INTERNAL COMBUSTIN ENGINE PERFORMANCE

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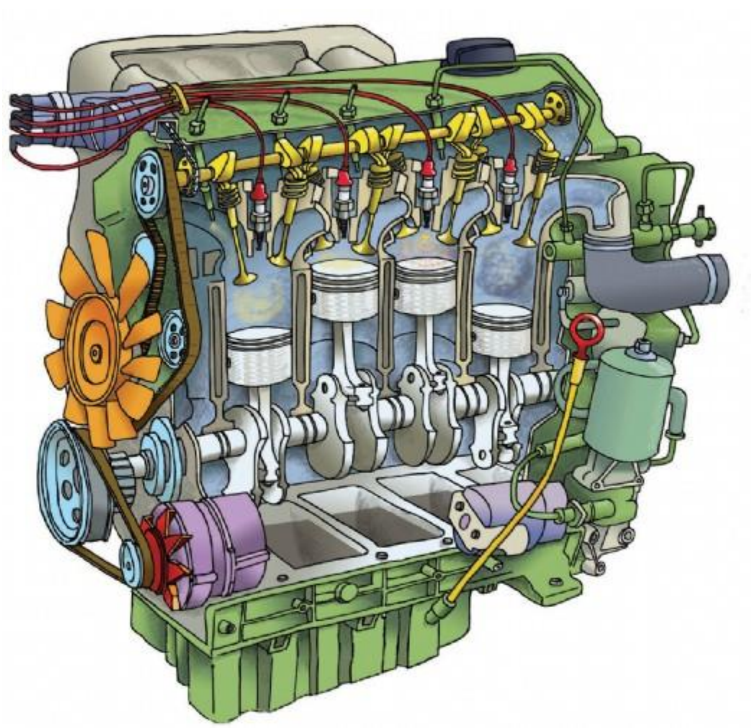

MARCH 22, 2018

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ABSTRACT

The purpose of this experiment was to compare the performance of the two most common forms of internal combustion engines: compression-ignition and spark-ignition. By comparing their performance parameters and accounting for errors, it was concluded that a CI engine was a better performer in most categories including greater torque, power, thermal and volumetric efficiencies. An analysis of the differences between the real and ideal cycles was also considered in the process.

The experimental setup consisted of 2 Volkswagen engines with specifications provided. By using additional instrumentation such as sensors, factors such as exhaust temperature and fuel flow rate were measured and recorded. The data was then logged onto a software and exported for data processing.

A comparison of efficiencies suggested that the CI was significantly more efficient than the SI engine. Looking at thermal efficiencies, at an arbitrary selected high speed of 1600 RPM, the diesel CI engine was around 58% whereas the SI engine was approximately 24%.

Given also that the higher compression ratio of the CI is analytically more efficient than an SI for most engine speeds, and based on the results for torque, power and AFR, this experiment clearly shows why the CI engine is more powerful, efficient and reliable than the SI engine.

1 Introduction

Reciprocating internal-combustion (IC) engines have numerous applications including road vehicles, railway locomotives, ships, marine craft, onsite electricity generation and more. The output power ranges from being below 1kW (lawn-mower engines) to more than 20MW (stationary engines). The aim of this lab was to determine and compare the principal performance characteristics of two 4-stroke engines – a compression-ignition (CI) and spark-ignition (SI), as a function of speed at constant throttle settings.

The difference between the compression-ignition and spark-ignition engine lies in the fuel types and ignition of the two engines. CI engines mostly use Diesel as the fuel, whereas SI engines commonly use Gasoline. Diesel is a heavier hydrocarbon with a higher ignition temperature. Therefore, the two engines, SI and CI, are approximated by two different ideal thermodynamic cycles: the Otto and Diesel cycle respectively.

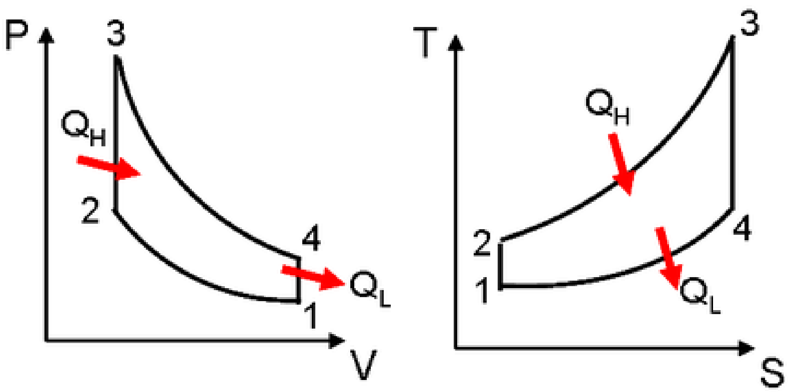

Figure 1: Pv and Ts diagrams of Otto cycle

<u>OTTO CYCLE</u>

1-2: ISENTROPIC COMPRESSION OF AIR

$$P_1V_1^{\gamma} = P_2V_2^{\gamma}$$

2-3: ISOCHORIC HEAT TRANSFER TO AIR

$$Q = C_v\Delta T$$

3-4: ISENTROPIC EXPANSION OF AIR

$$P_3V_3^{\gamma} = P_4V_4^{\gamma}$$

4-1: ISOCHORIC HEAT TRANSFER FROM AIR

$$Q = C_v\Delta T$$

<u>DIESEL CYCLE</u>

1-2: ISENTROPIC COMPRESSION OF AIR

$$P_1V_1^{\gamma} = P_2V_2^{\gamma}$$

2-3: *ISOBARIC HEAT TRANSFER TO AIR*

$$Q = C_P\Delta T$$

3-4: ISENTROPIC EXPANSION OF AIR

$$P_3V_3^{\gamma} = P_4V_4^{\gamma}$$

4-1: ISOCHORIC HEAT TRANSFER FROM AIR

$$Q = C_v\Delta T$$

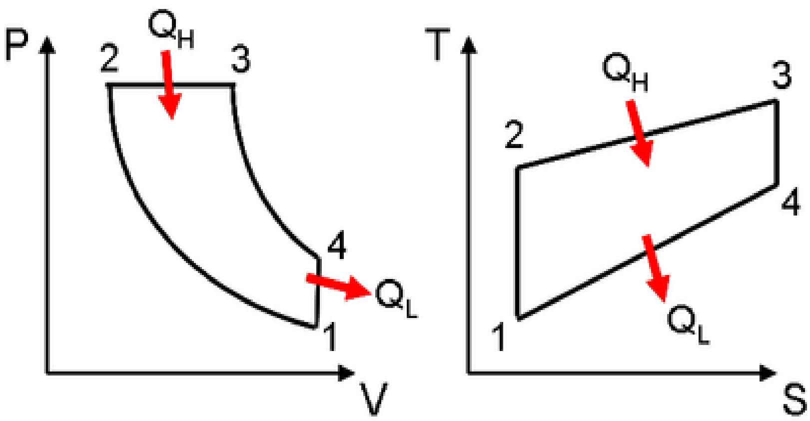

Figure 2: Pv and Ts diagrams of Diesel cycle

The difference in the processes occurs in step 2-3. A slower combustion of diesel causes temperature rise alongside volume change, resulting in an isobaric heat transfer rather than isochoric.

However, in reality the engine cycle differs in having rounded corners and a region of negative work (an anti-clockwise loop) required to overcome the pressure drop through the valves during the exhaust and induction strokes. Irreversibilities include friction and the combustion process, moreover the compression and power strokes are non-adiabatic. All these factors contribute to a lower thermal efficiency. Therefore, calculations performed using ideal cycle approximations often deviate from reality.

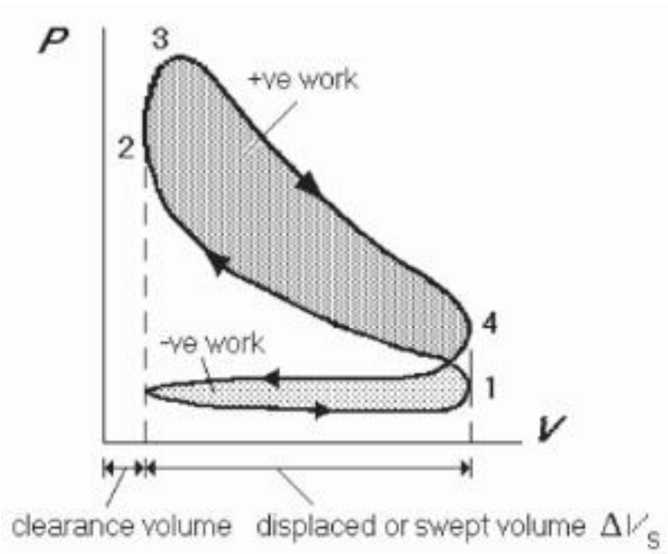

Figure 3: P-V diagram of a real Otto cycle, which has rounded corners and a negative work loop

Performance Parameters

Detailed analyses of engine performance are necessary to suitably decide on the best-suited application for each engine type. The performance parameters used to compare the engines in this experiment included the following:

- Thermal efficiency – Efficiency of a heat engine measured by the ratio of net work done by engine to the heat supplied to it (calorific value of fuel entering the engine)
- Brake mean effective pressure – net work per cycle per unit of displaced volume. 2 crank shaft revolutions are equivalent to one thermodynamic cycle for each displaced volume of the fuel/air mixture
- Volumetric efficiency - the ratio of mass of air induced per cycle to the maximum capacity of the cylinder
- Torque – A measure of the rotational force produced by the engine
- Brake power – net shaft power generated by the motor
- Air to fuel ratio (AFR) - the ratio between the volume of air and volume of fuel induced across each thermodynamic cycle.
- Cut-off ratio (Diesel cycle) – the ratio between volumes of process 2-3. This ratio describes the volume difference during the combustion process.

2 Experimental set-up and instrumentation

1) Engines

The two engines are an industrial CI engine and an automotive SI engine.

The CI engine is a Volkswagen 1.9 litre four-cylinder diesel engine, with indirect injection (the fuel being injected into a whirl chamber rather than directly into the cylinder) and a compression ratio r_v of 22.0

Fig 4 – CI engine, Volkswagen catalogue

The SI engine is a Volkswagen 1.2 litre three-cylinder petrol engine with a compression ratio r_v of 10.3. The petrol is delivered through a common rail system with three intake manifold-sited fuel injectors.

Fig 5 – SI engine, Volkswagen catalogue

The sensors attached to the engine test beds are placed to record several different properties. These properties are critical towards calculating some performance parameters, which are evaluated later. Table 1 below summarises information on both types of engines:

Table 1: CI and SI engine specifications

	CI engine	SI engine
Number of cylinders	4	3
Cylinder bore (mm)	79.50	76.5
Piston stroke (mm)	95.50	86.9
Cylinder capacity or total displaced volume (litre) Vd = $(\pi/4)$ x bore^2 x stroke x no of cyls	1.9	1.198
Compression ratio r_v	22.0	10.3
Fuel density ρ_f (kg/m^3) (nominal)	840	770
Fuel net calorific value CV_f (kJ/kg) (nominal)	42500	44000
Fuel stoichiometric air-to-fuel mass ratio AFRst (nominal)	14.5	14.6

2) Instrumentation

The quantities to be measured in this experiment are shaft speed, torque, air flow rate and fuel flow rate. These quantities were measured and recorded by external instruments, and then saved on a computer. The equipment and method used to measure these quantities are explained below:

- Shaft speed – A **tachometer** on each engine uses a magnetic sensor whose pulses are converted to rev/min.
- Torque – the **dynamometer** unit comprises of a rotor mounted on a shaft running in bearings that rotates within a casing. When supplied with DC, two field coils within

the casing create a magnetic field across the air gap at either side of the rotor. When this happens, eddy currents are induced creating a braking effect between the rotor and the casing. The **rotational torque** is measured by a **strain gauge** load cell incorporated between the casing and dynamometer bed plate.

- Air flow rate – the engines are fitted with a **laminar element flow meter**, where we can use laminar flow theory to derive the velocity profile in each small tube of the element as a function of pressure drop. From this, the mean velocity in each tube is found and therefore the total air flow rate can be found too. The equation for mass flow rate is: $\dot{m}_a = C_{lfm}\rho_a \Delta P$. A manometer measures the pressure drop.
- Fuel flow rate – the fuel flow rate is measured in cm^3min^{-1}. A **small turbine flowmeter** rotates with a speed proportional to the flowrate. Due to rotation, interruptions to a high intensity infrared beam produces a **stream of pulses**. The frequency of these pulses is converted to a flowrate.

3) Procedure and Control

The engines were run at one throttle setting and left idling till steady state conditions were obtained. The throttle was controlled using an analogue input, which changed the motor speed. The engine speed was varied only by changing the load/torque applied by the dynamometer. An issue faced was that the engine lagged throttle control by approximately 5 seconds and that the throttle did not correspond to linear variations in the speed. This led to slow, imprecise increments of engine speed which could be improved with a digital input and PID system.

The data for engine speed, fuel and air flowrates, barometric pressure and inlet air temperature was then recorded every 5 seconds for 60 seconds at each engine speed value using the logging software.

3. Data Processing

After recording the data on a spreadsheet, some essential calculations were performed to produce engine performance parameters. These parameters were then used to compare the performance of the two engines. The equations involved and the unit conversions are explained in table 2 below.

Table 2: Performance Parameters Data Processing

Performance Parameter	Formulae	Unit Definition
Brake Power	• $\omega = v \; x \; \frac{2\pi}{60}$ • $\dot{W}_b = T\omega$	v = engine speed in rpm T = torque in Nm ω = angular velocity in rads^{-1} $\dot{W}_b$= brake power in W
Thermal Efficiency	• $\dot{m}_f = f_f \; x \; \frac{1}{10^6} \; x \; \frac{1}{60} x \; \rho_f$ • $\eta_{th} = \frac{\dot{W}_b}{\dot{m}_f \; x \; CV_f \; x \; 1000}$	η_{th} = thermal efficiency in % $\dot{m}_f$ = fuel flow rate in kgs^{-1} CV_f = calorific value of fuel in kJkg^{-1} f_f = fuel flow in cm^3min^{-1} ρ_f = fuel density in kgm^{-3}

Volumetric Efficiency (equation is dependent on thermal cycle speed)	• $\dot{m}_a = C_{lfm} \, x \, \Delta P \, x \, \rho_a$ • $m_{a/c} = \frac{\dot{m}_a}{C}$ • $\eta_{vol} = \frac{m_{a/c}}{V_d \, x \, \rho_a} \, x \, 100\%$	$\dot{m}_a$ = air flow rate in kgs^{-1} $m_{a/c}$ = mass of air per cycle in kg C_{lfm} = flowmeter calibration constant ρ_a = ambient density of air in kgm^{-3} C = cycles per second s^{-1} V_d = volume displaced in m3
Air to Fuel Ratio	• AFR = $\frac{\dot{m}_a}{\dot{m}_f}$	
Cut-Off Ratio (assuming isentropic perfect gas and that γ = 1.4 (pure air mixture))	• $\frac{V_3}{V_2} = \frac{T_3}{T_2} = (\frac{T_4}{T_1})^{\frac{1}{\gamma}}$	T_4 = engine outlet temperature in K T_1 = engine inlet temperature in K γ = ratio of specific heats

It's useful to note that the engines in this equipment are not equipped to obtain indicator diagrams. Very-low speed engines allow the use of a purely mechanical indicator mechanism but not these engines. The difference between indicator power and brake power $\dot{W}_\iota - \dot{W}_b$ is called friction power, the power absorbed by friction in the engine's mechanism. Friction power is also a potentially useful parameter for performance.

The equations in table 2 were applied to the data sets of each individual engine speed increment. The diesel engine spreadsheet also requires a cut-off ratio calculation for each engine speed increment whereas the petrol engine does not.

4. Results and Discussion

From our resulting data sets for the petrol and diesel engine, comparisons must be made by observing trends between engine speed and each performance parameter. Firstly, the performance parameter values were calculated as time averaged values and stored in a table as below:

Table 3: Performance Parameters Data CI Diesel Engine

Speed (rpm)	Torque (Nm)	Brake Power(W)	Thermal Efficiency %	Volumetric Efficiency %	Air to fuel ratio	Cut-off ratio
1247.48	22.71	2966.739	573.117	96.323	1913.491	4.263
1327.69	30.36	4221.114	89.688	93.993	218.573	4.749
1508.19	59.10	9334.094	111.814	89.596	133.434	6.424
1573.52	75.77	12485.28	64.704	91.867	61.754	7.306
1710.52	88.24	15806.01	56.138	92.719	46.434	7.944
1804.57	100.68	19025.92	51.015	99.925	39.857	8.616

Table 4: Performance Parameters Data SI Petrol Engine

Speed (rpm)	Torque (Nm)	Brake Power(W)	Thermal Efficiency %	Volumetric Efficiency %	Air to fuel ratio
1211.20	9.06	1149.771	165.812	18.581	288.660
1280.61	11.28	1513.24	108.621	19.054	155.783
1480.83	20.91	3243.89	24.34	22.851	22.560
1590.58	30.65	5106.048	22.742	27.022	17.048

1642.61	46.96	8079.171	28.911	35.054	18.320

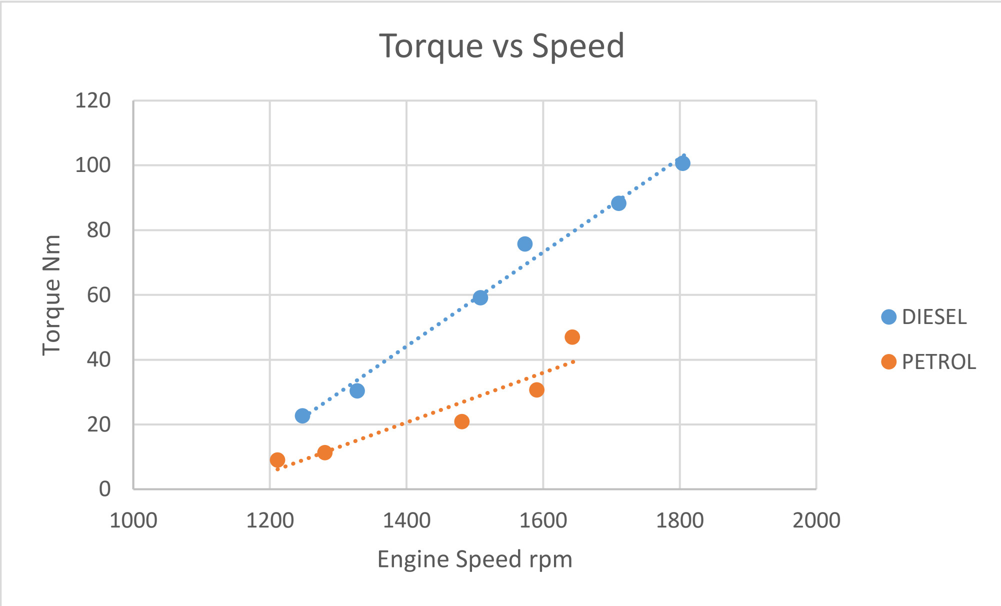

Figure 6: Trend between Torque and Engine Speed

The trend suggests that an increasing torque results in an increasing engine speed. The slope or rate of increase is greater for the Diesel CI engine, and the positive vertical offset is greater for the CI engine too. The Diesel engine can provide a higher torque for the same engine speeds, suggesting that this engine can bear heavier loads before reaching maximum speed. Therefore, according to the experimental data from figure 6 above, the Diesel engine performs much better than the Petrol engine because it runs at a higher compression ratio and has a greater number of cylinders.

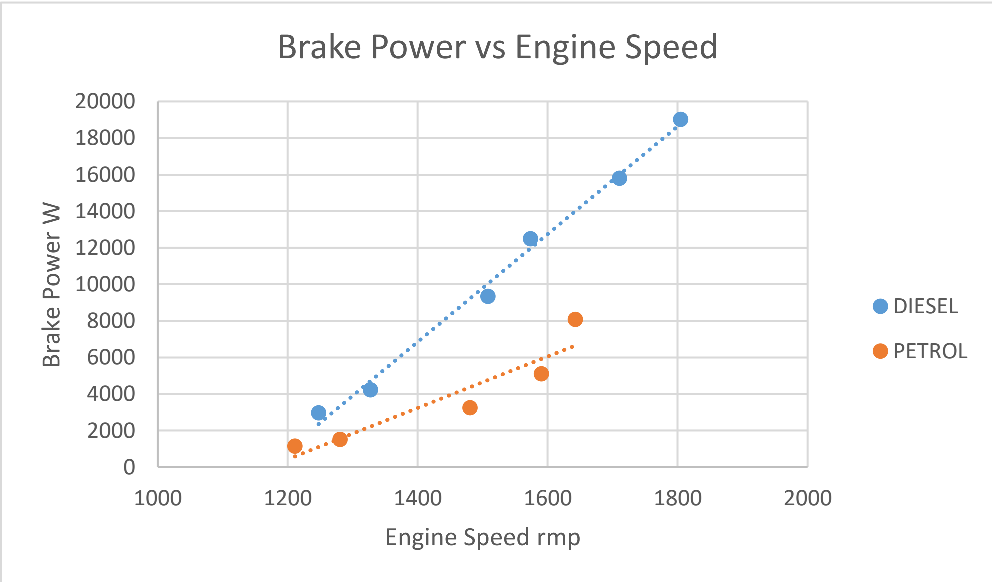

Figure 7: Trend between Brake Power and Engine Speed

Since break power is torque multiplied by angular velocity, I expected the graph trends from figure 6 and 7 to be proportional. As the trend before suggests, the Diesel engine provides a greater amount of break power for the same engine speed than the Petrol engine. This is a valuable characteristic because the CI engine is able to deliver greater power, enabling greater acceleration for the crankshaft. It also allows the engine to perform a greater capacity of work in a shorter duration of time. This results in a greater acceleration if the same mass is being driven.

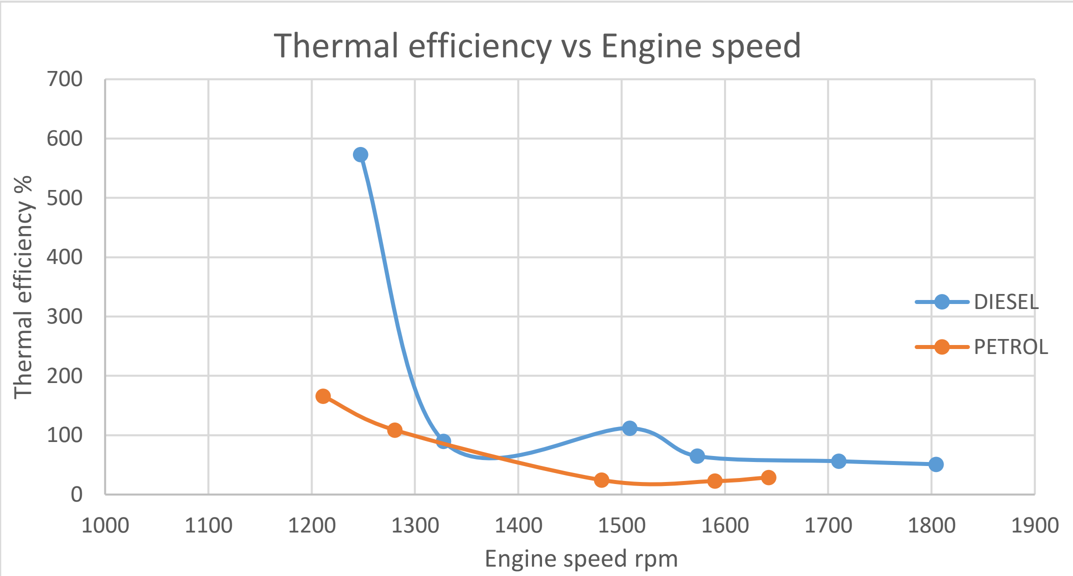

Figure 8: Trend between Thermal Efficiency and Engine Speed

The general trend seems to be a decreasing thermal efficiency with an increase in engine speed however the reliability of these curves is questionable. Firstly, the calculated efficiencies for lower engine speeds is significantly greater than 100%. The point at 1250 RPM is particularly noticeable as an anomaly around 600% efficiency. Secondly, at 1350 RPM the thermal efficiency of the Diesel falls below the Petrol engine's value of thermal efficiency. This is unexpected as CI engines have a much higher engine compression ratio, and therefore should expect a greater efficiency.

At very high speeds, the thermal efficiency of petrol rises whereas the thermal efficiency of diesel decreases. This is likely due to a balancing act between a fall in mechanical friction increasing efficiency and a faster rate of combustion decreasing efficiency. As the speed of the engine increases, the ratio of mechanical friction to break power falls, increasing efficieny. However, as the combustion rate increases, there is likely to be more incomplete combustion in the slower combusting Diesel engine, reducing efficiency. Petrol doesn't face this issue as it's combustion is almost instantaneous.

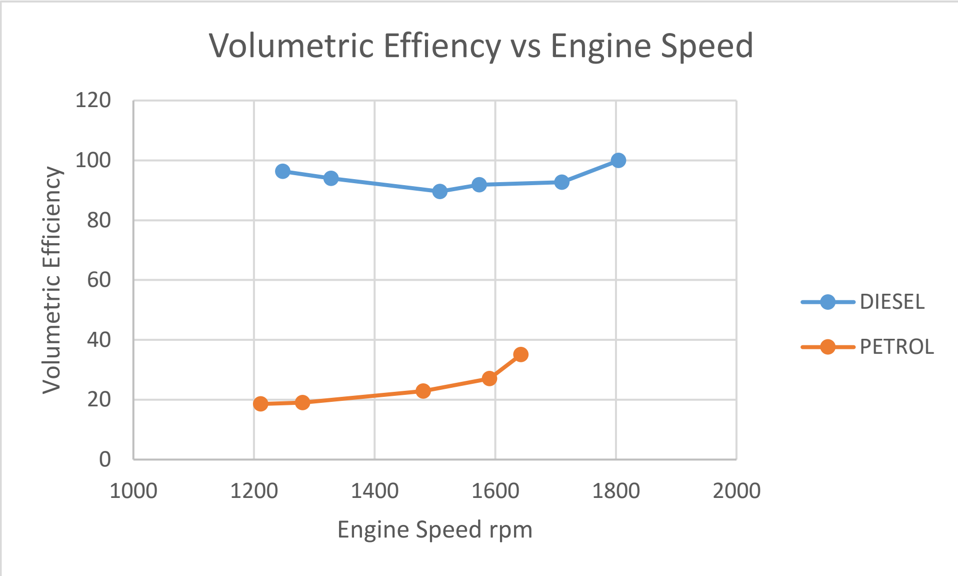

Figure 9: Trend between Volumetric Efficiency and Engine Speed

Both Diesel and Petrol engines demonstrate a minor increase in volumetric efficiency with engine speed. The Diesel engine is significantly more efficient than petrol, approximately 4.5 times. This difference can be explained due to the fact that SI engines use a throttle valve which constricts the inlet airflow at low engine speeds. This causes a large reduction in volumetric efficiency.

The general trend of an increasing efficiency is unexpected theoretically, however can be explained due to a large pressure difference during the intake stroke because of a faster moving piston head.

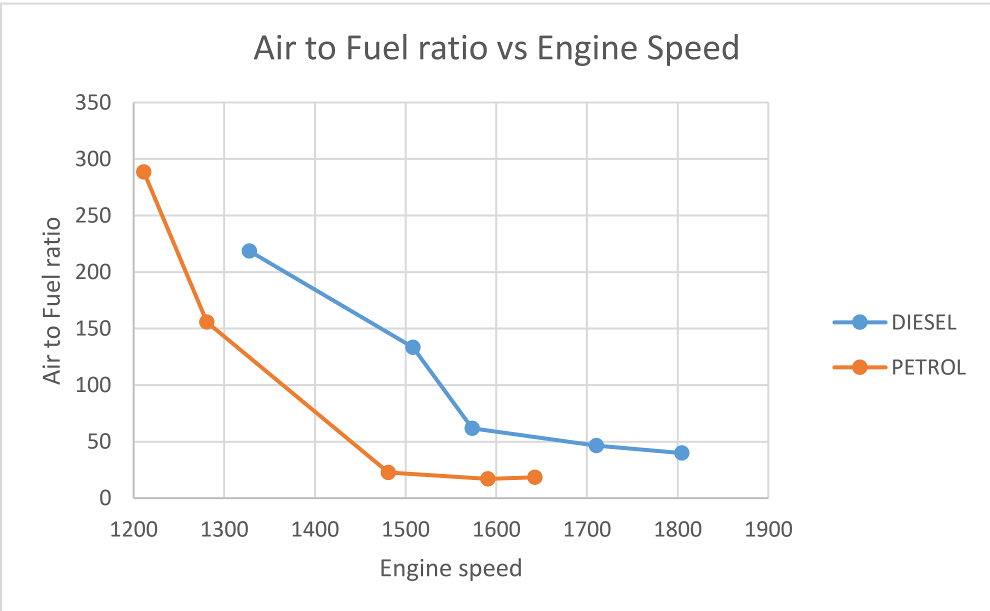

Figure 10: Trend between AFR and Engine Speed

There is a decrease in AFR for both engines with an increasing engine speed. According to the engine specifications (table 1) both engines have a stoichiometric ideal value around 14.5. At this value, the engines generate the maximum possible power and naturally approach this value as the brake power increases. Although all values are above 14.5, burning lean allows for a better thermodynamic efficiency and more complete fuel atomisation. The ease of burning well at lean mixtures means that Diesel engines with turbochargers will remain well within operating conditions at higher AFRs.

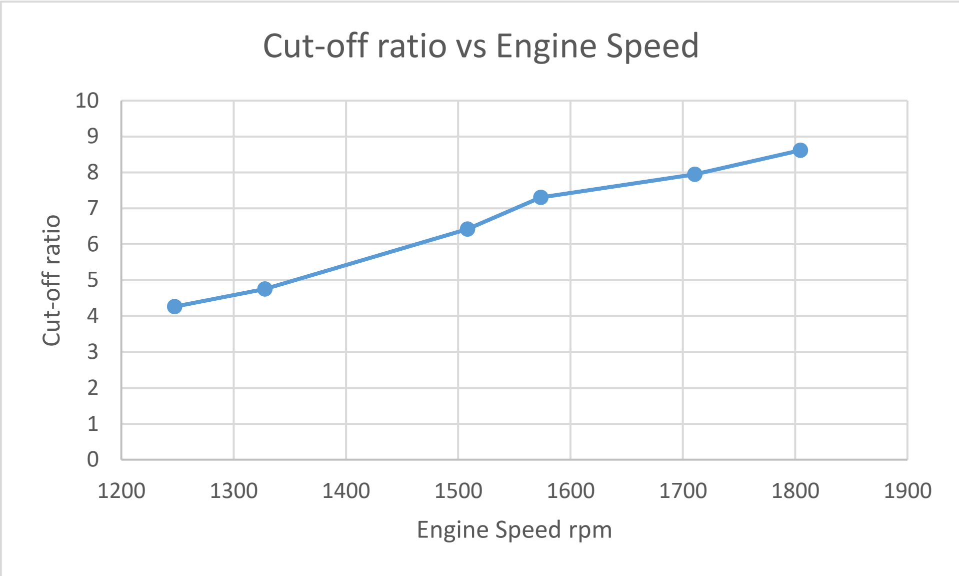

Figure 11: Trend between Cut-off ratio and Engine Speed

The cut-off ratio was calculated for the Diesel engines. Assuming that the mixture was a perfect gas, this gave a range of values between 4 and 8. The linear rising trend is expected, as the cylinder head moves faster at a higher engine speed, resulting in a larger volume displaced over the course of the combustion process.

5. CONCLUSION

The performance parameter analysis between the two engines suggests a clear performance advantage of the Diesel CI engine. The CI engine provides greater torque and power at the same engine speed, higher levels of thermodynamic and volumetric efficiencies whilst having a greater AFR ratio.

The errors in the experiment however did suggest several inaccuracies. Firstly, assumptions were made around the mixture being a perfect gas (pure air). This impacts the real cut-off values, because in reality the processes are no isentropic or isobaric. Moreover, the fuel flow rate measurements weren't recorded well at low values, likely causing an underestimation. Furthermore, the thermal efficiencies were greater than 100% for a few points which is inaccurate. Improvements to the experiment could be using more sensitive recording equipment and repeating the test results to account for the variation in the transient phases of recording results.

By analysing the cut-off ratios, the CI engine's values increase linearly with engine speed due to a faster piston head. Considering that CI engines also have high values of AFR above the stoichiometric, it was recommended that turbochargers be considered for the Diesel engines.

6. References

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