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Engine Performance Characteristics

Vehicle Technology ACHA4004

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1 Introduction

An engine dynamometer is a device mechanically connected to the engine and is used to measure the torque of the engine using resistive forces to generate load on the engine, allowing the performance impact of specific variables to be tested (Tibola *et al.*, 2016). There are many types of dynamometers, for example eddy current dynamometers, such as the one used in this report, and water brake dynamometers (Romandoni *et al.*, 2021). An Eddy current dynamometer applies a load against the engine using an electromagnetic field (José Rosa Nunes and Miguel Ribeiro Proença Brojo, 2020).

1.1 Aims & Objectives

This report aims to use data gathered from an engine by running tests on a dynamometer to gain a better understanding of the performance of the Honda 1.6 litre VTEC engine through analysis of Power, Torque, SFC, Thermal Efficiency, and Cylinder Pressures.

2 Methodology

A Honda 1.6 litre 4-cylinder VTEC engine was used for the testing, with the throttle controlled by a computer through a throttle actuator. The engine transmitted power to the dynamometer via a driveshaft connected to the crankshaft. The engine used had a 71.9 mm bore diameter d , 82 mm stroke L , and was fuelled with 95 RON petrol.

The tests were conducted using a Schenck WS230 dynamometer. The dynamometer was an Eddy current dynamometer with a maximum load rating of 230kW and a 10,000 RPM limit.

The fuel flow metre to measure the volume of fuel used over a 45 second period for the specific fuel consumption (SFC) test.

The tests were carried out in an enclosed, temperature-controlled environment to reduce variation in air temperature and pressure, and exhaust fumes were extracted. At the time of testing, the ambient pressure and temperature in the test location were 1024 mBar and 24 °C.

The sweep test method was used to gather data on the RPM force exerted, the fuel-mass consumed, and the cylinder pressure combined with respect to crank angle. The tests were run from 800 RPM to 6500 RPM. Data from the tests was recorded and exported into an Excel spreadsheet to be used later in MATLAB for calculations.

3 Results & Analysis

3.1 Torque & Power against RPM

The Torque T was calculated in MATLAB using the following equation:

$$T = W * L$$

W was the force recorded by the dynamometer during the test, and L was the radius of the dynamometer, which was 0.8m.

The Power P was calculated for every RPM from T using the equation:

$$P = \frac{2\pi}{60} * RPM * T$$

And was then used to find the value for the Corrected Power P_s using a value of 1.01325 bar and 293 Kelvin for Standard Pressure and Standard Temperature respectively:

$$P_s = P * \left(\frac{\text{Standard Pressure}}{\text{Ambient Pressure}} \right) * \left(\sqrt{\frac{\text{Ambient Temperature}}{\text{Standard Temperature}}} \right)$$

A graph was then created with the T and P_s plotted against the RPM at full load (Figure 1).

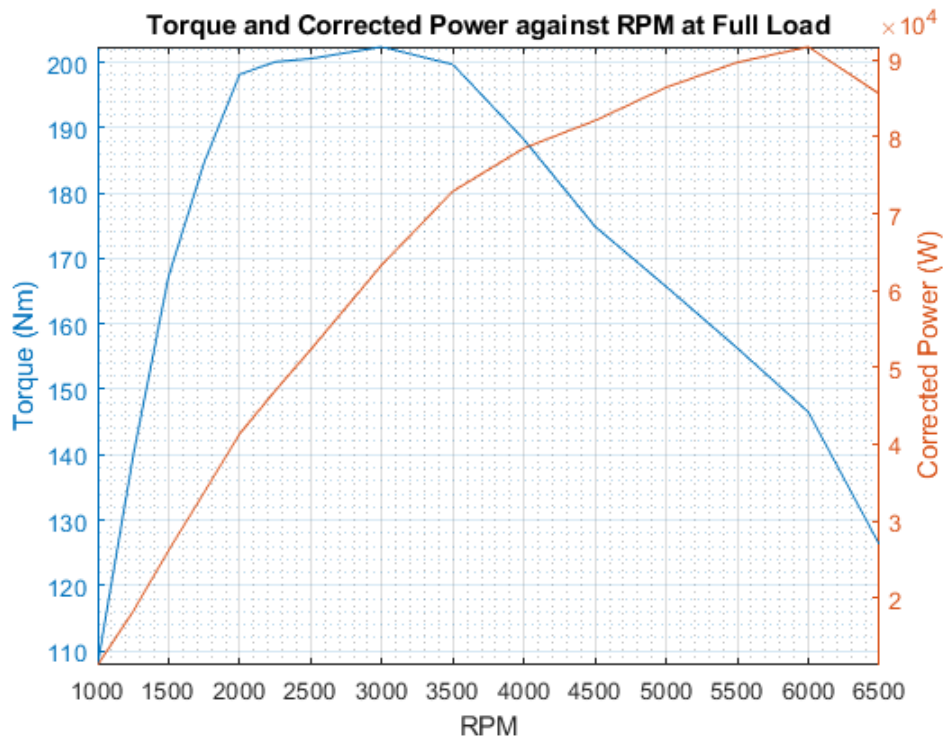


Figure 1 - Torque and Corrected Power against Engine RPM at Full Load (MATLAB, 2021)

Shown in Figure 1, T drops off after reaching the peak of 202 Nm at 3000 RPM and the maximum P_s of 92 kW was observed at 6000 RPM. P_s increases despite the reduction in T as there are more cycles applying T .

When the engine RPM passes 3500 RPM it declines steadily. This is due to the volumetric efficiency decreasing as the valves have less time to expel exhaust and intake air,

3.2 BMEP & Thermal Efficiency at Full Load

The area A of the bore was calculated to be 0.0041 m² through the equation

$$A = \pi * r^2 = \pi * \left(\frac{d}{2}\right)^2 = \frac{\pi}{4} * d^2.$$

This allowed for the BMEP to be calculated using the equation

$$BMEP = \frac{60 * 2 * P_s}{L * A * RPM * n}$$

where n is the number of cylinders of the engine, in this case 4.

Following this, the SFC was calculated, shown in section 0, and the value for Thermal Efficiency η_T was determined from this using the equation

$$\eta_T = \frac{3600 * 1000}{SFC * CV}$$

where CV is the calorific value of the fuel used, found to be approximately 44 MJ/kg (Department for Business, 2021).

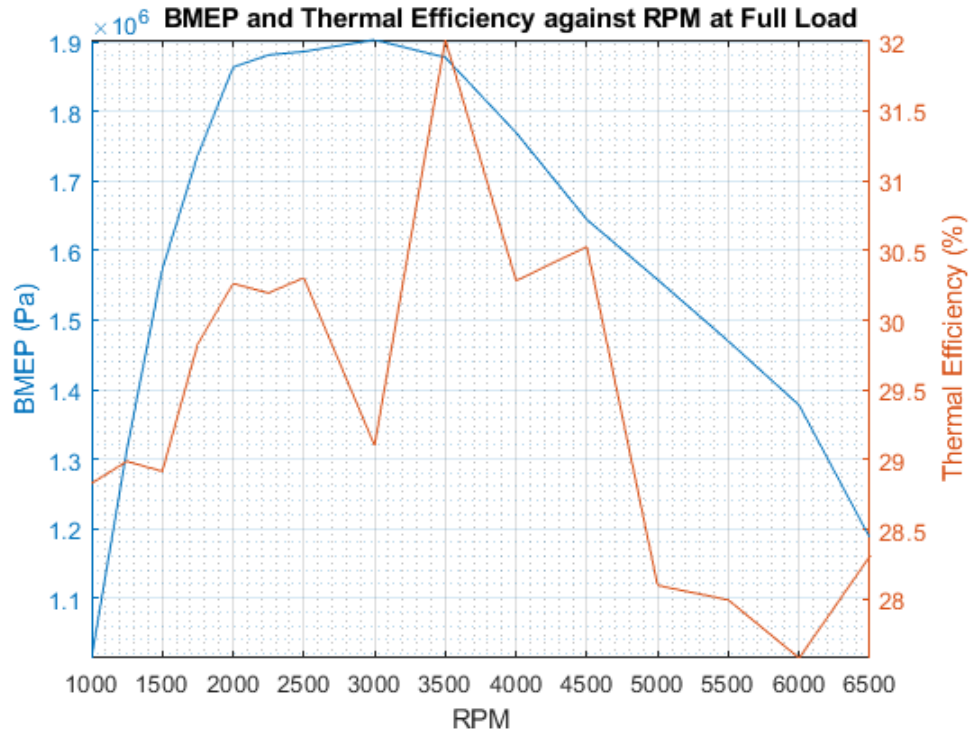


Figure 2 - BMEP and Thermal Efficiency against RPM at Full Load (MATLAB, 2021)

A peak BMEP of 1900 kPa corresponds well with the value of 1800 to 2400 kPa given by Heywood (2018) for the peak BMEP of a turbocharged SI engine.

3.3 SFC Heat Map

The SFC test recorded the volume of fuel used, V , over a 45 second period, t , and calculated the SFC through the equation

$$SFC = \frac{V * SG * 3600}{t * 1000 * P_s}$$

where SG is the specific gravity of the fuel used, the SG of petrol is 0.739 at 15.6°C (www.engineeringtoolbox.com, 2003).

A heat graph with Load against RPM was created, with the colours of the cells representing the SFC value.

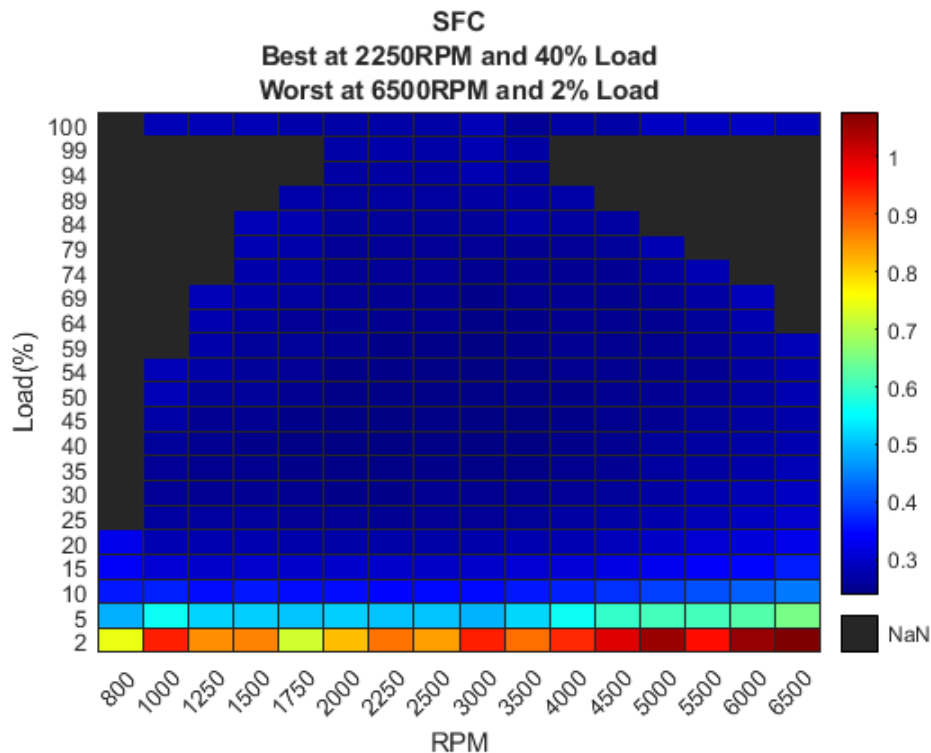


Figure 3 - Heat Map of SFC plotted across a range of Engine Load and RPM (MATLAB, 2021),

Figure 3 shows that the SFC is best at 2250 RPM at 40% load. This is likely because the lower engine speed means it suffers from less frictional losses, but not low enough to be overly affected by pumping losses.

At low RPMs, SFC is higher due to the throttle restricting the intake and increasing pumping losses (Alkidas, 2007).

At high RPMs, SFC is higher due to the reduction in volumetric efficiency mentioned in 3.1.

At low loads, the SFC is extremely high because there is hardly any power being produced but the engine is still using fuel to overcome the previously mentioned losses.

In a real-world scenario, it's expected that the SFC would be higher than shown in Figure 3 as there would be increased losses from the drivetrain (Eckert *et al.*, 2022).

3.4 Cylinder Pressures across RPMs and Crank Angles

The cylinder pressure was graphed against crank angle and RPM in Figure 4, this shows the peak pressure of 86 bar at 3500 RPM. The power stroke was determined to be taking place between 0 and 180 degrees of crank angle. This was identified by the peak pressure of 51 bar, seen better in Figure 5, before the crank angle of 0 degrees being the maximum pressure of the compression stroke from compressing the air into the smallest cylinder volume at TDC.

This is followed by the cylinder volume increasing, reducing pressure prior to the air-fuel mixture being ignited, creating an even higher peak pressure of approximately 85 bar, reducing as the piston moves towards BDC and expands the volume of the cylinder. At 180 degrees a pressure drop can be observed, this is likely due to the exhaust valve opening, releasing most of the exhaust gases out of the cylinder through the exhaust port.

The other downward stroke, being intake, was presumed to be exactly 360 degrees out of phase of the combustion stroke, suggested by the minimal change in pressure between the crank angle of -360 and -180 degrees.

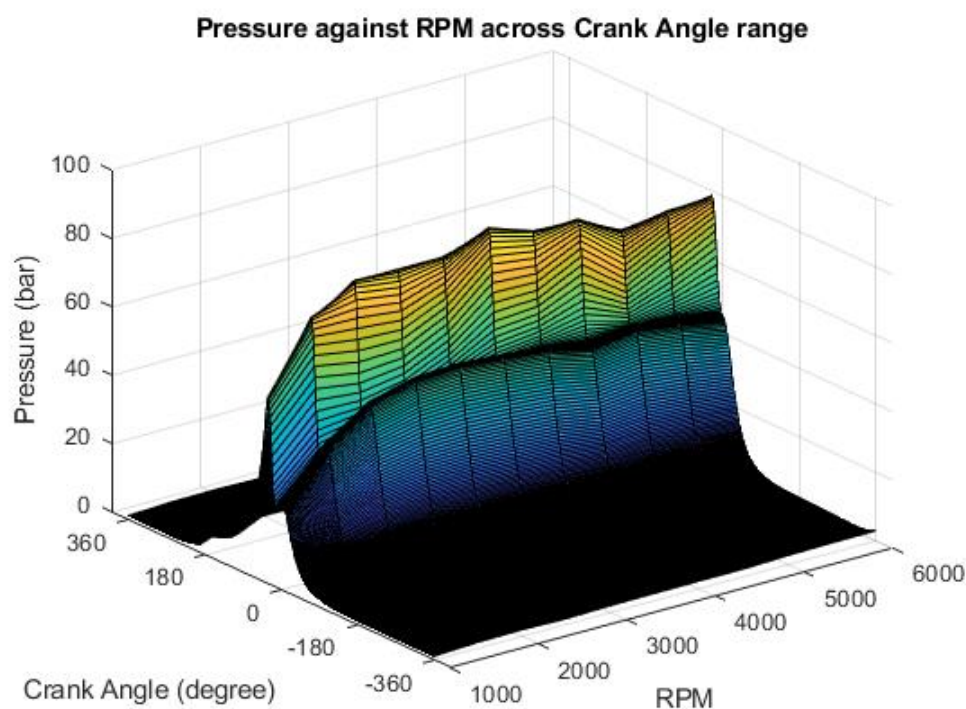


Figure 4 - Surface Chart of the Pressure Against RPM across the Crank Angle range (MATLAB, 2021)

The decrease in cylinder pressure on the compression stroke after 3500 RPM, seen in Figure 4, suggests that at higher RPM the engine is incapable of supplying as much air per cycle due to the shorter period the intake valve is open.

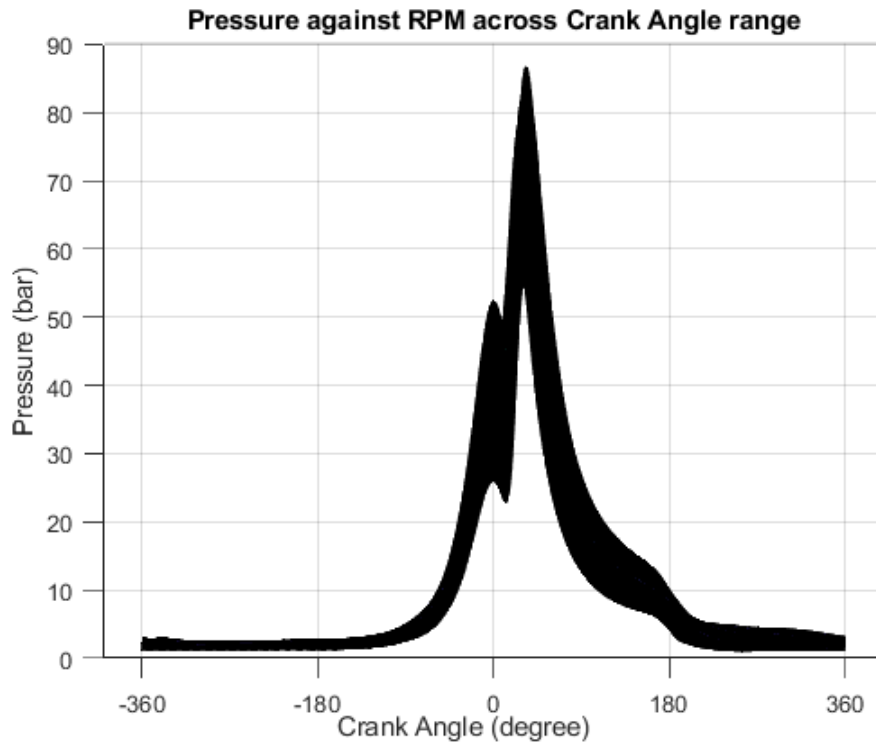


Figure 5 - A side view of the Surface Plot of Pressure against RPM across Crank Angle Range (MATLAB, 2021)

The power stroke was identified as being between 0 and 180 degrees of crank angle, shown by the peak pressure of 51 bar in between -180 and 0 degrees being the maximum pressure of the compression stroke followed by the cylinder volume increasing, reducing pressure prior to the air-fuel mixture being ignited, creating an even higher peak pressure of approximately 85 bar, reducing as the piston moves towards BDC and expands the volume of the cylinder. At 180 degrees a significant drop in pressure can be observed. This shows the exhaust stroke beginning, as the reduction in pressure is from the exhaust valve opening and releasing exhaust gases. The intake stroke is between -360 and -180 degrees seen by the minimal change in pressure as the piston moves from TDC to BDC and the space left is filled with air, from approximately 1.2 bar at 1000 RPM up to 2.7 bar at 3000 RPM due to the turbocharger.

4 Conclusion

The peak T occurred at 3000 rpm at full load with T reduced at low RPMs and less so at high RPMs, stemming from pumping losses and reduced volumetric efficiency respectively. Pumping losses were also responsible for increased SFC at low RPM, and volumetric efficiency was responsible for increased SFC at high RPM with the worst SFC at 6500 rpm and 2% load. SFC was particularly high when there was no load due to the engine not producing much power compared to the fuel used to produce that power. The peak P_s of 92 kW occurred at 6000 rpm despite lower T due to an increased number of cycles per unit time.

5 References

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