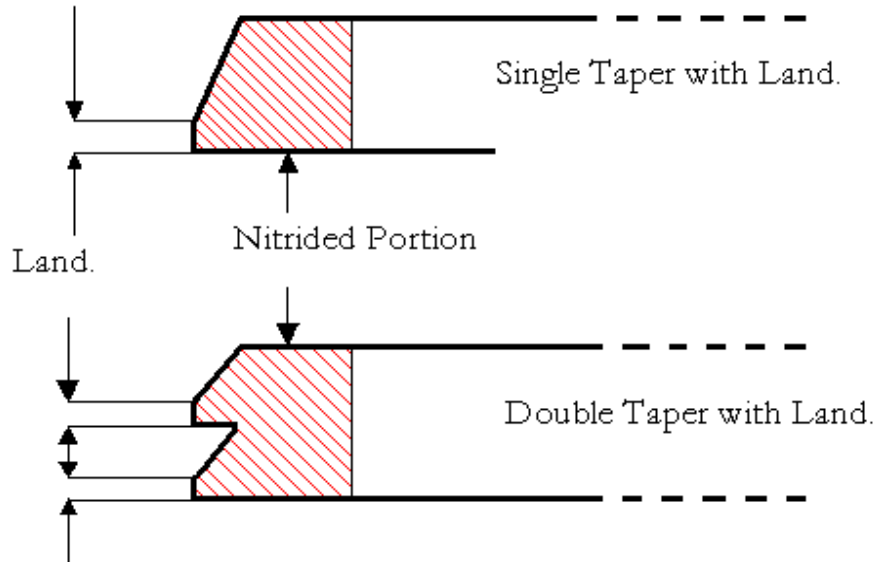


Figure 1: Piston rings

Cylinder liners are cooled directly by water flowing around them. This cooling could be greatly enhanced by adding fins, similar to those found on air-cooled engines. More effective cooling means that there is less coolant needed and that the engine warms up quicker from the more efficient flow. This effective warm-up results in the engine reaching optimum running temperature sooner and increased engine efficiency and reducing emissions which occur through the inefficient early combustion process. These fins are a great advantage to engine efficiency, but require complex casting which will often be offset by the price of the finished product.

Connecting Rods

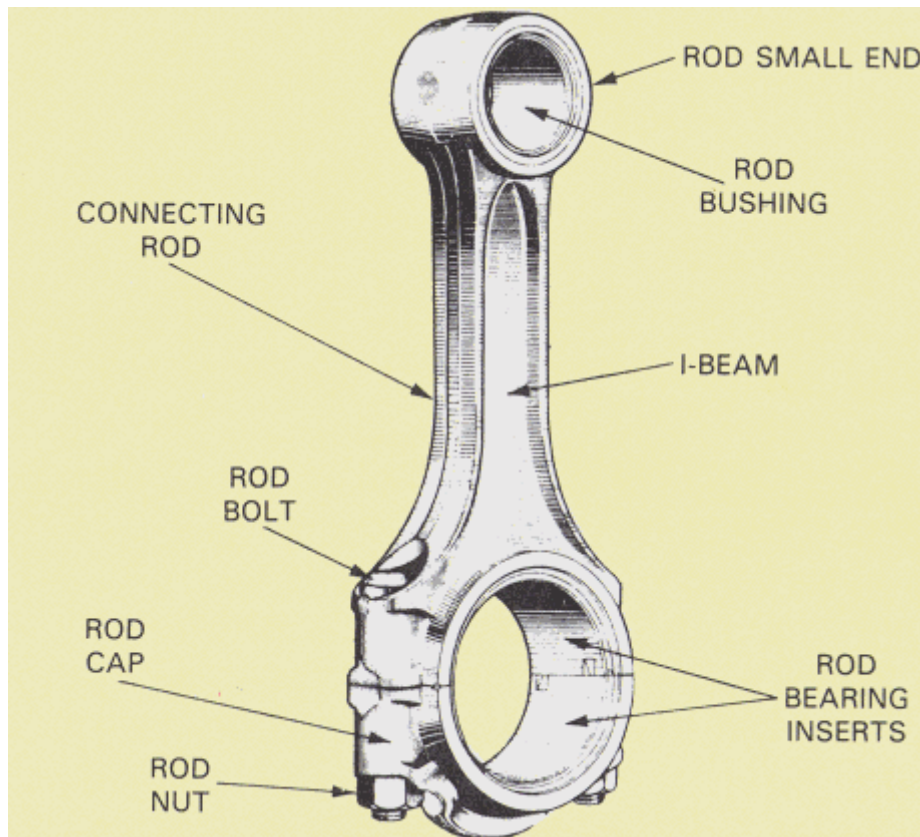


Figure 2: Connecting rod

Loading due to gas pressure from combustion has a buckling effect on the shank in the direction of the crankshaft axis. This tendency to buckle is supported by the gudgeon pin and the crankshaft, but because there is no support for buckling sideways the rod is made from 'I' section.

Big-End

Some connecting rods consist of piece, and have a solid-eye big end that can only be fitted to crankshafts made of many parts. Normally Automotive engines are fitted with two-piece connecting rods such as the rod featured in figure 2, which is split at the big end to fit on to a single piece crankshaft. This split wherever possible is at right angles to the length of the connecting rod (figure 2). In order to give as large as possible big end (for strength) it is sometimes necessary to split the big end obliquely. The nature of the way these big ends are split determines the forces that the bolts need to withstand. The connecting rods split at right angles to the rod undergo severe tension as the forces through the engine cycle try to pull the cap off the big end. The obliquely split big ends undergo

this tension but, also experience shear forces. In order to reduce these forces the mating faces are often serrated to inhibit any movement (figure 3).



Figure 3: Serrated mating faces

Small-End

The small end is fitted to the piston via a semi floating gudgeon pin. This is a heavy-duty piece of tube which is pushed through the small end of the connecting rod and the piston. It is called semi floating because the gudgeon pin is fitted within the connecting rod and cannot move once in place but is free to rotate within the piston. This close fitment of the gudgeon pin is known as an interference fitment, this is where the pin is actually too small to fit in the small end of the connecting rod. In order to fit the gudgeon pin the connecting rod is heated up so that it expands and the pin can be placed easily. Once cool the connecting rod will have contracted and gripped the pin securely.

Shank

As mentioned above the shank is prone to buckling when under the stress of combustion and in order to retain as much integrity of the rod but still keeping it light the 'H' section is used. This can be turned through 90° to an 'I' shape so that the rod can move through the oil in the lower parts of the engine better at high engine speeds.

Also shanks that taper dramatically channel stresses through the rod to the narrowest and hence potentially weakest point. Non tapered shanks spread potentially harmful stresses over a larger area dispersing the loading.

Polishing

All grinding/polishing must be done lengthwise to encourage any imperfections to follow the shape of the rod discouraging any cracks from propagating across the shank. As little material as possible should be removed from the cross sectional area as this is the narrowest part of the rod and needs

the material to maintain rigidity and strength. Removing any casting flash, nicks or imperfections on the outer side of the rod and round off all edges removes surface imperfections or sights where stress raisers could develop into cracks.

A draw back to rod polishing is the possible removal of any case hardening, nitriding or other surface treatments that would have to be put back into the part once all the other process have been completed.

What are connecting rods usually manufactured from?