AN INVESTIGATION INTO THE DEVELOPMENT OF AN ELECTRICALLY ASSISTED TURBOCHARGER WITH ENERGY RECOVERY.

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Statement of Publications Arising from this Thesis

The following publications have arisen from my research detailed in this thesis:

"An electrically assisted turbocharger system for Formula Student"

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Contribution of the candidate: I performed all the research and wrote the entire manuscript with minor input and editorial changes from my co-author. It is indicated where text from this publication has been reproduced in this thesis.

Abstract

An experimental electrically boosted turbocharger system has been developed for a small motorcycle engine intended for use in Formula Student competitions. The system was developed and tested to produce elevated levels of torque at low engine speeds through the use of computer simulations validated by dynamometer testing.

This research took a standard inline two-cylinder engine with a displacement of 471cc and improved its torque output throughout the speed range. The engine was considered as originally designed, in a naturally aspirated state, then modified to be turbocharged, both the NA and the turbocharged engine being simulated and tested. There was less than 5% error between the simulation and test, therefore the simulation was considered suitable as a basis for further development and the turbocharger was modified to accommodate for a 9kW electric motor.

Results from the simulation show that there is a 25% increase in torque at engine idle for the electrically boosted turbo system. Furthermore, the electro-turbocharger engine starts to produce boost pressure at 5500rpm, with the system producing 0.19 bar of gauge boost pressure at engine idle. The same amount of pressure is not produced by the standard turbocharged engine until 6800rpm. The electrically boosted turbocharger system was used to overcome design issues such as turbo-lag and torque levelling. The optimised design demonstrates significant improvements in engine torque and driveability.

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Dedications and Acknowledgements

"In the name of ALLAH, the most Beneficial and Merciful." This project would not have been reachable if not for God himself bestowing me with knowledge.

Firstly, I would like to acknowledge my Mother for her dedication, from the times she would take me to tuition to aid me with my Dyslexia. Without her help I wouldn't be in the position I am today.

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Finally, I would like to acknowledge Manchester Metropolitan University's Formula Student team who lent me an engine for my project.

Introduction

Formula Student is an event which takes place every year at Silverstone. Teams usually use large capacity engines up to 710cc. These engines were chosen based on their characteristics of power and torque. The rules dictate that the vehicles must have a 20mm throttle restriction when running on petrol and a 19mm restrictor when running on E85, which limits their power, torque and efficiency. 710cc is the largest capacity allowed under Formula Student rules however, downsizing the engine can give a better performance through weight reduction but without the torque required.

Though forced induction is a great idea to make more power and increase efficiency, there are major setbacks with using a turbocharger on a downsized engine such as the Honda CB500x which is a 471cc two-cylinder. The correct turbo must be sized to the engine however, the main problem, the turbo-lag. Delay in torque response, known as turbo-lag, occurs when the driver presses the throttle after a period of sustained braking. An electric assisted turbocharger could overcome this issue.

Electric turbocharger compounds also known as ETC use electric motors built into the turbocharger housing, which overcome turbo-lag by accelerating the compressor wheel up to speed and regenerating electricity when excess turbine power is available. This technology is new and recently introduced into the world of Formula One. Several turbocharger manufacturers are developing these new turbo-systems but have not yet published enough detail about them (Latif & Allport, 2018).

Analysis of the Formula Student tracks show that the tracks are very slow and tightly cornered. This means a powertrain, which can produce large amounts of torque at low engine speeds, would be ideal to maximise acceleration through corners.

The main problem remains lost energy. Internal combustion engines are very inefficient. Around 30-40% of the fuel burned is transmitted to the driven wheel as usable energy; The other 60-70% is lost as heat energy, exhaust gases and noise. This means that if a car produces 100 kW of power on a dynamometer, around 200 kW is waste energy. Could this lost power be harnessed and used?

The answer is yes. Turbochargers harness the waste energy from the exhaust gases to compress atmospheric air and push it into the engine increasing the engine's volumetric efficiency, however energy is still lost. The 21st century answer to maximise engine efficiency and power is to capture waste gas energy through a turbogenerator and translate it into electrical energy.

This report looks at an inline two-cylinder engine which will be turbocharged and modified to accommodate an ETC system. The electric motor will also maximise turbocharger, boost pressure at low engine speed and when the engine does not produce enough exhaust gasses to accelerate the turbocharger, the turbocharger rotor will be accelerated using electrical power.

Research problem

In the past, racing teams in the Group B rally started to produce high horsepower rated engines using turbochargers. They saw problems when the vehicles had to slow down for corners as the turbochargers used were large in comparison to the engine displacement. After corners, the turbo-lag caused problems as torque demand was not available reducing the initial acceleration of the vehicle. To counter this problem combined charging was developed; this is when turbochargers feed one another or use superchargers to work hand and hand with one another.

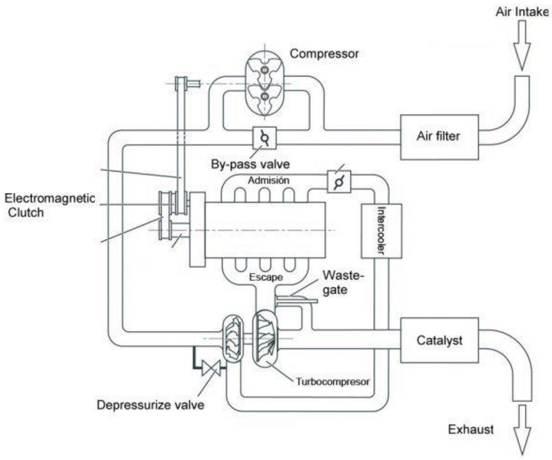


Figure 1 Combined Charging supercharger and turbocharger (Moderator, 2017)

Figure 1 shows one combined forced induction setup. The system uses a supercharger which is driven from the engines crankshaft to produce the initial boosted pressure. This boosted air travels into the compressor of the turbocharger. The boost pressure is then multiplied according to the turbocharger's compression ratio; it is then passed through a cooling system and into the engine.
There are several benefits of this charger setup. The first being the supercharger working alongside the turbocharger. As superchargers produce boosted air pressure throughout the engine's speed range, they produce boost pressure much lower in the engine's power band range whereas a turbocharger cannot. This means the supercharger helps to overcome the lag of the turbocharger.

Another benefit of combined charging is the turbocharger can be oversized for top end engine’s power band as the supercharger overcomes the turbo-lag. The powertrain can produce high engine speed power as well as torque at the lower engine speed range via the supercharger.

However, there are several drawbacks with combined setups using superchargers. Firstly, superchargers take a third of the mechanical power produced by the crank and give back 66% of the original 100% made by the naturally aspirated engine. This means an engine producing 100 kW with a supercharger it will now make 166 kW. This is generally the case with most superchargers although it can change using different crank pullies and setups. Nonetheless, it reduces its mechanical power.

The second is heat. When air is compressed it becomes hot due to thermodynamics. When air is compressed into boost pressure, the air is compressed at a high speed therefore the adiabatic process occurs. Therefore, the charged air becomes less dense after the supercharger. This charge of boost pressure then must pass through the turbocharger which multiplies the charged boost pressure, as well as increasing the temperature. The cooling of the boost temperature is vital because as air temperature increases so does the density. Consequently, as the density increases the amount of oxygen with in the pressurised air decreases (AET-turbos, 2018).

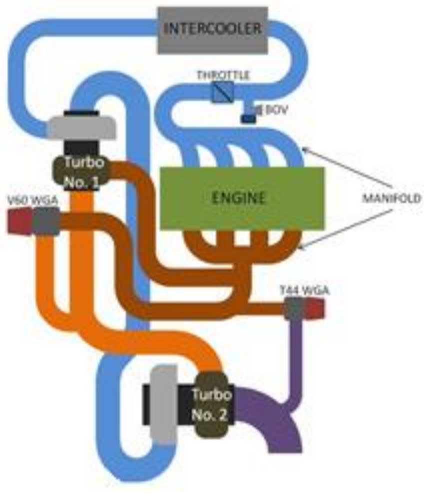


Figure 2 Combined Twin-Turbocharger system diagram (Gala, 2014)

Figure 2 shows another combined setup. This system uses two individual turbochargers which work with one another. Turbo one is smaller in comparison to turbo two. The second turbo draws in ambient air, passing it through turbo one, which multiplies the pressure before the pressure is absorbed by the engine’s cylinders.

The exhaust is routed through turbo one which then feeds from the turbine out into the turbine in on turbocharger two. Therefore, the exhaust gasses are being used to their maximum efficiency as they are being used twice.
This combined charger system uses the smaller turbocharger to reduce turbo-lag. However, if the compressor is not large enough, it will choke. On the other hand, if the turbine is not large enough it will increase the backpressure in the exhaust manifold if large amounts of boost is being fed into the cylinders.
This system has similar pros and cons to combined charging with a supercharger. Furthermore, the main problem with both these systems in order for them to overcome turbo-lag is the mass of the components. As Newton's second law states $F = ma$ if a (acceleration) is the constant or restricted then decreasing the m (mass) would increase the F (force). Therefore, to achieve the largest force when racing applications, mass must be reduced.
Formula One is the largest racing series in the world; engineers work their hardest to reduce the vehicle's mass and improve their vehicles performance as well as efficiency. In 2015 Formula One engines downsized to 1.6L in displacement. Rules demanded turbochargers to be used to increase the vehicles power and torque efficiency.
As Formula One turbochargers rotate at 100 krpm which translates to 1500 times per second, the pressure and temperatures in the turbine is used and converted into electrical energy. This is done by the user of a Motor Generator Unit – Heat or better known as the MGU-H.

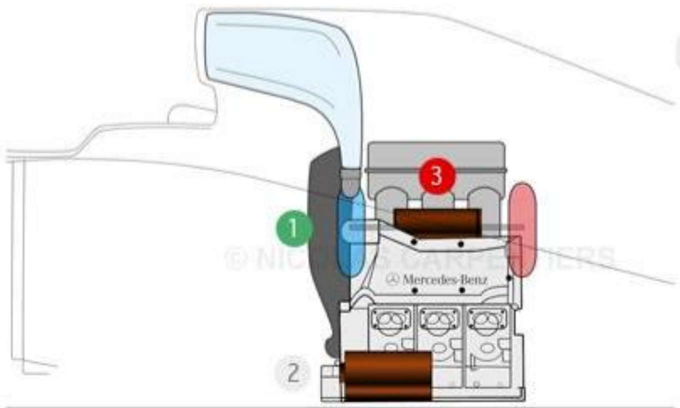


Figure 3 Diagram of Mercedes 2015 powertrain (Madier, 2016)

Figure 3 shows the engine and the turbocharger used by Mercedes in their 2015 competition vehicle. The MGU-H is based within the turbocharger bearing housings. The system generates electricity from the rotational velocity of the turbocharger and the heat energy within the turbine this electricity is either stored or deployed to prevent the turbocharger dropping off boost pressure.
The MGU-H acts as a motor and a generator reducing the overall mass of the powertrain without additional support from either a supercharger or a smaller turbocharger. Other

benefits are also achieved, such as the MGU-H being used as a wastegate as well as compensating for the turbo-lag.

The MGU-H regulates the turbocharger speed like a wastegate by braking the turbine. The braking limits the turbocharger speed, instead of bypassing the exhaust gasses into the inlet of the turbine housing like a generic wastegate does.

The MGU-H acts like an ETC system when the vehicle is braking into corners and the engine speed drops. The lack of exhaust gasses which are required to accelerate the turbocharger up to its efficiency regions to generate boosted air pressure is not there. The MGU-H compensates for the lack of the engine's exhaust gas at low engine speeds and accelerates the turbocharger using an electrical input (Madier, 2016).

Moreover, in Formula Student the acceleration cannot be improved due to the inlet being restricted, consequently the mass must be reduced. This can be done by using a downsized engine however, this reduces power, torque and efficiency. Therefore, turbocharging the downsized engine can match the power, torque and efficiency of the larger displacement engine but introduces a new problem of turbo-lag. Therefore, adding an ETC to the downsized powertrain can reduce the lag whilst keeping the overall powertrain's mass down.

What we aim for

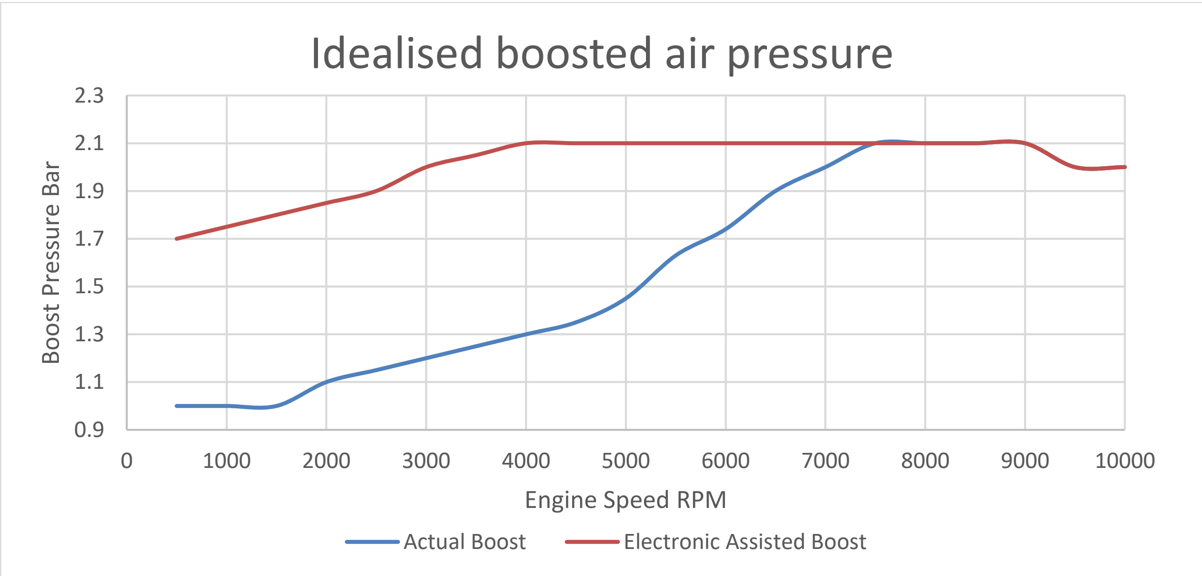


Figure 4 Actual boost pressure against electronic assisted boost pressure aim

Figure 4 plots an idealised behaviour in which the project aims to fulfil. Actual boosted air pressure (Blue), is the response that turbocharged engines usually have. The turbocharger in this case starts to produce boost pressure at around 4000 rpm. The electronically assisted turbocharger (Orange) is the response this report looks to achieve. When electronically assisting the rotational velocity of the turbocharger the boost pressure at engine idle can be instant therefore the turbocharger itself will not rely upon the engines ability to produce exhaust gasses to rotate the turbine to make the boost pressure.

Formula Student rules

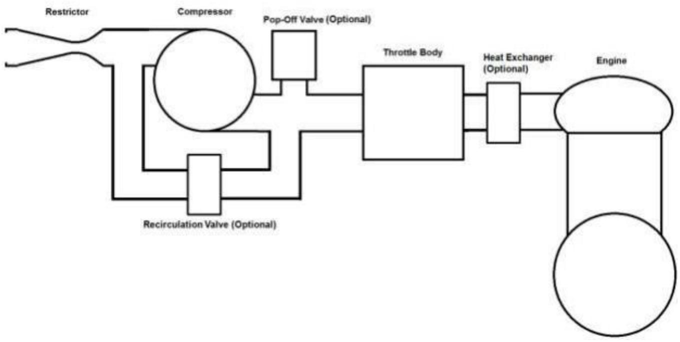


Figure 5 Formula Student Powertrain rules diagram (Formula Student, 2018)

For the project, the engine and turbocharger system must comply with the Formula Student rule. Figure 5 shows a diagram illustrates the order with which the turbocharger system must comply. This diagram illustrates that the restrictor must be in front of the turbocharger compressor, a recirculation or blow off valve can be used, and the intercooler must be located downstream of the throttle body which is after the turbocharger.

The several rules (Formula Student, 2018) influence the project and must be followed to have a system which complies to those rules;

IC1.7.1 The restrictor must be placed upstream of the compressor and the throttle body must be placed downstream of the compressor. Thus, the only sequence allowed is restrictor, compressor, throttle body, engine.

IC1.7.2 The intake air may be cooled with an intercooler (a charge air cooler). Only ambient air may be used to remove heat from the intercooler system. Air-to-air and water-to air intercoolers are permitted. The coolant of a water-to-air intercooler system must comply with Rule T8.1.

IC1.7.3 If pop-off valves, recirculation valves, or heat exchangers (intercoolers) are used, they may only be positioned in the intake system as shown in IC1.6.1.

IC1.7.4 Plenums anywhere upstream of the throttle body are prohibited. For definition, a "plenum" is any tank or volume that is a significant enlargement of the normal intake runner system. Teams are encouraged to submit their designs to the Rules Committee for review prior to competition if the legality of their proposed system is in doubt.

IC1.7.5 The maximum allowable ID of the intake runner system between the restrictor and throttle body is 60 mm diameter, or the equivalent area (i.e. 2827 mm^2) if non-circular.

IC1.7.6 If an intercooler/aftercooler is used, it must be located downstream of the throttle body.

T8.1 Coolant Fluid Limitations Water-cooled engines must only use plain water. Electric motors, accumulators or HV electronics may use plain water or oil as the coolant. Glycol-based antifreeze, "water wetter", water pump lubricants of any kind, or any other additives are strictly prohibited.

Anti-lag systems are not prohibited in Formula Student. The ETC concept can be deemed as an anti-lag system, as the electric motor accelerates up the turbocharger to create boost pressure when the turbocharger cannot, thus, acting as an anti-lag system.

ICE theory

BSFC

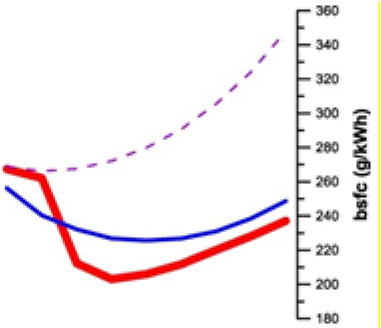


Figure 6 BSFC graph showing N/A, turbocharged and twin-staged Turbocharged (Giakoumis, 2016)

Brake Specific Fuel Consumption (BSFC) is a unit of measurement that determines the amount of fuel consumed by the engine to produce unit power for a specific engine load and speed. Figure 6 above shows a N/A engine (dotted line) against a single stage turbocharger (solid blue line) against a twin-staged turbocharger (solid red line). (Edgar, 2008)

$$bsfc\left(\frac{g}{kW.hr}\right) = \frac{\dot{m}_{fuel}(\frac{kg}{hr})}{P(kW)} \qquad [1]$$

Equation 1 is the BSFC equation. Turbocharging increases the charge density of air which in turn leads to higher fuel being burnt per stroke relative to the equivalant N/A engine. The decreased strokes to burn the same amount of fuel implies significantly lower frictional losses in a turbocharged engine leading to higher power per unit of fuel burnt.

Pressure/Volume

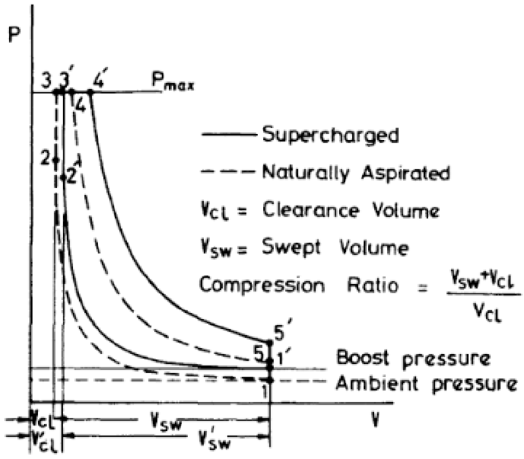


Figure 7 Pressure/Volume diagram showing turbocharged against N/A(Watson & Janota, 1982)

Figure 7 above is a Pressure/Volume (P/V) diagram, it indicates the combustion cycle at point 1 is the start of the compression stroke. From 1-2 the piston is compressing the volume in the cylinder increasing the pressure decreasing the volume. 2-3 is the combustion stroke as 3-4 is a continuation of this stroke it occurs at constant pressure. Points 4-5 are the expansion process following the end of the combustion. Point 5-1 are the exhaust valves opening allowing for the gasses to escape and the cycle to start over again.

The P/V diagram shows work that an engine outputs. Therefore, points 1-2-3-4-5-1 show the work output per cycles completed. Figure 7 represents both the work output of an engine through both N/A and forced induction in this case supercharging.

The supercharged engine starts at a higher pressure than the N/A engine. At points 2'-4' more fuel can be burnt due to the availability of more air mass. The same volume as the cylinder volume is constant the pressure changes due to the higher density. Consequently, the forced induction engine produced more power due to its area within the working points.

BMEP

Torque is the standard unit of measurement that is used to measure an engines ability to do work. Another useful way to measure and engines performance is to divide the work per cycle by the cylinder volume per cycle. In turn this unit of measurement has a force per unit and is called Brake Mean Effective Pressure (BMEP).

$$BMEP(kPa) = \frac{P(kW)n_R \times 10^3}{V_d(dm^3)N(\frac{rev}{s})}$$

The BMEP equation above shows how the Brake Mean Effective Pressure can be calculated. Firstly, the power (P) is multiplied by the number of crank revolutions (n_R) with two being the number for four-stroke engines. This is then divided by the (V_d) which is a calculation of the engines swept volume multiplied by the engines speed at any given point in the speed range. N/A engines which are Spark Ignition (SI) engines usually have a BMEP of 850 to 1050 kPa. At the point of maximum engine torque BMEP values are usually 10-15% lower. Turbocharged SI engines have a maximum BMEP of 1250 to 1700 kPa, at maximum rated power BMEP is between 900 to 1400 kPa.

Turbo Compressor Theory

Velocity Diagrams

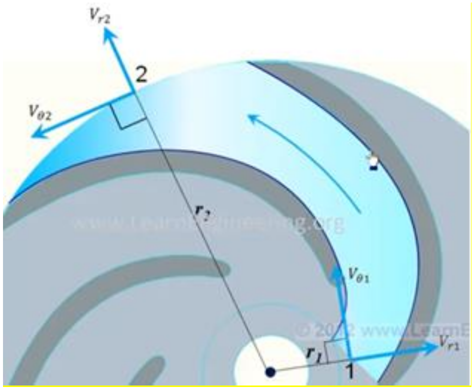


Figure 8 Velocity diagram of air entering a turbo compressor (Learn Engineering, 2013)

To understand how turbochargers increase the flow rate of air to an engine, a velocity diagram shown in Figure 8 has to be understood. For a turbocharger to work, a torque has to act upon the system; this torque will be the exhaust gasses acting upon the compressor of the turbocharger and in turn spinning the compressor. This equation of torque acting upon the compressor is shown below.

$$Torque = \dot{m}(r_2 V_{\theta 2} - r_1 V_{\theta 1})$$

If the turbocharger is rotating an angular velocity (ω), the power required to maintain the flow of the compressed air would be the torque multiplied by the angular velocity. 'Power required for this fluid motion can be taken as difference in product of blade velocity times tangential fluid velocity.'

$$power_{req} = \dot{m}(r_2 V_{\theta 2} - r_1 V_{\theta 1}) \times \omega$$

By dividing the power by the weight of the fluid flow, the energy head that is required to maintain a specific flow rate would be acquired.

$$h = \frac{1}{g}(U_2 V_{\theta 2} - U_1 V_{\theta 1})$$

Therefore, if the power by the fluid flow is greater than zero the system is acting as a compressor, if the power is less than zero the system is acting as a turbine. (Learn Engineering, 2013)

In simpler terms a turbocharger is a pressure increasing machine, it consists of a rotating impeller which has the job of increasing the energy of the fluid in this case air. The impeller increases the pressure and velocity of the air entering the inlet also known as the eye of the diffuser. The pressurised air is created in the diffuser as the flow leaved the impeller, it is converted into pressure energy. This is collected by the volute and directs the flow into the outlet.

Thermodynamics (power required to increase power for given flow rate)

$$W_u = \frac{\dot{W}}{\dot{m}} = \frac{\left(\frac{2\pi N}{60}\right)\tau}{\dot{m}} = C_p(T_{out}, 0 - T_{in}, 0) \quad [1]$$

$$n_s = \frac{\dot{W}_{is}}{\dot{W}} = \frac{T_{in}, 0(TT_{t,t}^{\gamma-1/\gamma} - 1)}{(T_{out}, 0 - T_{in}, 0)} \quad [2]$$

Power is always required to increase pressure for any given flow rate. The power required to increase the compressor outlet pressure for a given mass flow rate can be derived from the above equations. Specific work incurred by the compressor to operate at a particular rotational speed can be derived from the (eq.1) which in-turn can be determined on the basis of required pressure rise (eq. 2) for a given isentropic efficiency. This power along with the mechanical shaft losses are thereby demanded from ETC.

Boost pressure against charge air density

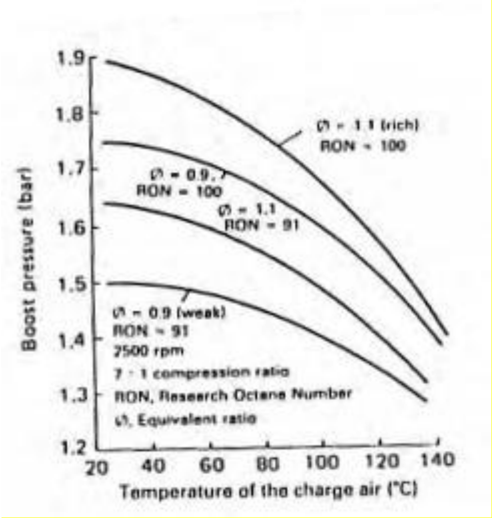


Figure 9 Effects of density change to boost pressure (Stone, 1992)

Figure 9 above shows the effects of pressure against temperature for different fuels. Here there is an engine running at a constant speed of 2500 RPM with a compression ratio of 7:1. The Figure also shows the Air to Fuel ratios (AFR) that have to be used to prevent knock. Turbochargers compress ambient air and increase the pressure and temperature. Fuels which have a higher-octane rating are less prone to knock as the knock limitation is reduced due to the higher octane. Knock limitation as well as AFR's will be discussed later in this report.

Nevertheless, as temperature is increased the boost pressure is decreased. This is why charge cooling is important as to increase efficiency of an engine, a cooler operating temperature is usually desired. Cooler operating temperature allow for higher boost pressure to be used therefore increasing the engines power output.

There are three parameters which effect a turbocharged engine is affected by, temperature and pressure at the inlet as well as the manifold pressure. The effects of the temperature and pressure at the inlet have been discussed. However, the pressure in the manifold can restrict power output, having a pressure build up in the manifold will restrict the turbines ability to do work and slow the turbine down. (Stone, 1992)

Compressor efficiency of Air density

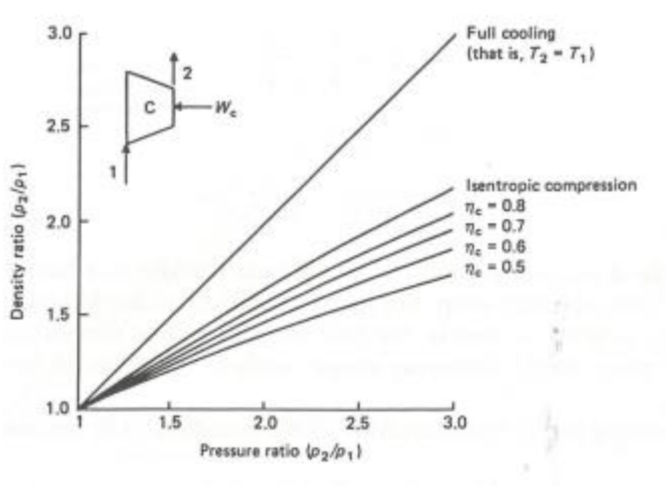


Figure 10 charge cooling effects (Stone, 1992)

Figure 10 above shows the effects of the compressor efficiency on charge density the effects of cooling can also be seen. The temperature rise from the compressor decreases the density ratio, especially at the higher-pressure ratios. The figure shows the maximum effects of cooling and the minimum. (Stone, 1992)

The maximum effects of cooling is known as "full cooling", this is when the inlet temperature of the turbocharger is the same temperature that the engine draws the pressure from the turbocharger to the cylinders also known as T2=T1. However, this maximum cooling effect can be very hard to achieve due to the thermodynamic principles of turbochargers.

Figure 10 shows the effects of the isentropic compression, this can be noted as efficiency of modern turbocharged internal combustion engines. Therefore, the maximum effects of cooling where T2=T1 is the ideal value to achieve, however this is not achievable as modern engines supply air to the engine at a temperature of 45°C after turbocharger cooling (Stone, 1992). Hence, the isentropic compression of 0.7 is the actual value to benchmark.

The isentropic compression at 0.7 can be calculated with the induction temperature of 20°c at the turbocharger inlet. This result shows the effective cooling of 70%, this is the most realistic value. For example, temperature of the pressure leaving the compressor in a turbocharger is around 150°c this means that after the cooling cycle the intake air temperature is 45°c.

Academic Aim

To design and develop an electrically assisted turbocharger with energy recovery based around the Formula Student rules and regulations, to minimise the boost threshold and increase efficiency.

Objectives

- Create a mathematical model of the base powertrain unit
- Design an electrically assisted turbocharger with energy recovery
- Demonstrate that the engine develops more torque low down in the RPM range
- Demonstrate that the engine has an Increased torque efficiency throughout the RPM range
- Show that the simulated results and the dynamometer results match

This project aims to design and develop the ETC. Energy recovery within the turbocharger is an achievable aim but due to the short timeframe of this project it will remain a theoretical design. This means the report will discuss the feasibility of developing the energy recovery system but investigates the main academic question which is, “Is an ETC viable for a small engine?”

Chapter 1 Literature review

Development of the "hybrid turbo," an electrically assisted turbocharger

MHI's Hybrid turbocharger is electronically assisted, this turbocharger has improved torque response, fuel consumption and purer exhaust gasses.

Figure 11 MHI's Hybrid Turbocharger (Ibaraki, Yamashita, Sumida, Ogita, & Jinnai, 2006)

MHI have integrated their electronic motor into the turbocharger housing shown on Figure 11 but effectively use it to accelerate their turbochargers compressor at low engine speeds, when the turbocharger is effectively up to speed the rotational velocity of the turbocharger shaft energy is converted into electricity, using their turbocharger as a generator as well.

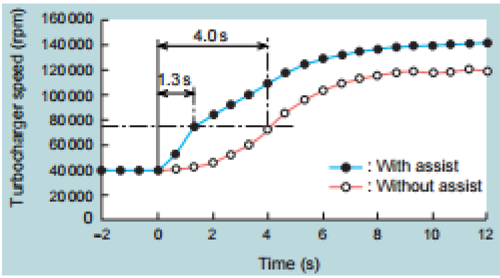


Figure 12 MHI's Hybrid Turbocharger results with and without electrical assist (Ibaraki, Yamashita, Sumida, Ogita, & Jinnai, 2006)

The turbocharger's response time to create boost is increased by 67.5% up to 70% this means that the turbocharger has had a reduction of 70% in turbo-lag meaning the engine will be a lot more responsive and driver friendly whilst also increasing torque by 50% in the low engine speed (Ibaraki, Yamashita, Sumida, Ogita, & Jinnai, 2006).
This report has promising results as well as showing their response of their electronic control over the turbocharger. This is shown in Figure 12.

Garrett Electric Boosting Systems (EBS)

Garrett's electric boosting system incorporated an electric motor into their turbochargers oil bearing housing shown on Figure 13 & 14. Garrett's idea was to maximise their engines characteristics at low engine speeds to increase torque response, volumetric efficiency, fuel consumption, fast response and power generation.

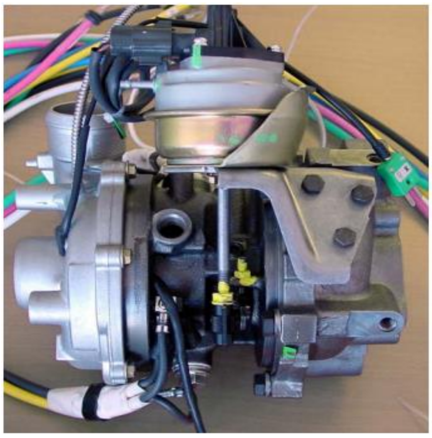


Figure 13 Garrett's Electric Boosting System (EBS) (Arnold, et al., 2005)

The design is unique with an incorporated variable geometry wastegate turbocharger. This design looks to still use their wastegate (and not the electric motor) to monitor the exhaust gasses around the turbocharger, to prevent the turbocharger from surging or chocking.
As well as a unique design their turbocharger's response drastically increased. At 1000 engine rpm the torque without the electronic turbo was 325 Nm however, with the electronic boost was 400 Nm that is an increase of 23%. The electric motor does need a supply of 2.1Kw of electrical input (Arnold, et al., 2005).

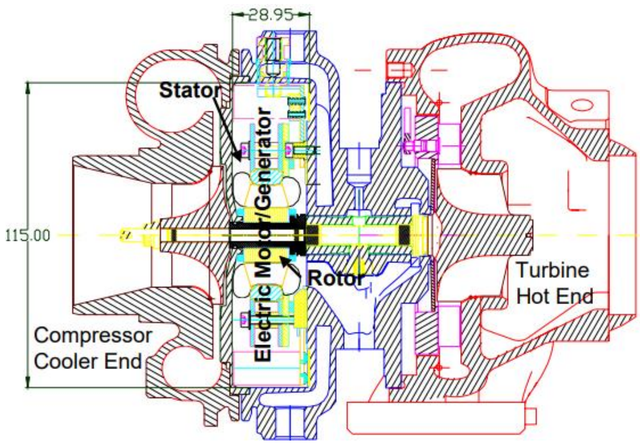


Figure 14 EBS diagram (Arnold, et al., 2005)

Garrett extended their baring house to compensate for an electric motor/generator. This design raises several problems as well as several excellent design aspects. Firstly, the electric motor/generator was designed to fit into the cool side of the bearing housing behind the compressor. This allows for the motor/generator system to not get drastically heat soaked as well as keeping the compressor clear for maximum airflow.

Secondly there are several drawbacks. The turbocharger shaft is rotating at high speeds over 100,000 rpm and for this to spin with ease, rotating mass must be kept to a very minimal and light weight. Magnets for electric motors on the turbine shaft would effectively reduce the rotational performance of the turbocharger itself. Furthermore, the design must effectively elongate the turbochargers shaft to allow for the design to be implemented. This results in several design failures such as heat soak and magnet damage.

Design of an Ultra-High-Speed Permanent-Magnet Motor for an Electric Turbocharger Considering Speed Response Characteristics

Lim, Kim, Hwang, & Hong in 2017 designed and analysed the effects of a 4 kW and 150 krpm high speed motor. The design aimed to achieve torque and speed requirement. The study also considered a fast-response as an objective to eliminate turbo-lag effectively. With the results of a 44.9%, the rising speed of the boost pressure is improved; this was due to the motor design (Lim, Kim, Hwang, & Hong, 2017).

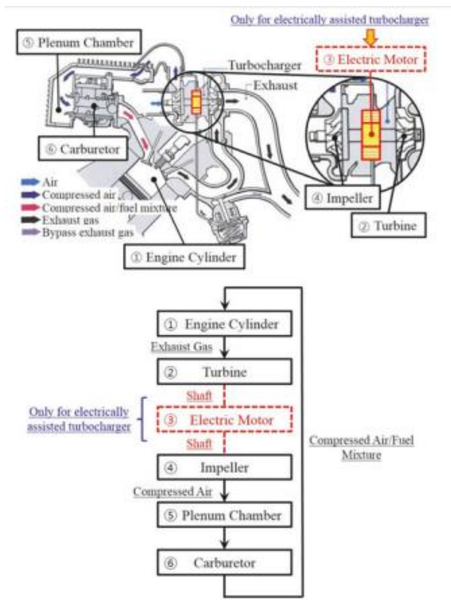


Figure 15 Ultra-High-Speed PM design (Lim, Kim, Hwang, & Hong, 2017)

The diagram in Figure 15 shows the design implement of the ETC turbocharger. The high-speed motor is implemented within the turbocharger's bearing house, with a compressor and turbine on either side.

Their motor design was to use a permanent-magnet motor also known as a PM motor. This was chosen for the characteristics of a PM motor with high-power density, this leads onto high-speed system response. However, this PM motor does cause problems at high speed such as iron loss, voltage limitation, windage loss and structural problems. This is all due to the high speeds that these motors can reach.

SPECIFICATIONS OF THE DESIGNED MOTOR

Quantity	Unit	Value	Versus Prototype
Power	kW	4	+14.3%
Torque	N·m	0.38	-
Rated speed	rpm	100,000	+14.3%
Max. speed	rpm	150,000	-
Outer diameter	mm	90	-
Axial length	mm	13	−35.0%
Core material	-	20PNF1500	-
Magnet material	-	Nd-Fe-B	-
Response time (80 krpm)	ms	193.47	−35.1%

Figure 16 PM Specifications (Lim, Kim, Hwang, & Hong, 2017)

Figure 16 shows the specifications of the PM electric motor. This motor specification will be considered when the project electric motor will be specified. However, 4 kW is drawn from the source to achieve speeds of over 100 krpm this amount of power will be difficult to achieve from a 12 V vehicle battery.

Development of High-Speed Motor and Inverter for Electric Supercharger

Nishiwaki, Iezawa, Tanaka, Goto, & An in 2013 developed an electric supercharger, it works besides a turbocharger to reduce the turbo-lag. This system uses a conventional turbine of a turbocharger and an electric motor called an electric supercharger or E-Supercharger for short (Nishiwaki, Iezawa, Tanaka, Goto, & An, 2013).

As the E-Supercharger works besides a conventional turbocharger it's a developed two stage turbocharger. In performance tests the response time of the compressor to 90 krpm is 0.7 sec, this is the maximum speed of the E-Supercharger, the two-stage system demonstrates a 43% improvement in response time at 1500 rpm at engine idle compared to a conventional two stage turbocharger system. This system was tested on a 1.5 litre petrol engine (Nishiwaki, Iezawa, Tanaka, Goto, & An, 2013).

Battery System	12 V
Maximum Motor Power / Continuous Operation Power	2.4 kW / 1.0 kW
Maximum Speed	90,000 rpm
Time to Maximum speed	< 0.7 sec
Maximum Current Draw	< 320 A

Figure 17 motor specification (Nishiwaki, Iezawa, Tanaka, Goto, & An, 2013)

Figure 17 shows the motor specifications used in the E-Supercharger. Compared with the motor specification shown on Figure 16 the motor uses a lot less power as well as showing a similar result however this motor is specified for a max rotational velocity of 90 krpm whereas the other motor needs a max rotational velocity of 150 krpm.

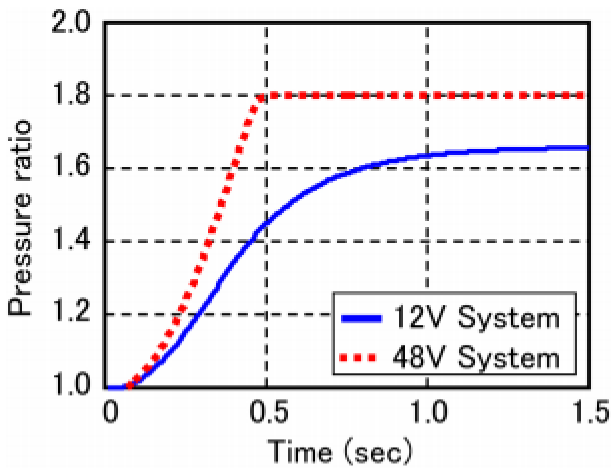


Figure 18 12V system compared with 48V system (Nishiwaki, Iezawa, Tanaka, Goto, & An, 2013)

Furthermore, this research report developed a 48 V system to see the difference in results. The pressure ratio is the compressor discharge pressure over the compressor inlet pressure. This shows the absolute pressure of the systems. The 48 V system is a lot more effective producing boosted air pressure as it makes max absolute pressure within 0.5 sec therefore showing a faster response compared to the 12 V system which achieves max absolute pressure ratio at 1.5 sec with a second difference in response and around 0.75 pressure ratio difference (Nishiwaki, Iezawa, Tanaka, Goto, & An, 2013).

Design and Characterization of an E-booster Driven by a High-Speed Brushless DC Motor

As high-speed brushless motors spin their mechanical stress increases this is due to high centrifugal forces. Tavernier's and Equoy's in 2013 investigated this problem and it was decided to make retaining sleeve to encase the magnets from damage caused by mechanical stress. The material choice was titanium, Inconel and carbon fibre (Tavernier & Equoy, 2013). The mechanical stresses (σr and σθ) are calculated using the equilibrium equation: with σ being the stress tensor and F the force density. Using the different symmetry, this equation [1] can be simplified: [2] with (σr , σθ) being the mechanical stresses and Fr the radial force density (Tavernier & Equoy, 2013).

$$div\bar{\bar{\sigma}} + \vec{F} = \vec{0} \quad [1] \text{ (Tavernier \& Equoy, 2013)}$$

Using the different symmetry, this equation [1] can be simplified: [2]

$$\frac{d\sigma_r}{dr} + \frac{\sigma_r - \sigma_\theta}{r} + Fr = 0 \quad [2]$$

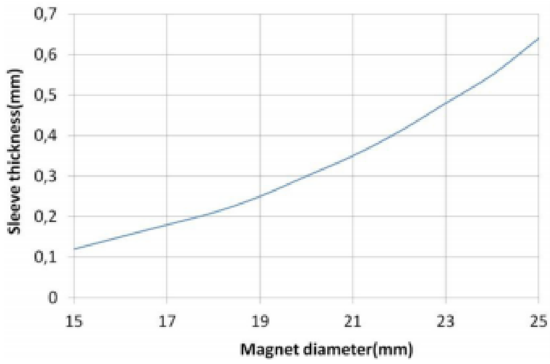


Figure 19 magnet diameter against sleeve thickness (Tavernier & Equoy, 2013)

From the calculations for an electronic motor working at maximum rotational velocity of 100 krpm, Figure 19 shows the sleeve wall thickness compared to the magnet diameter. It is a linear relationship with wall thickness to magnet diameter. Furthermore, the material selected to be the sleeve on the magnets was titanium this was chosen due to its ease of manufacturing to develop a prototype.

This report simplified a problem which now can be solved. In the project a high-speed electric motor will be used. The problem was indicated and now there is a method to calculate wall thickness of sleeves due to radial stress.

Optimization of an Electric Turbo Compounding System for Gasoline Engine Exhaust Energy Recovery

Weilin, Huang, Wei, Zhang, & He in 2011 looked to optimise an ETC system on an ICE engine, the report simulated the response using GT-Power a 1D engine simulation software. The report highlighted the fact that 33% of fuel burned in combustion is waste energy down the exhaust system. The ETC system was placed parallel to a conventional turbocharger. The exhaust gasses were controlled over the ETC by a fixed geometry turbine (FGT) or variable nozzle turbine (VNT) (Weilin, Huang, Wei, Zhang, & He, 2011).

However, this report looked into a simpler solution to generate electricity from a turbocharger by placing an ETC system which is effectively a turbo-generator. These systems are much simpler compared to complex ETC systems which improve transient responses as well as recover energy. The ETC system in this report was a generator attached to a turbochargers turbine, therefore making a turbo-generator system.

Furthermore, the engine in the simulation is a 1.8L engine, so compared with the engine used in the project is much larger. Consequently, a larger displacement engine can use a larger turbocharger. As turbochargers get larger in size, they have a negative correlation with their speed line bands on their maps. Effectively if the calculations for turbocharger sizing fits to a larger turbocharger then it possibly could be a better option to use a larger turbocharger. To the lower rotational speed line bands, it could be more effective generating electricity through a larger turbine, as well as to also improve the transient response.

This report assumed the efficiency of the turbo-generator to be 70% and the engine’s alternator to be 55%. The ETC with an FGT operating at a constant speed resulted in a 3.8% improvement in fuel efficiency abiding by the US06 driving cycle. Whereas the ETC with a VNT improved the fuel efficiency by 8% compared with the FGT system. Nevertheless, this report made assumptions to achieve these results.

Diesel Engine Electric Turbo Compound Technology

Hopmann & Algrain in 2003 looked to demonstrate the fuel efficiency increase of adding an electric turbo compound system on a class 8 truck engine. Class 8 trucks are the largest commercial trucks allowed on the US roads. The benefits of their simulations showed that the ETC system increased fuel consumption; at rated power, the fuel consumption can possible be reduced by 10% (Hopmann & Algrain, 2003).

The report described the design and test of their ETC system, the design consisted of an electric motor and generator located in the turbocharger’s housing. Figure 20 shows the design of the turbocharger.

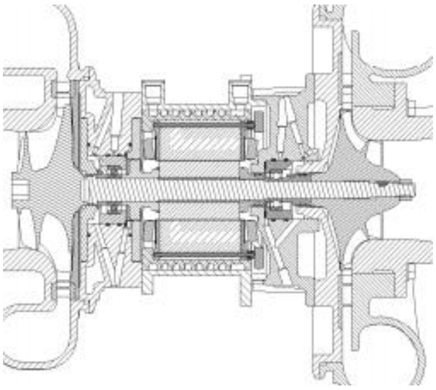


Figure 20 Electric Turbo Compound design (Hopmann & Algrain, 2003)

Furthermore, this report had to analyse the thermodynamic effects of heat transfer within the system. This is because turbochargers reach temperatures of over 1000°c at the exhaust side; this heat radiates down the turbocharger and up to the 'cold side' better known as the compressor housing. Consequently, the magnets within the motors and generators cannot withstand temperatures such as these and break down rapidly losing their magnetic properties.

On the other hand, the design allows both the intake side and exhaust side to be free of any restrictions, this will allow for a smoother airflow into the turbocharger as well as letting the exhaust gasses escape more easily with minimal back pressure.

The electric turbocharger compound system also provides better control over the power extracted by the turbocharger and gives far greater control of the turbocharger's boost pressure and therefore, the engines air to fuel ratio. In addition, it improvs driveability by controlling the boost pressure developed by the turbocharger.

Extracting the information above from the review of this research allows for a greater control over a turbocharged vehicle. In the past most, turbocharged vehicles had very low transient behaviour meaning that they had a large lag period to make any boost. This is also true for many older turbocharged race cars making them difficult to control and difficult to predict when boost will 'kick in.'

Turbocharged Formula Student vehicles are notorious for this problem, therefore having greater control over the air entering the engine will allow for greater drivability from the driver. Additionally, with greater control over the air entering the vehicles engine air to fuel ratios could be tuned and adjusted easily this is due to having control over both the air and fuel entering the engine.

Case study Innovation in engines – New Full Electric Turbocharger enabling higher control and power whilst lowering emissions

Aeristech's ETC can reduce turbo-lag and gain 14.3% more power in transient operation. The ETC uses a 35kW motor compressor to be accelerated to 150,000 RPM in 0.5 seconds. When tested the steady state torque was 675Nm at 1200rpm with the ETC without the ETC the steady state torque is 635Nm, which is an increase in 6.3%.

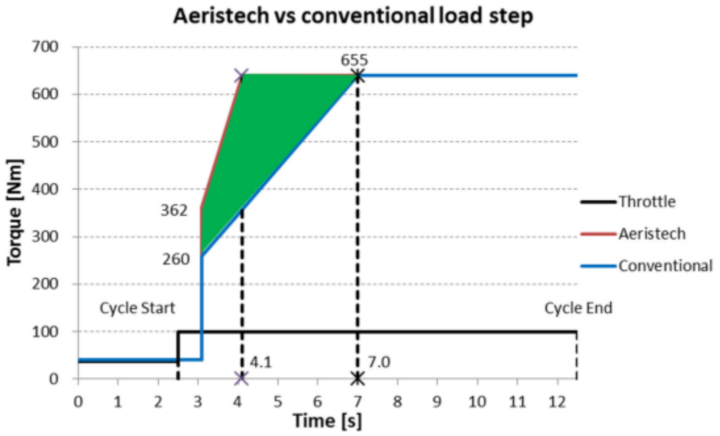


Figure 21 Aeristech ETC against conventional Turbocharger (Aeristech, 2015)

The green area produced within Figure 21 is the additional torque produced by the ETC, the ETC has an improvement of 41% in transient response. The engine only uses 12mg/stroke more fuel when comparing the ETC to a conventional turbocharger. Therefore, the engine saves up to 9.6% of fuel at a steady state of 1200RPM, there is a 14.3% engine downsizing effect between the ETC and a conventional turbocharger. (Aeristech, 2015)

This study shows a promising effect which can take place using an ETC. Figure 21 show that the Aeristech ETC has a greater area under the curve compared to the conventional turbocharger, from 2.5s when the throttle is opened to 100% it only takes 1.6s for the engine to reach maximum torque. Compared to the conventional turbocharger which takes 4.5s to reach maximum torque.

Not only does the ETC reduce lag it also created a reduction in fuel consumption per stoke compared with the conventional turbocharger. The study shows promise of a 9.6% fuel saving at a steady state, therefore showing that the ETC has more promise than just turbo-lag system.

Furthermore, the study states that a 35kW motor was used in the ETC therefore, stating that a high-powered motor will have to be used in any ETC for performance gains. The turbochargers inertia and other losses will have to be calculated to size up an appropriate motor.

Study on engine efficiency and performance improvements through hybrid turbocharging assisting

This study looks to simulate different engines speeds and see the performance improvements. The ETC has a massive increase in transient response and improves it by 70% at 1200rpm with a 3kW motor. (Dobre, 2014)

However, this study also looks at several different motor power effects on the ICE. The ICE used is a VEP 2.0l SI, the ICE was simulated on GT-Power, the study simulated steady state responses at various engines speeds and altered the motor power to establish the effects of the ETC for a given engine speed.

Furthermore, the study states that when high wattage was applied by the motor would make their turbocharger '*go beyond the surge limit*' this can cause the turbocharger to fail. The study also states that with the same power applied to all speed ranges by the ETC the higher the engine speed the less effect the ETC has to the torque produced. This can come down to the exhaust gasses mitigated the electric motors applied power onto the turbine shaft.

This study states that the lower the motor power the lower the response from the ETC. However, there can be a known sweet spot, meaning that a certain power implied by the motor can effectively reduce lag but also aid in transient response and torque levelling. However, if too much power is applied by the ETC the system mass flow will remain the same the pressure will increase until the turbocharger starts to act in surge. This could be potentially reduced by adding in features to limit the motor power so that the ETC effects can be altered to the ICE.

Development of Two-stage Electric Turbocharging system for Automobiles

Electric compressors or electric superchargers are systems which take an electric motor to power a turbocharger compressor. MHI have developed a two-stage electric turbocharger this is to reduce turbo-lag as well as improve emissions. (An, Shibata, Suzuki, & Ebisu, 2015)

MHI electric compressor uses a 12V 3kW motor to accelerate the compressor to 90krpm with on 0.6s. MHI used a 1.5l engine and tested the Two-stage electric turbocharger system, the engine produced a maximum torque of 230Nm and a maximum power of 113kW. MHI compared their results to a conventional two-staged turbocharger and a single turbocharger. Results showed that an improvement in the transient response was up to 43% compared with the two-staged turbocharger as well as, a 70% improvement in exhaust gas pressure. As the exhaust gas temperature is improved the study states that the knock limit was also reduced due to the reduction in gas remaining with in the engine's cylinders, furthermore, aiding in pumping loss and combustion instability.

Additionally, the two-stage electric turbocharger also showed an increase in engine torque when compared to both conventional two-staged turbocharger and single turbocharger. As well as stating that '*improvement in fuel consumption and drivability with ultra-high EGR can be expected*.'

Furthermore, MHI simulated their response using GT-Power and concluded that a 1.1L engine with the two-stage electric turbocharger can improve the transient response by up to 42% compared to an engine with 36% more displacement. Additionally, the two-stage electric turbocharger can expect a 10% reduction in fuel consumption by the ICE.
MHI have shown through this study that performance gains can be made mating electric motors to boosted systems and show a promising increase in emissions as well as fuel consumption.

Literature review conclusion

This conclusion will complete the literature review section of the report. Firstly, the designs of these turbochargers within the literature review are complex and will take time to manufacture. Examination of these designs shows failure can occur within these systems due to heat radiating with in the turbocharger destroying magnets. Solutions such as mounting the motor onto the intake side of the turbocharger before the compressor will decrease the heat stresses imposed onto the system, however this will also impose an air restriction.
Secondly, demonstrating the response of the turbocharger over time will aide in understanding the motor/generator needed for the system. Furthermore, it allows a comparison between the conventional turbocharger and the ETC system. This can be done by fitting the turbocharger with a speed sensor and measuring the rpms of the turbocharger against time.
Thirdly designing a motor to withstand speeds of over 100 krpm will be difficult and take up a lot of time whilst also straying away from the project's aims and objectives; henceforth a motor will be specified as well as a turbocharger to be compatible and work within the turbocharger's map. As this project looks to increase engine performance in the low engine speed ranges, peak horsepower as well as torque do not have to be focused on; this will then allow for a lower turbocharger speed.
Lastly taking the points made in the paragraph above an adequate turbocharger should be sized up with low speed bands on the turbocharger map. This will then allow for a motor to be bought in and used while being able to produce boost pressure.

Chapter 2 Generating Electricity

Theoretically an electrical motor can be used as a generator as well as a motor to transmit drive. Hence, this section of the report will be to look at the electricity that can potentially be generated from the turbocharger as well as Electronic Speed Controller (ESC) and generator controllers.

Generating power from turbochargers

Turbine generators have been in the industry for years, with wind turbines and other forms of energy generation. There has been great focus in the 21st century on reducing engine emissions for this. Manufacturers have been downsizing engines and improving emissions with forced induction and hybridised vehicles. However, it is already established that the modern combustion engine translates only 33% of combustion into a moving force. Therefore, large amounts of energy are wasted with gasses, heat, noise and vibration within the engine. It is possible to use the 33% being wasted as exhaust gasses for electrical energy generation. This can be done by adapting a turbocharger to have a generator maximise the wasted exhaust gas energy into electrical energy.
Looking at the report 'Electric Turbocharging for Energy Regeneration and Increased Efficiency at Real Driving Conditions' can show the amount of electrical energy that is possible to be made from turbochargers.

Electric Turbocharging for Energy Regeneration and Increased Efficiency at Real Driving Conditions

Dimitriou, Burke, Zhang, Copeland, & Stoffels in 2017 investigated the response time and electrical generation from the motor-generator within a 1D engine simulation software. The report looked to reduce the transient response of the turbocharger; this in turn makes an engine more responsive with a 90% simulated improvement in the engine's response. Furthermore, the turbocharger generated up to 6.6 kWh of energy whilst improving the thermal efficiency (Dimitriou, Burke, Zhang, Copeland, & Stoffels, 2017).

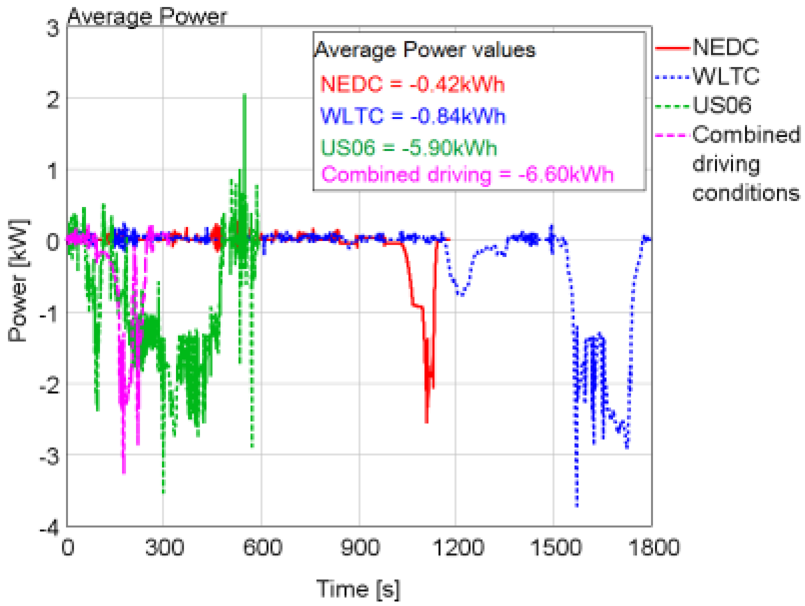


Figure 22 Power produced over time with different driving conditions (Dimitriou, Burke, Zhang, Copeland, & Stoffels, 2017)

The energy study in this report shown in Figure 22 was conducted using NEDC, WLTC, US06 cycles, as well as real driving condition cycles for which the study gathered data.

The reports simulated different responses to the turbocharger. The study stated that the motor-generator at low to medium load is ineffective. However, at 'extra-urban' driving conditions, it can generate large amounts of electrical energy. Nevertheless, the amount of energy recovered from the generator in this case depends on the driving conditions. From this understanding a simulation can, be made to simulate the electrical generation response of a racing vehicle turbocharger.

The report tested their simulation under various driving conditions and proved that the energy generated can exceed 1 kW of electricity with an average of 6.6kW (Dimitriou, Burke, Zhang, Copeland, & Stoffels, 2017). The simulation did not consider any electrical losses, such as alternator, converter and battery losses which, will reduce the amount of energy generated.

From this report the information that can be taken away to demonstrate the turbocharger's ability to make electrical energy, can be simulated. Possibly using Matlab and Simulink to create a generation simulation where the turbocharger's speed, shaft torque and power can be placed as inputs to see the amount of electricity the turbocharger can theoretically produce. Many different variables such as inertia, electrical losses and battery losses, can be considered to provide a more realistic simulation.

Investigation of an electrically assisted turbocharger with energy recovery for heavy duty diesel engine

Rezk, Jeff, Jupp, Keith, & Allport in 2016 simulated a heavy-duty diesel engine's response using GT-Power 1-D engine simulation software. As Original Equipment Manufacturers (OEMs) are now being forced to tackle tighter emissions regulations, they have no choice but to reduce engine capacitates and opt for smaller displacement engines. However, smaller engines struggle to produce the desired response that larger displacement engines. Therefore, to support downsized engines an advance in boosting technologies is required to compensate for specific power output and improvements in engine efficiencies.

One form of advanced boosted technology is an electrical turbo assist, eTurbo™ which when simulated and tested provided more freedom to manage the engine AFR. Additionally, the report highlighted that addition to their eTurbo™ lead to a reduction in engine pumping, resulting in higher engine efficiency, as well as a reduction in mass flow and pressure ratio.

Through the use of the Variable Turbine Geometry (VTG) Turbocharger and the Electronic Turbocharger Assist (ETA) on a heavy-duty diesel engine. The steady state performance lead to, a reduction in engine back pressure and pumping work. From this reduction an overall engine efficiency and fuel economy increase was shown with results of 4.1% in engine efficiency and 3.9% in fuel efficiency (Rezk, Jeff, Jupp, Keith, & Allport, 2016).

Generation simulation

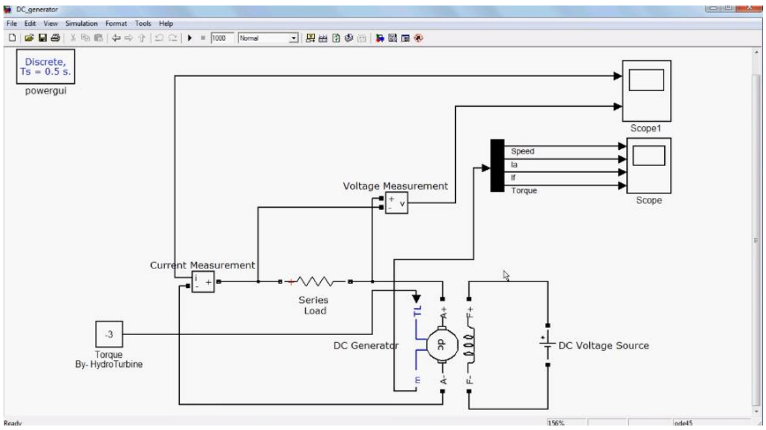


Figure 23 Tutorial of a DC Generator (Kumar, 2012)

The simulation in Figure 23 is a tutorial by Satendra kumar who designed a simple DC generator. Nevertheless, this simulation can be used as a foundation for the turbocharger's generator simulation. This simulation uses torque from a hydro-turbine to rotate a DC generator to see the response of voltage and current as well as the speed, torque, armature current and field current.

However, this simulation is a steady state simulation which can only tell the user the responses placed into the simulation by the user. Therefore, this simulation was designed and tested to be modified to a transient state however, when this is done, the DC generator does not work. Therefore, it is possible in making a simulation to produce the response that the report looks for. However, limited knowledge of Simulink and MATLAB means that large amounts of time is required to learn and understand the software. Time restraints prohibit the use of, MATLAB and Simulink if the project to be finished on time.

Simcenter Amesim is another piece of industrial software that can possibly simulate the whole driveline including the ETC. However, like MATLAB this software is complicated. Amesim has features that can simulate the engine including the ETC and can potentially give the data of the electricity used to accelerate the turbocharger and the electricity generated from the turbocharger. Just like MATLAB, Amesim is vastly complicated and difficult to understand therefore due to the time constraints of the project using Amesim will not allow for the project to be finished in time.

On the other hand, the project will continue and go into detail and design a plan with boundary conditions. This plan will assume the electrical response from the turbocharger can be simulated into an appropriate software.

Boundary conditions and Plan

Setting boundary conditions for this simulation with a corresponding plan can simplify the aim for the generation simulation. Firstly, boundary conditions will be set; these boundary conditions will also contain parameters such as inertia, mass and motor parameters. Secondly, the plan will compile the boundary conditions into a worked plan of how the simulation should work and the responses that will be generated.

Boundary conditions

1) Inertia of turbocharger including the inertia of the motor.
2) Motor will accelerate the turbocharger to 75krpm.
3) Motor specifications such as field current, shaft diameter and material.
4) Electricity will be generated when the exhaust gasses take over from the electric motor.
5) Data produced should show power production from the motor/generator as well as transient response, in addition to torque, crank power, mass air flow and pressure.
6) Data will need to be recorded from the vehicle when around a race track.

Boundary condition 1 – Will ensure that the simulation will have the inertia of the turbocharger in addition to the electric motor/generator. This is because the simulation without the electric motor/generator will produce different results through the transient response.

Boundary condition 2 – The motor/generator will accelerate the turbocharger to 75 krpm; this is because the turbocharger produces boost at 70 krpm from its map data. Therefore, being 5 krpm over its lower boundary it can ensure that the turbocharger is at the correct speeds to pre-boost the engine with boosted air pressure. However, this boundary condition will have to agree with boundary condition 3.

Boundary condition 3 – This boundary condition will consider all the specifications of the electric motor/generator. This will ensure that the simulation has all the motor's parameters as well as material specifications. If all the specifications of the electric motor are simulated it will minimise error in order to produce accurate results.

Boundary condition 4 – The motor/generator will start to produce electricity once the exhaust gasses take over from the turbocharger. However, as this is an unknown variable the engine will have to be simulated to understand when there will be enough exhaust gas flow to take over from the electric assist.

Boundary condition 5 – To understand the effects of the electric assist to the powertrain data from the simulation should consider the overall powertrain performance to see if the electric assist can aid the performance characteristics of the powertrain. Nevertheless, data from boundary condition 6 will need to be considered.

Boundary condition 6 – This boundary condition will need to be done before the electric assist; this is because data of the turbocharger's speed will need to be placed into the simulation to see if the overall effects of the electrical assist will affect the powertrain. Furthermore, this data from the simulation can show if the electrical assist can improve racing lap times.

Plan

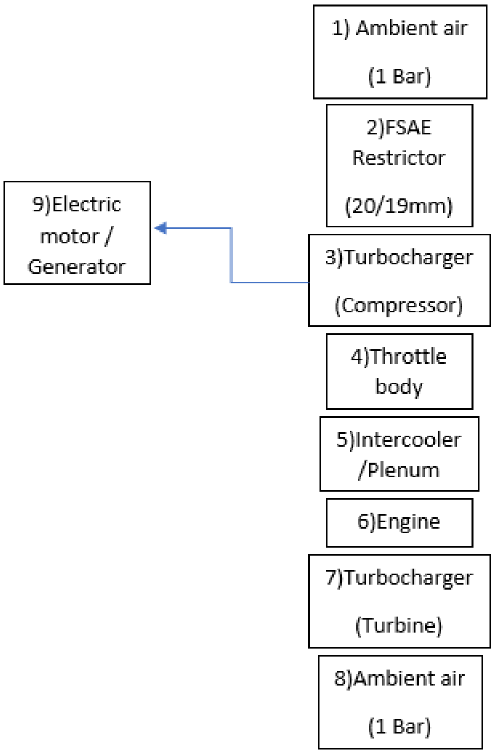


Figure 24 Generating Electricity Simulation Plan

The plan for this simulation is simple just like the Ricardo wave simulation shown on Figure 24. From part 1 ambient air - the simulation can work through the system from 1 to 8, simulating a usual engine with a turbocharger. However, part 9 of the plan is the motor/generator this can be installed to the turbocharger through the compressor side, turbine side or inside the bearing housing. Where the motor/generator is attached in testing, is where the motor/generator will be mounted and attached to the actual tested turbocharger. Just like the literature review in this section of the report the Figure 22 can be copied and directed from the simulation. Figure 25 shows an idealised average power graph over time. This can represent the power taken to accelerate the turbocharger and the power generated from the turbocharger. In this reports case from time at 0sec, the motor/generator is being used as an electronic motor to accelerate the turbocharger until around 20sec where it passes the x axis. The motor/generator acts as a generator until 30sec where the vehicle has started to brake and decelerate. As the vehicle is decelerating the turbocharger is still spinning at high speed; this means that the motor/generator is still producing electricity. Until around 55sec where the turbocharger has completely dropped off, the boost pressure and the motor/generator is acting as a motor to drive the turbocharger back up to speed, to produce boost pressure.

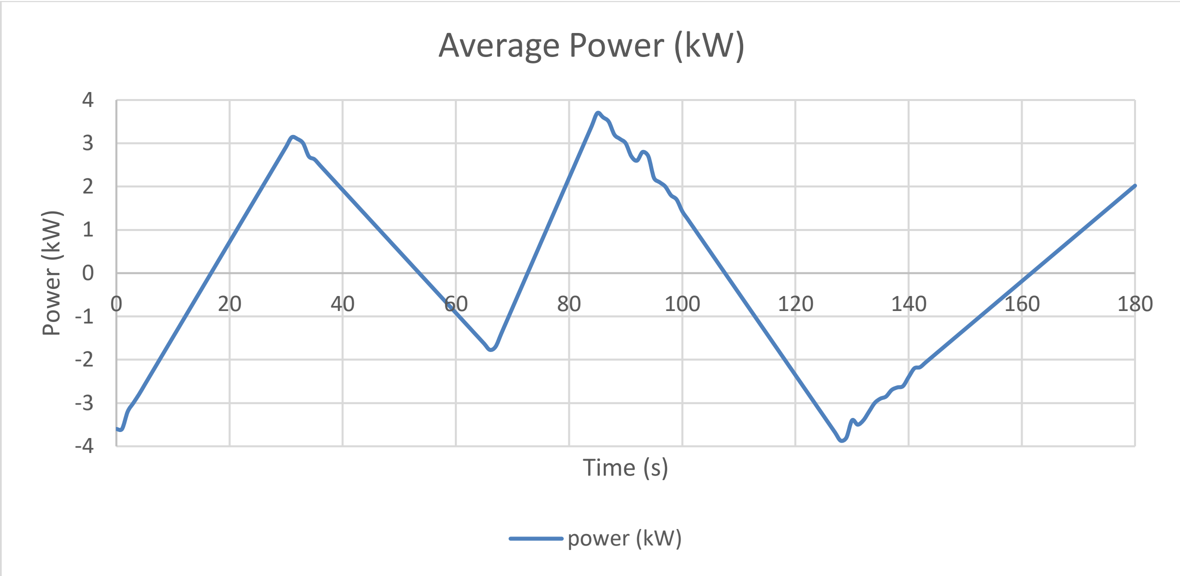


Figure 25 Average power generated over time

Furthermore, other graphs which represent the turbocharger's transient behaviour can be analysed to see if the motor/generator can affect the turbocharger transient behaviour over time.

Chapter 3 Electrical drive

Difference between ESC and Generator controllers

An ESC is used to vary motor speeds as well as power and torque. Whereas a generator uses a motor/alternator to generate electricity from a moving component, for instance it can use a turbines speed to generate electricity. Figure 26 shows a simplified diagram for a turbine generator. The circuit shows the design of a circuit board which can generate electricity.

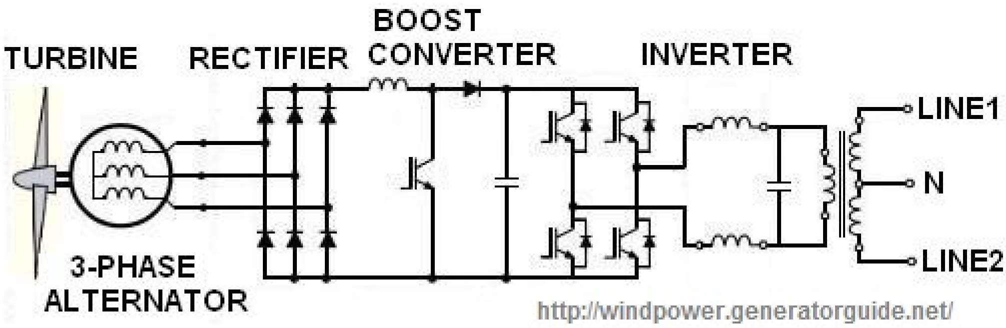


Figure 26 Turbine Generator diagram (Rozenblat, 2009-2014)

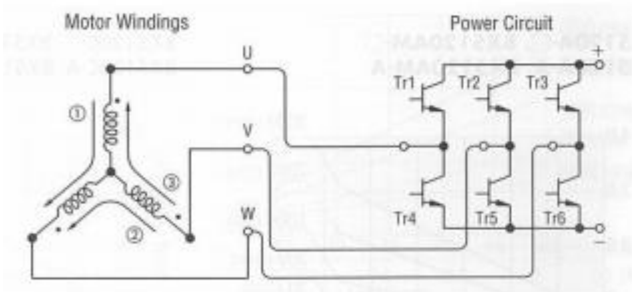


Figure 27 ESC for a 3-phase motor diagram (Shirahata)

Figure 27 shows a simplified ESC for a three-phase electric motor. The ESC regulates the voltage supplied from the battery or batteries to the motor to regulate the power, torque and speed. Both the circuit boards on their own are simple designs, however combining both these circuit boards together can complicate the workings behind the project. It is possible to combine both circuits to allow the turbocharger to accelerate up electrically and then when the turbocharger is at high speeds the electric motor is not in operation. The turbocharger can become an electrical turbine and draw electrical energy through the rotational velocity of the turbocharger by the electrical motor.

Looking at the designs of the circuits separately does not impose problems, combining both the circuits does. A control device such as an ECU can operate the working conditions of the combined circuit board, however this requires complicated code to be designed, as well as sensors to pick up on the electrical signals of the motor when the motor is not in operation. This is not within the scope of the project and requires large amounts of detailed work and research to make a system to control the ETC turbocharger.

What is the difference between electric motors and generators

Electrical energy can be transferred from any mechanical object that has rotating or moving parts, commonly known as a generator. On the other hand, an electric motor can be seen as a hydraulic pump as it needs energy to move an object. Therefore, one makes electrical energy from moving objects and the other uses electricity to move an object.
Despite the difference between electric motors and generators they are closely connected through Faraday's law of electromagnetic induction. Faraday's law of electromagnetic induction is the physics of how a magnetic field interacts with an electrical circuit, aiming to produce electromotive force also known as EMF. This law of EMF allows for mechanical energy to be translated into electrical energy, as well as to create both motors and generators.
The law of induction proves that when a *"magnetic field across a conductor, such as a wire coil, electrons are forced to move perpendicular to that magnetic field. This generates electromotive force, which creates a flow of electrons in one direction. This phenomenon can be used to produce electricity in an electric generator"* (Manney, 2016).
Although there is a difference between generators and motors the actual mechanisms themselves are the same. The control units, that control the motors and generators, are the primary difference between them. Motors can be used as generators and vice versa. Understanding the differences and similarities is key, however the problems encountered is high speeds.

AC and DC motors

There are two categories of motors Alternating Current motors (AC motors) and Direct Current motors (DC motors). Under these two categories there are several different motors.
Brushed motors have four main components a stator, rotor, brushes and a commutator. Like a brushless motor the brushed motor also has permanent magnets on the outside. Copper windings are fixed to the motor shaft on the stator or armature. When current is applied, the motor turns but power is supplied through the brushes which are directly attached to the commutator. As power is supplied through the brushes it energises the commutator to spin the motor shaft.

However, as the brushes are directly attached to the commutator and supply electricity to turn the commutator, there is a large amount of friction created and this can also create sparks which in turn reduce the motors life as there is wear and tear. This means that the brushed electric motors are not very reliable and do not have as long a life as brushless motors.

Brushless motors give higher power and higher efficiency than their counterpart. These motors also allow longer running times and greater speeds compared to a brushed system the same size. As brushless motors have no brushes acting upon them, they do not require a physical commutator; this means that the system need very little maintenance.

Brushless motors also have three wires as there are three electromagnet phases inside the motor. When this electromagnet energises, the opposite polarities attract to the permanent magnets, on the outer face of the motor. As they reach closer to the magnets, the next phase is energised and so on, spinning the electric motor. As there is no contact between the electromagnets and the magnets there is no friction allowing the motor to spin faster than a conventional brushed motor (ThinkRC, 2017).

In AC motors, current is applied to the stator which is stationery. The magnets are connected to the rotor and will rotate. As the current varies in the stator it moves the magnetic field causing the rotor to rotate. AC motors have several variations to create a magnetic field around the rotor, some AC motors have an "electromagnets drive" which is powered by a DC power source (Moyer & Chicago, 2010).

By understanding the differences in motors, we have a simple choice of which motor to use. The project will use a brushless DC motor for their capabilities of high speed and minimal maintenance.

Motor capabilities

Most modern turbochargers have great performances with little losses. This makes turbochargers great but for a compressor to work well they need to spin up to high speeds, which electric motors are not capable of withstanding. Turbochargers idle at 40 krpm; the turbocharger used in this project starts to produce boost pressure at 70 krpm. At this speed the magnets within an electric motor will start to break and chip off; this can cause catastrophic damage to the turbocharger.

Can there be a solution? Boeing manufacture large jet engines which correspond with British standards and safety legislations. During their jet engine tests, they destroy one of the blades within a jet engine with a controlled explosive. This allows them to test their sleeved body around the jet engine which is made up of Kevlar or another composite material that can handle large forced impact. The blade hits the Kevlar liner within the engine at high speeds with a tremendous amount of force damaging the jet engine but containing the impact.

Therefore, this solution can suggest a design implement. As electric motor magnets cannot handle high speeds when attached to a turbocharger, they can be banded with Kevlar or carbon composites to stop the magnets within motors from breaking down being destroying.

Chapter 4 ECU tuning

An Electronic Control Unit also known as an ECU is a computer which controls the engine's parameters. ECU's need sensors to operate; this can be seen in table 1. An ECU can make changes and compensate for several factors, for instance if fuel is bad or if there is a lack of air, the ECU can compensate for the conditions that the engine is enduring and make correctional factors to change the engines fuel and ignition maps to keep the engine running at optimal. The ECU can only do this by being rigged up with sensors which can indicate difference due to air, temperature etc…

Sensor category	Sensor type
Position sensors	Crank angle, cam angle, throttle position
Exhaust gas composition sensors	Narrowband and wideband sensors
Temperature sensors	Exhaust gas, air, coolant, oil
Pressure sensors	Manifold absolute pressure, barometric pressure, fuel pressure, oil pressure
Air-meter sensors	Mass airflow, vane airflow
Knock sensors	Knock

Table 1 ECU Sensors

To recap simply engine's need fuel and spark to operate. Fuel is supplied by a modern method of fuel injectors and fuel pumps and spark is given by sparkplugs and coil packs.

The engine used in this project is a 471cc inline two-cylinder engine. Modern engines use a crank position sensor; this sensor determines the Top Dead Centre also known as TDC. It measures this in degrees and as the crank rotates, sends a signal to the coil packs to ignite the air and fuel mixtures in the cylinders.

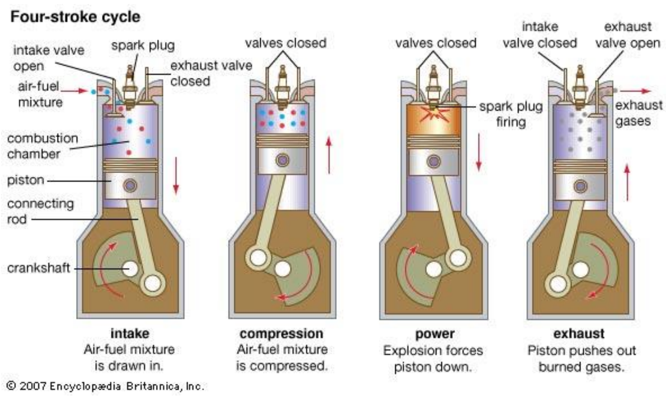


Figure 28 the four-stroke engine cycle (Proctor & Cromer, n.d.)

Figure 28 shows the four-strokes in a four-stroke engine cycle; these four-strokes are intake, compression, power and exhaust. These four-strokes happen over 720° with power being the combustion stroke. An engine will send a spark to ignite fuel before TDC; this spark signal is usually mapped into an engine's ECU. If the intake cycle is 180°. Therefore, the other three-strokes are 180° as well. So, if 540° is TDC usually the ignition spark will usually retard a few degrees before sending spark. This is called ignition timing.

Fuel & Ignition Maps

Calibration is the optimisation of the map. Tuning/calibrating the fuel and ignition maps helps to improve the efficiency of an engine. To produce the best possible performance for the given engine. In addition, it can also improve fuel consumption and emissions control.

Fuel Map

Throttle % -> RPM	0	4	7	11	16	22	29	38	47	57	66	76	85	95
500	3.30	3.94	3.94	5.01	5.18	5.40	5.55	5.47	5.50	5.37	5.74	5.44	5.19	5.39
1000	2.24	3.00	3.60	3.60	4.80	5.24	5.53	5.70	5.77	5.79	5.79	5.03	5.76	5.79
2000	2.23	3.00	3.60	3.60	4.80	5.24	5.53	5.70	5.77	5.79	5.79	5.03	5.76	5.79
3000	2.00	3.00	3.60	3.60	4.80	5.24	5.53	5.70	5.77	5.79	5.79	5.03	5.76	5.79
4000	2.00	3.00	3.60	3.60	4.80	5.24	5.53	5.70	5.77	5.79	5.79	5.03	5.76	5.79
4500	2.00	3.00	3.60	3.66	4.86	5.39	5.78	6.02	6.17	6.22	6.24	5.73	6.30	6.29
5000	2.00	3.00	2.98	3.40	4.62	5.07	5.41	5.74	5.85	5.97	5.98	5.99	6.05	6.04
6000	2.00	3.00	2.46	3.08	3.88	4.40	4.83	5.11	5.26	5.36	5.38	5.41	5.64	5.40
6500	2.00	3.00	2.84	3.25	4.19	4.68	5.06	5.31	5.43	5.50	5.52	5.28	5.68	5.53
7000	2.00	3.00	3.22	3.43	4.49	4.96	5.30	5.50	5.60	5.65	5.65	5.16	5.72	5.66
7500	2.00	3.00	3.41	3.04	4.22	4.76	5.20	5.53	5.73	5.80	5.84	5.53	5.92	5.89
8000	2.00	2.40	2.40	2.64	3.95	4.56	5.09	5.56	5.85	5.94	6.03	5.89	6.11	6.11
8500	2.00	2.40	2.40	2.68	3.87	4.63	5.26	5.80	6.22	6.31	6.38	6.13	6.42	6.42
9000	2.00	2.40	2.40	2.72	3.79	4.69	5.43	6.04	6.58	6.67	6.72	6.36	6.72	6.72
9500	2.00	2.40	2.40	2.72	3.42	4.65	5.36	6.05	6.58	6.75	6.81	6.61	6.89	6.72
10000	2.00	2.40	2.40	2.72	3.05	4.61	5.28	6.05	6.58	6.83	6.90	6.85	7.06	6.72
10500	2.00	2.40	2.40	2.72	3.79	4.69	5.43	6.04	6.58	6.67	6.72	6.85	6.72	6.72
11000	2.00	2.40	2.40	2.72	3.79	4.69	5.43	6.04	6.58	6.67	6.72	6.85	6.72	7.08
12000	2.00	2.40	2.40	2.72	3.79	4.69	5.43	6.04	6.58	6.67	6.72	6.85	6.72	7.08
13000	2.00	2.40	2.40	2.72	3.79	4.69	5.43	6.04	6.58	6.67	6.72	6.85	6.72	7.08

Figure 29 Fuel Map

Figure 29 is the Fuel map the columns are throttle position percentage, the rows are engine speed in RPM and the values in the map are in milliseconds. This is the time that the injectors are held open and is called the injector pulse width.

The finished map should be smooth without any spikes in it. As for the Air to Fuel ratios (AFRs) that an engine needs at specific load and speed points, there are some approximate guidelines but every engine differs from another. The guidelines for a naturally aspirated petrol engine are shown in table 2.

OPERATING CONDITION	AFR
Idle	14.7:1
Max torque	13.4:1
Max power	12.4:1

Table 2 AFR for different operating conditions N/A

AFRs are very important when engines are tuned for optimal performance. Most commonly a wideband AFR system is used to calibrate the fuel map; a wideband measures the oxygen content left after combustion in an exhaust system. From the leftover oxygen content after combustion the wideband determines the AFRs after some calculations. For instance, if the vehicle runs an AFR value of 12:1, it means that the engine is running rich and the oxygen content in the exhaust system will be very low; therefore the wideband will make its calculations and determine the exact air to fuel ratios.

A turbocharged AFRs are different and simpler, this is shown in table 3. In a turbocharged platform engine idle and low engine speeds without boosted air pressure the AFRs can be around 14.7. Furthermore, under boost pressure load the AFRs have to be at a 12:1.

OPERATING CONDITION	AFR
Idle	14.7:1
On Boost Pressure	12:1

Table 3 AFR for different operating conditions turbocharged

Ignition map

Throttle % -> RPM	0	4	7	11	16	22	29	38	47	57	66	76	85	95
500	20.0	25.0	20.0	25.0	25.0	17.1	17.1	17.1	17.1	17.1	17.1	17.1	17.1	17.1
1000	20.0	25.0	30.0	30.0	30.0	30.0	30.0	30.0	30.0	30.0	30.0	30.0	30.0	30.0
2000	20.0	25.0	30.0	30.0	30.0	30.0	30.0	30.0	30.0	30.0	30.0	30.0	30.0	30.0
3000	20.0	25.0	30.0	30.0	30.0	30.0	30.0	30.0	30.0	30.0	30.0	30.0	30.0	30.0
4000	20.0	25.0	30.0	30.0	30.0	30.0	30.0	30.0	30.0	30.0	30.0	30.0	30.0	30.0
4500	40.0	35.0	35.0	39.7	37.2	34.2	35.0	34.2	35.0	35.0	34.8	35.0	34.8	34.6
5000	40.0	35.0	56.2	56.7	40.3	35.9	35.0	34.8	33.4	33.0	33.0	32.8	32.8	32.9
6000	40.0	35.0	56.0	55.9	52.7	51.7	45.2	44.8	42.1	40.9	40.4	37.3	37.2	37.4
6500	40.0	35.0	57.9	57.9	55.4	53.6	51.3	49.9	46.1	44.8	44.5	43.8	43.1	43.3
7000	60.0	60.0	60.0	59.8	58.0	55.5	57.4	54.9	50.0	48.7	48.5	50.3	49.0	49.2
7500	60.0	60.0	60.0	59.8	55.7	52.4	51.2	49.5	48.0	47.3	47.5	48.4	47.5	47.7
8000	60.0	60.0	60.0	59.8	53.4	49.3	44.9	44.0	45.9	45.9	46.4	46.4	45.9	46.2
8500	60.0	60.0	60.0	59.8	50.7	47.5	45.3	43.6	43.8	43.8	44.0	44.2	43.8	43.9
9000	60.0	60.0	60.0	59.8	47.9	45.7	45.6	43.1	41.6	41.7	41.6	41.9	41.6	41.5
9500	60.0	60.0	60.0	59.8	48.4	47.4	47.4	45.1	44.3	43.1	42.9	43.7	43.4	43.3
10000	60.0	60.0	60.0	59.8	48.8	49.1	49.1	47.1	47.0	44.4	44.2	45.5	45.2	45.1
10500	60.0	60.0	60.0	59.8	47.9	45.7	45.6	44.0	44.0	44.0	44.0	44.0	44.0	44.0
11000	60.0	60.0	60.0	59.8	47.9	45.7	45.6	44.0	44.0	44.0	44.0	44.0	44.0	44.2
12000	60.0	60.0	60.0	59.8	47.9	45.7	45.6	44.0	44.0	44.0	44.0	44.0	44.0	44.2
13000	60.0	60.0	60.0	59.8	47.9	45.7	45.6	44.0	44.0	44.0	44.0	44.0	44.0	44.2

Figure 30 ignition map

Figure 30 shows the ignition map. The columns and rows are the same as the fuel map and the values in the map are degrees Before Top Dead Centre (BTDC). BTDC is when the ECU demands the spark plug to fire.

Ignition timing is mostly tuned through the use of a dynamometer, the engine is held at a specific cell within its ignition map, the timing is advanced or retarded until the maximum point of torque is reached. The maximum torque will be seen on the dynamometer’s screen.

Mapping sensors

Several different types of sensors can be used to map the individual loads calibrating both the fuel and ignition map. The Manifold Absolute Pressure (MAP) sensor can be used; this sensor picks up the amount of pressure built in the intake manifold. Throttle Position Sensor (TPS) could also be used; this sensor picks up the angle at which the throttle opens. When engines are on a dynamometer for tuning, the dynamo is held at certain loads and individual cells are tuned for the fuel and ignition maps.

It is possible to tune an engine on either of these sensors. However, with a turbocharged motor bike engine, tuning is a lot more difficult as it is harder to hold the engines under specific loads therefore a MAP sensor will be constantly changing, making tuning difficult. So, a TPS senor will be more reliable as the throttle could be held open at a constant value and not change therefore making the tuning simpler. However, both sensors will be installed to make controlling and regulating the turbochargers boost pressure easier.

Closed loop

Closed loop is a perimeter which needs a wideband sensor in the ECUs parameters for target AFRs. Figure 31 shows the target AFRs, the parameters work in the same way as the fuel and ignition maps. However, the parameters are filled with target air to fuel ratios, with 14.7 being the stoichiometric air to fuel ratio and 13.23 being a richer mixture.

Target A/F

Throttle % -> RPM	0	4	7	11	16	22	29	38	47	57	66	76	85	95
500	14.700	14.700	14.700	14.700	13.230	13.230	13.230	13.230	13.230	13.230	13.230	13.230	13.230	13.230
1000	14.700	14.700	14.700	14.700	13.230	13.230	13.230	13.230	13.230	13.230	13.230	13.230	13.230	13.230
2000	14.700	14.700	14.700	14.700	13.230	13.230	13.230	13.230	13.230	13.230	13.230	13.230	13.230	13.230
3000	14.700	14.700	14.700	14.700	13.230	13.230	13.230	13.230	13.230	13.230	13.230	13.230	13.230	13.230
4000	14.700	14.700	14.700	14.700	13.230	13.230	13.230	13.230	13.230	13.230	13.230	13.230	13.230	13.230
4500	14.700	14.700	14.700	14.700	13.230	13.230	13.230	13.230	13.230	13.230	13.230	13.230	13.230	13.230
5000	14.700	14.700	14.700	14.700	13.230	13.230	13.230	13.230	13.230	13.230	13.230	13.230	13.230	13.230
6000	14.700	14.700	14.700	14.700	13.230	13.230	13.230	13.230	13.230	13.230	13.230	13.230	13.230	13.230
6500	14.700	14.700	14.700	14.700	13.230	13.230	13.230	13.230	13.230	13.230	13.230	13.230	13.230	13.230
7000	14.700	14.700	14.700	14.700	13.230	13.230	13.230	13.230	13.230	13.230	13.230	13.230	13.230	13.230
7500	14.700	14.700	14.700	14.700	13.230	13.230	13.230	13.230	13.230	13.230	13.230	13.230	13.230	13.230
8000	14.700	14.700	14.700	14.700	13.230	13.230	13.230	13.230	13.230	13.230	13.230	13.230	13.230	13.230
8500	14.700	14.700	14.700	14.700	13.230	13.230	13.230	13.230	13.230	13.230	13.230	13.230	13.230	13.230
9000	14.700	14.700	14.700	14.700	13.230	13.230	13.230	13.230	13.230	13.230	13.230	13.230	13.230	13.230
9500	14.700	14.700	14.700	14.700	13.230	13.230	13.230	13.230	13.230	13.230	13.230	13.230	13.230	13.230
10000	14.700	14.700	14.700	14.700	13.230	13.230	13.230	13.230	13.230	13.230	13.230	13.230	13.230	13.230
10500	14.700	14.700	14.700	14.700	13.230	13.230	13.230	13.230	13.230	13.230	13.230	13.230	13.230	13.230
11000	14.700	14.700	14.700	14.700	13.230	13.230	13.230	13.230	13.230	13.230	13.230	13.230	13.230	13.230
12000	14.700	14.700	14.700	14.700	13.230	13.230	13.230	13.230	13.230	13.230	13.230	13.230	13.230	13.230
13000	14.700	14.700	14.700	14.700	13.230	13.230	13.230	13.230	13.230	13.230	13.230	13.230	13.230	13.230

Figure 31 closed loop target AFR

Furthermore, when the close loop setting is activated the ECU works alongside the wideband sensor to over-ride the fuel and ignitions maps. For instance, at wide open throttle at 13 krpm the target AFR is 13.23. If the wideband picks up that the actual AFR is below or above the target number, it will bring in correctional factors to alter the fuel and ignition maps and maintain a 13.23 target AFR. Mostly the close loop function in the ECU is used to fine tune engines and maintain a flawless AFR when on open loop.

'Boost'

Forced induction machinery such as turbochargers or superchargers make 'Boost;' this is the term used for a turbocharger or supercharged vehicle. Boost pressure is measured in standard SI units (bar).

This project is not looking to produce high amounts of boost pressure; this is because this project looks into a proof of concept of a turbocharger and not performance results of high performing engines. For this reason, the target boost pressure will be around 0.5bar of boost pressure.

Boost pressure is usually controlled via a wastegate actuator on turbochargers, a wastegate is effectively a spring which regulates the amount of exhaust gas flow around the turbocharger's turbine, a pipe which connects from the compressors housing leads into the wastegate housing. The boost pressure acts on the spring. When the target pressure is met, the wastegate actuator will open restricting the amount of gas flow therefore restricting the boost pressure to a desired value.

Boost pressure can be controlled by changing wastegate springs or using a manual or electronic boost controller. These controllers regulate the pressure of the compressor to permit higher boost pressures. However, higher boost pressures alters the AFRs therefore when in the mapping process, the fuel map will need to be constantly altered as more air pressure is added.

3-way boost solenoid

A 3-way boost solenoid is an electronic boost controller which can be integrated into the engine wiring loom and therefore the boost pressure can be controlled by the ECU. A 3-way boost controller is one of the most effective ways to control boost pressure, Figure 32 shows two diagrams A and B. Diagram A shows the 3-way boost solenoid regulating the boost pressure; it does this by connecting the line from the turbocharger directly to the wastegate actuator shown here by the solid blue line. Furthermore, the system can produce higher amounts of boost pressure by venting the line to the wastegate then directly to the atmosphere. This then produces higher boost pressure as the wastegate is being tricked into believing that the turbo is making a bar of atmospheric pressure.

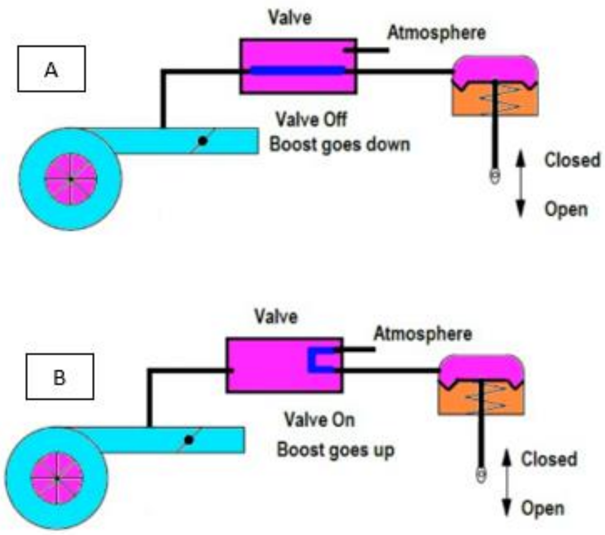


Figure 32 3-way boost solenoid diagram

Turbocharging the petrol engine

Most commonly turbocharging was associated with diesel engines this is due to maximising diesel fuel efficiency. With diesel engines the air is drawn into the cylinders, then compressed. Fuel is only added when the air is compressed, until the required amount of fuel is added the mixture self-ignites. This is because the compressed air temperature is high. However as discussed previously petrol engines work differently; they draw in fuel and air at the same time into the cylinders or cylinder.

Furthermore, self-ignition or pre-detonation can occur with petrol engines; this can cause all sorts of problems. As the temperature rises within the cylinders this can cause pistons to melt or seize the engine. Therefore, modern turbocharged petrol engines have low compression ratios around 8 to 10:1. This low compression ratio allows for the temperature within the engine to be controlled allowing for the petrol and air mixture to keep below its self-igniting temperature.

Knock occurs when the spark for ignition does not occur at the optimal point and the air and fuel mixture can reach its self-detonation temperature. The self-detonation occurs due to the compression of the air and fuel mixture. As gasses and liquids are compressed, they become less dense therefore meaning they increase in temperature.

There are several ways to indicate 'knock.' Firstly, a knock sensor can be installed to the engine; the knock can be indicated by this sensor. Secondly, retarding the engine's ignition timing to not be as aggressive. Lastly reducing the turbocharger's boost pressure; this is because the air pressure has increased from the turbocharger. As temperature rises from compressing the air, this air will be slightly hotter than a lower boost pressure setting even after intercooling. This hotter air can increase the fuel's temperature in the inlet port to self-igniting point. These three measures can control knock and minimise engine damage (Stone, Introduction to Internal Combustion Engines, 2012).

Turbocharger Lag

Turbocharged petrol engines are a lot more 'laggy' compared to a diesel engine. This is due to certain features of petrol engines. Petrol engines are usually lighter and smoother than diesel engines, so lighter flywheels are used in petrol engines to make them more responsive however, this delays the turbocharger to produce the air supply to the engine more intrusive (Stone, Introduction to Internal Combustion Engines, 2012).

Secondly petrol engines operate in a much wider mass flow range this is from idle to full-speed un-throttled. From this understanding the air flow required can rapidly change resulting in turbocharger lag. This lag is usually felt by the driver of the vehicle, the Brabham BT53 Formula One car was phenomenal for turbo-lag. The 1.5l engine produced 671kW at the wheels. However, drivers complained that the vehicle after corners did not accelerate well and was 'laggy' (Stone, Introduction to Internal Combustion Engines, 2012).

Turbo sizing

Calculating turbocharger size is critical, this is because a 471cc engine is very small in displacement compared with most modern turbocharged vehicles which are much larger in displacement.

The definition of size in turbocharger terms means the turbocharger supports the engines application. The application that the engine will be used in is for racing therefore a sufficiently sized turbocharger which produces boost at a low engine speed and maintains boost throughout the powerband is required.

Nevertheless, other applications such as drag racing vehicles require, turbochargers that are larger compared to their engine's displacement. As the engines are mostly in the top powerband and in much higher engine rpm ranges the engine can produce enough exhaust gas flow to support their application. Furthermore, drag racing vehicles have a very peaky power band which is mostly at the top range of their powerband therefore a drag racing vehicle which is racing on a track would be hopeless due to the large amount of turbo-lag.

Assumptions

This project's target is to make 65 kW (80 HP) at the engines crank. The engineering assumption is as a 99% volumetric efficiency (VE). The VE was assumed high; this is because motorbike engines such as the CB500 use individual throttle bodies and this allows for a larger volume of air to enter with little resistance from the intake valves.

The Brake Specific Fuel Consumption (BSFC) was set low this is because the assumptions of air that enters the turbocharger after the restrictor at high engine rpm will be constant due to the restrictor's airflow reaching the Mach number of one, effectively choking. Therefore, only a certain amount of fuel will be added to the engine to reach the correct air to fuel ratios.

Intake manifold temperature was set to 115℉ which is equivalent to 46°c. This assumption was made due to the study shown under the subheading of 'cooling efficiency.'

Calculations

Target horse power = 80 hp

Engine displacement = 471cc

Maximum RPM = 10000rpm

Engine volumetric efficiency = 99%

Intake manifold temperature = 115℉

Brake specific fuel consumption = 0.5

Wa= Airflow actual (lb/min)

HP = Horsepower Target

$\frac{A}{F}$ = Air/Fuel Ratio

BSFC = Brake Specific Fuel Consumption

$$Wa = Hp \times \frac{A}{F} \times \frac{BSFC}{60} \qquad [1]$$

$$Wa = 80 \times 14.7 \times \frac{0.5}{60} = 9.8\ \frac{IB}{min} \quad [2]$$

Wa = Airflow actual (lb/min)

R = Gas Constant

Tm = Intake Manifold Temperature (degrees F)

VE = Volumetric Efficiency

N = Engine speed (RPM)

Vd = Engine displacement (Cubic Inches)

$$\text{Manifold pressure } = \frac{Wa \times R \times (460+Tm)}{Ve \times \frac{N}{2} \times Vd} \qquad [3]$$

$$\frac{9.8 \times 639.6 \times (460+115)}{0.99 \times \frac{10000}{2} \times 28.74} = 25.3\ psia \qquad [4]$$

The next step will be to determine the pressure drop in the system, this is difficult to calculate. Though an assumption from 0-10 can be made, the assumption of 1psi was assumed this was because the length from the turbocharger outlet pipe to the intake will be very short therefore a minimal pressure drop will be seen throughout the system. Consequently, P2c is the discharge pressure of the compressor.

P2c = Manifold pressure + ΔP_{loss} = 25.3 + 1 = 26.3 psia

P1c takes the ambient air pressure of 14.7 psi (see level) and subtracts the loss in ambient pressure of 1 psi this is because the air filter and the 20mm restrictor come into play. These restrictors theoretically have some impact therefore a very minimal amount of pressure is subtracted.

P1c = $P_{amb} - \Delta P_{loss}$ = 14.7 - 1 = 13.7 psia

Now the pressure ratio can be calculated:

Pressure ratio = $\frac{P2c}{P1c} = \frac{26.3}{13.7} = 1.9$ [5]

From the calculated results with a presser ratio of 1.9 and correct air flow of $9.8\frac{IB}{min}$.

The calculations were not done in imperial units this is because most turbocharger maps are in imperial units (Garrett, 2018).

Turbocharger chosen

The Garret GT2056 was selected as the turbocharger for this project. Figure 33 shows the turbocharger's map; the red dot indicates the calculations shown above fitted onto the map. Therefore, indicating that the turbocharger will be working in the 70 - 71% efficiency island and also line close to the surge line.

Surge is when the compressor 'stalls' - either the flow of air through the compressor stops or reverses. This does not mean that the turbocharger stops working at all 'stall' it's an aerodynamic term which means that the air entering the compressor is not being compressed but chopping at the compressor wheel. The resulting compressed air before the throttle body is also pushed back into the turbochargers discharge inlet.

Surge under load is the most dangerous, this is because it can damage the turbochargers bearing from rattle but under this condition means that the compressor size is large for the application. The second surge can occur when the throttle is caused immediately after high boost pressure is made. However, surge can be overcome by adding a bypass valve or a blow of valve. This valve opens when the pressure in the plenum is not equal to the pressure in the turbocharger piping allowing some boosted air pressure to be vented into atmosphere to prevent the turbocharger from surging after the throttle is closed (Surge Line, 2018).

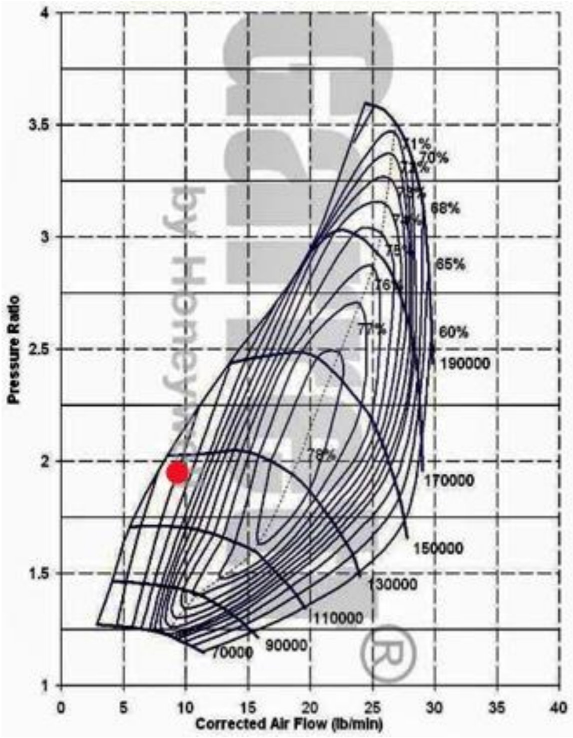


Figure 33 compressor map of specific turbocharger with red dot indicating fitment

Chapter 5 Mathematical modelling

Overview

The modelling of the initial Naturally Aspirated (N/A) engine was carried out previously by (Latif, Raja , 2018) as part of his undergraduate degree. This model was then used as the basis for further development incorporating turbocharging and electrically assisted turbocharging.

Ricardo wave is a 1-D engine simulation software; this software was used to validate the dynamometer turbocharged Honda CB500x engine. Firstly, the OEM engine would have to be created and validated with the Honda results, then the engine can be turbocharged in Ricardo following the Formula Student rules and the results of the turbocharged engine can be verified with the dynamometer results.
Specifications of the engine firstly were gathered shown in table 4, these specifications are to be placed into Ricardo Wave but items such as the catalytic converter and air box were omitted; this is due to the software not being able to support a catalytic converter. The air box was also omitted due to a complex design which could not be imprinted into Wave.

Name	specification
Displacement	471cc
Pistons	2
Firing order	1-2
Bore and stroke	67 x 66.8
Stroke	4
Intake valve lash	0.2mm
Exhaust valve lash	0.25mm
Piston clearance	0.8mm
Piston pin offset	0.8mm
Completion ratio	10.7
Firing order	2-1
Firing intervals	180°
Power	47 Bhp 35kW
Torque	50 Nm
Throttle body	2x 38mm
Intake camshaft profile	See page 51-53

Exhaust camshaft profile	See page 51-53
Intake camshaft max lift	7.3mm
Exhaust camshaft max lift	6.8mm
Intake valve diameter	25mm
Exhaust valve diameter	21.5mm
Air to fuel ratio	See page 57
Intake port length	120mm
Exhaust port length	60mm
Intake port diameter	28mm
Exhaust port diameter	21.5mm
Intake port angle	60°
Exhaust port angle	90°
Combustion model	Unknown

Table 4 gathered engine specifications for simulation (Latif, Raja , 2018)

Valve Lift Profile CB500X

For the simulation, the software needed camshaft profile and camshaft lift to simulate the engine. Therefore, a DTI gauge was set onto the valves of the exhaust and intake; the DTI gauge was not placed onto the camshaft itself as the Honda CB500x engine has a rocker which is effectively a multiplier. The rocker itself has a ratio for example, 2:1 and this means the camshaft can have a smaller lift for example, 3.5mm. This means that the lift of the valves will be 7mm as the rocker multiples the cam lift by the 2:1 ratio.

To keep the simulation simple and effective and in order to bypass the rocker and the camshaft, the DTI gauge will be placed onto the vales. This will then measure the valve lift and the rocker ratio and give the camshaft and rocker lift and angles as if the camshaft was directly positioned onto the valves. This is shown below on Figure 34.

Figure 34 DTI gauge on intake valve (Latif, Raja , 2018)

Firstly, top dead centre (TDC) of the engine was indicated by placing a piston stop into the sparkplug hole and marking the TDC position. Then a crank angle instrument was positioned onto the flywheel and an indicator was placed onto the crank angle instrument which showed the degree of rotation. This can be seen in Figure 35.

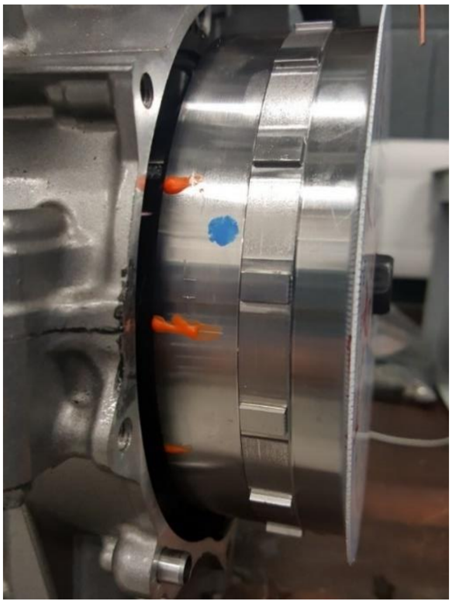


Figure 35 aligning engine to TDC before measuring lift profiles (Latif, Raja , 2018)

Secondly the DTI gauge was placed onto the intake and exhaust valve; the crank was turned every 2 degrees and the results were taken down from the DTI gauge and the results were plotted.

Results

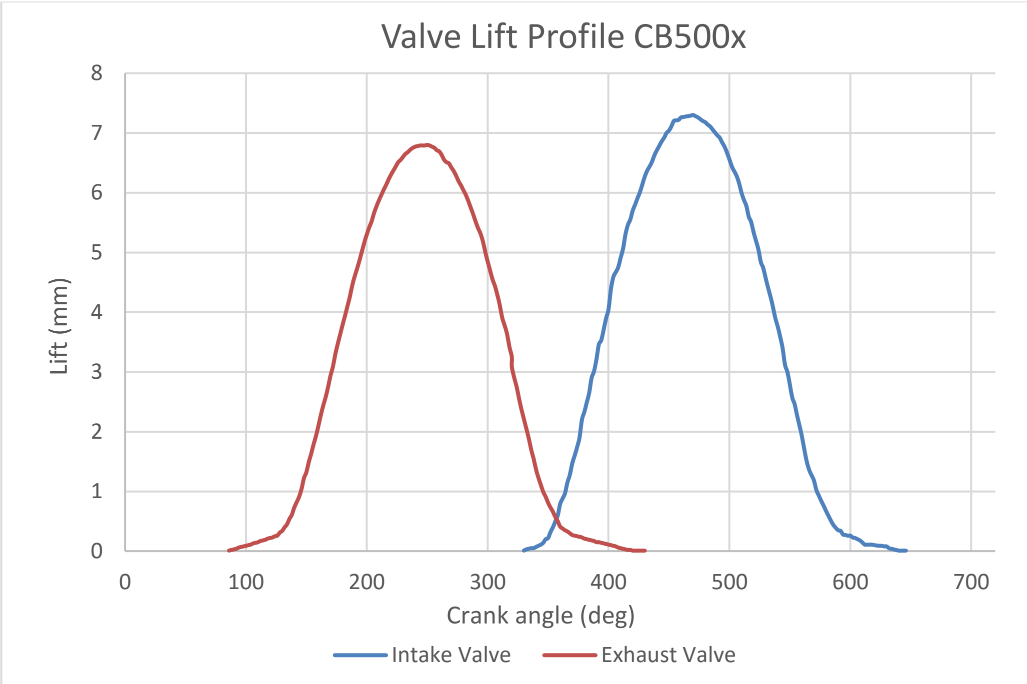


Figure 36 intake and exhaust lift profiles (Latif & Allport, 2018)

Figure 36 shows the camshaft profiles of both the intake and exhaust valves bypassing the rocker. The orange curve indicates the exhaust valve showing maximum lift is 6.8mm. The blue curve indicates the intake valve with maximum lift at 7.3mm (Latif & Allport, 2018). These results were verified by Hondas manual as both set of results are correct. The graph goes from 0° to 720°, this is because the Honda is a four-stroke engine, so between 0° and 360° the cycle 'Expansion' and 'Exhaust' will occur and between 360° and 720° 'Intake' and 'Compression' will occur completing the four-stroke cycle.

The blue dotted line shown is the intake camshaft profile, therefore the orange dotted line is the exhaust camshaft profile.

What is Overlap?

The results indicate that the engine valves are both open at around 330° to 430°; this region is called 'overlap.' This is before the piston reaches TDC on the exhaust stroke. This overlap is designed into the engine so that when the exhaust gasses are leaving the engine induction air is drawn in to clear out the cylinder of combustion gasses. If overlap was not designed into engines, exhaust gasses would mix and dilute induction air and fuel which would effectively decrease efficiency. Figure 37 clearly illustrates the overlap effect of a four-stroke cycle engine.

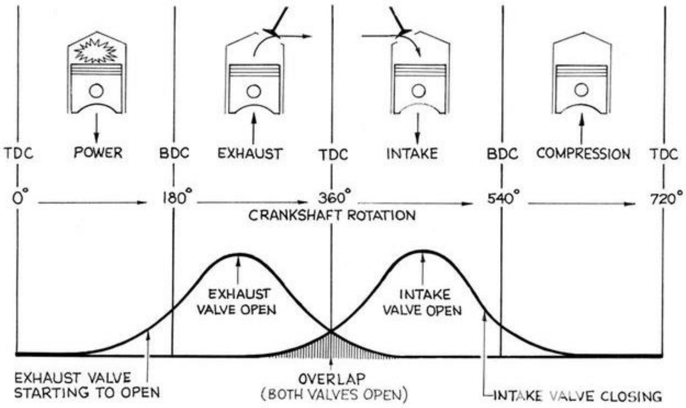


Figure 37 overlap and the four-stroke engine diagram (Davis, 1998)

Overlap on a turbocharged vehicle should be as small as possible; the CB500x has 100° of overlap if the vehicle is used for racing purposes this angle should be reduced to 55-60° (Latif R. , 2018). Due to the boost pressure escaping into the exhaust and not being concealed into the cylinders, effectively the turbocharger will have to work slightly harder on the CB500x to produce slightly more boost pressure.

Ricardo Modelling

To begin the Ricardo Wave model, the intake and exhaust camshaft models were applied, so that the model can start implementing other data such as engine parameters.

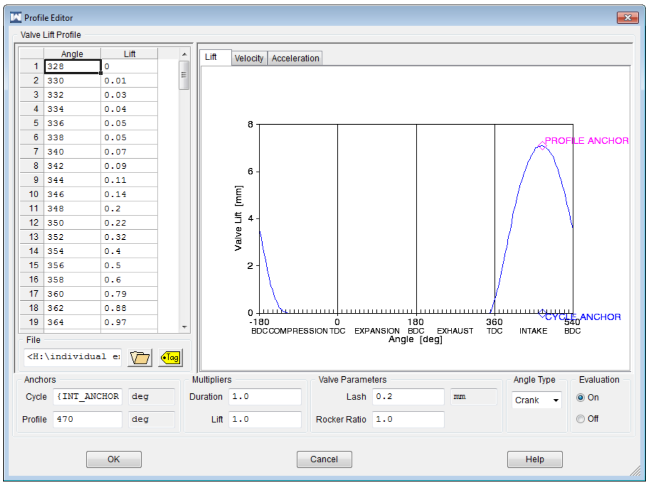


Figure 38 camshaft profile in Ricardo Wave

Figure 38 shows the general model of the intake camshaft profile from the DTI gauge placed into Ricardo Wave. The rocker ratio was left at 1 as this means that there is no rocker. There are rockers inside the CB500x engine but as the DTI guage was used to bypass the rocker and measure the valve lift profile. This can effectively give the camshaft profiles of the engine.

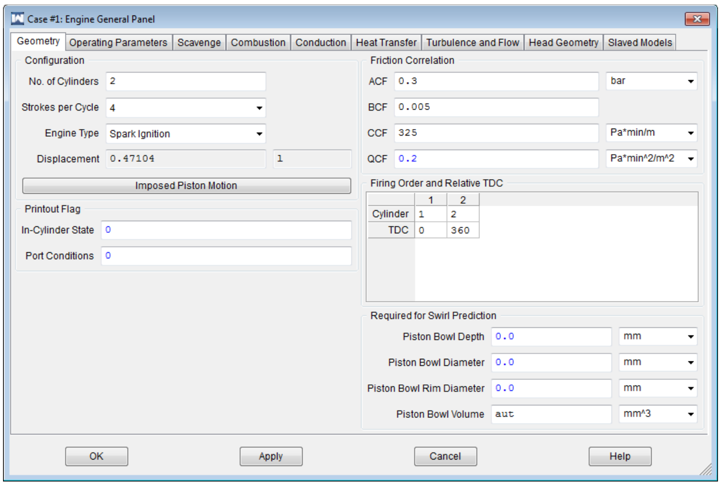


Figure 39 Ricardo Wave engine parameters

Figure 39 shows the engine parameters added to the model. However, as the frictional coefficient of the engine is a closely guarded secret with Honda the values were taken from another motorbike engine within Ricardo wave engine models.

Values for the scavenging, conduction and heat transfer were from Ricardo Waves engine models. This was to keep the simulation as simple as it can be without overcomplicating the model and researching for impossible values.

Combustion Model

The Combustion model detail is unknown for the engine as this is a close guarded secret with Honda, but a standard simulated model can be applied to the simulation. The combustion model reflects the ignition map of an engines ECU. On the properties panel the "location of 50% burn point" and "combustion duration (10%-90%)" were given general parameters nonetheless for the initial simulation this can give a good indicated model and can design a general but sufficient combustion model. Figure 40 shows the combustion model.

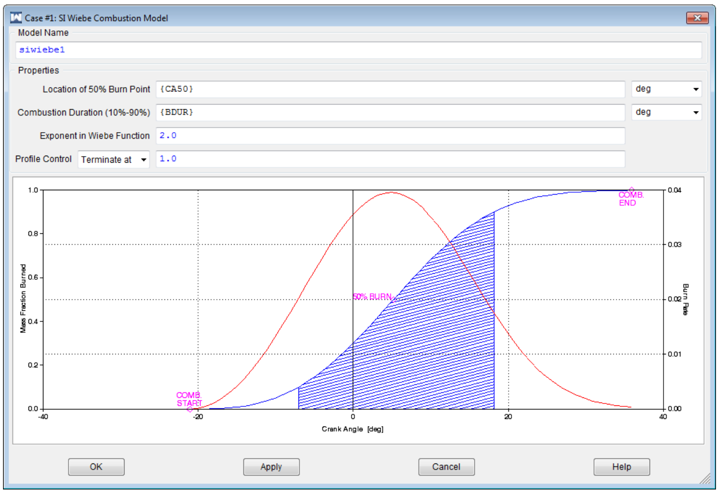
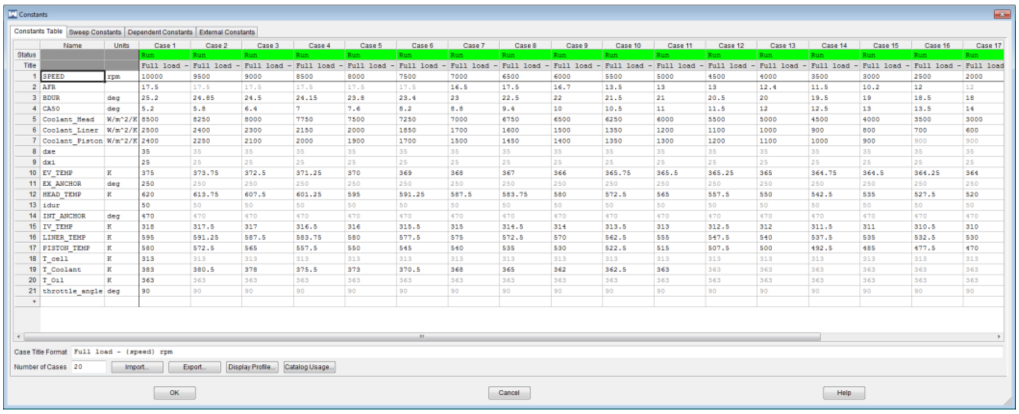


Figure 40 Combustion model in Ricardo Wave

Engine Parameters

The Engine parameters can be placed into a Ricardo Wave spread sheet shown in Figure 41. The constant "speed" indicates the RPM range for the model going up in steps of 500RPM to 10000RPM. The air to fuel ratio was an unknown constant but using the dyno sheet data given with the engine the AFR constants were interpolated from the results in Figure 42. They are placed into Ricardo working back to make 1-D models.


Constants

Constants Table | Sweep Constants | Dependent Constants | External Constants

	Name	Units	Case 1	Case 2	Case 3	Case 4	Case 5	Case 6	Case 7	Case 8
Status			Run	Run	Run	Run	Run	Run	Run	Run
Title			Full load -	Full load -	Full load -	Full load -	Full load -	Full load -	Full load -	Full load -
1	SPEED	rpm	10000	9500	9000	8500	8000	7500	7000	6500
2	AFR		17.5	17.5	17.5	17.5	17.5	17.5	16.5	17.5
3	BDUR	deg	25.2	24.85	24.5	24.15	23.8	23.4	23	22.5
4	CA50	deg	5.2	5.8	6.4	7	7.6	8.2	8.8	9.4
5	Coolant_Head	W/m^2/K	8500	8250	8000	7750	7500	7250	7000	6750
6	Coolant_Liner	W/m^2/K	2500	2400	2300	2150	2000	1850	1700	1600
7	Coolant_Piston	W/m^2/K	2400	2250	2100	2000	1900	1700	1500	1450
8	dxe		35	35	35	35	35	35	35	35
9	dxi		25	25	25	25	25	25	25	25
10	EV_TEMP	K	375	373.75	372.5	371.25	370	369	368	367
11	EX_ANCHOR	deg	250	250	250	250	250	250	250	250
12	HEAD_TEMP	K	620	613.75	607.5	601.25	595	591.25	587.5	583.75
13	idur		50	50	50	50	50	50	50	50
14	INT_ANCHOR	deg	470	470	470	470	470	470	470	470
15	IV_TEMP	K	318	317.5	317	316.5	316	315.5	315	314.5
16	LINER_TEMP	K	595	591.25	587.5	583.75	580	577.5	575	572.5
17	PISTON_TEMP	K	580	572.5	565	557.5	550	545	540	535
18	T_cell	K	313	313	313	313	313	313	313	313
19	T_Coolant	K	383	380.5	378	375.5	373	370.5	368	365
20	T_Oil	K	363	363	363	363	363	363	363	363
21	throttle_angle	deg	90	90	90	90	90	90	90	90

	Name	Case 9	Case 10	Case 11	Case 12	Case 13	Case 14	Case 15	Case 16	Case 17
Status		Run	Run	Run	Run	Run	Run	Run	Run	Run
Title		Full load -	Full load -	Full load -	Full load -	Full load -	Full load -	Full load -	Full load -	Full load
1	SPEED	6000	5500	5000	4500	4000	3500	3000	2500	2000
2	AFR	16.7	13.5	13	13	12.4	11.5	10.2	12	12
3	BDUR	22	21.5	21	20.5	20	19.5	19	18.5	18
4	CA50	10	10.5	11	11.5	12	12.5	13	13.5	14
5	Coolant_Head	6500	6250	6000	5500	5000	4500	4000	3500	3000
6	Coolant_Liner	1500	1350	1200	1100	1000	900	800	700	600
7	Coolant_Piston	1400	1350	1300	1200	1100	1000	900	900	900
8	dxe	35	35	35	35	35	35	35	35	35
9	dxi	25	25	25	25	25	25	25	25	25
10	EV_TEMP	366	365.75	365.5	365.25	365	364.75	364.5	364.25	364
11	EX_ANCHOR	250	250	250	250	250	250	250	250	250
12	HEAD_TEMP	580	572.5	565	557.5	550	542.5	535	527.5	520
13	idur	50	50	50	50	50	50	50	50	50
14	INT_ANCHOR	470	470	470	470	470	470	470	470	470
15	IV_TEMP	314	313.5	313	312.5	312	311.5	311	310.5	310
16	LINER_TEMP	570	562.5	555	547.5	540	537.5	535	532.5	530
17	PISTON_TEMP	530	522.5	515	507.5	500	492.5	485	477.5	470
18	T_cell	313	313	313	313	313	313	313	313	313
19	T_Coolant	362	362.5	363	363	363	363	363	363	363
20	T_Oil	363	363	363	363	363	363	363	363	363
21	throttle_angle	90	90	90	90	90	90	90	90	90

Case Title Format: Full load - {speed} rpm
Number of Cases: 20
Import... | Export... | Display Profile... | Catalog Usage...
OK | Cancel | Help

Figure 41 overall engine parameters in Ricardo Wave

As parameters such as piston temperature, oil temperature, coolant temperature etc… cannot be obtained these values were bought in from Ricardo's 600cc engine. Values for intake temperature range from 309K at 500rpm to 318K at 10000rpm, this ranges from 36° to 45°. The temperature increase comes from the engine radiating heat.

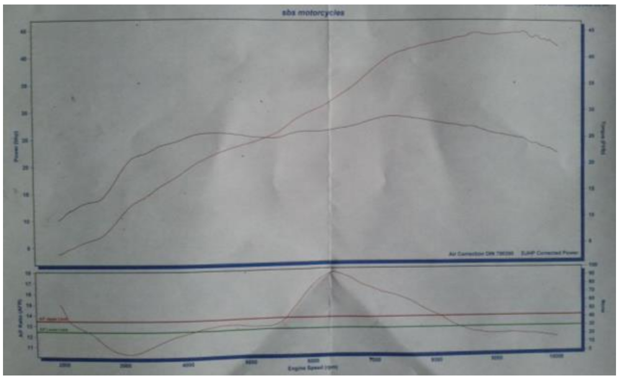


Figure 42 Honda's Dynamometered engine results including AFR (Latif, Raja , 2018)

Naturally Aspirated Simulation

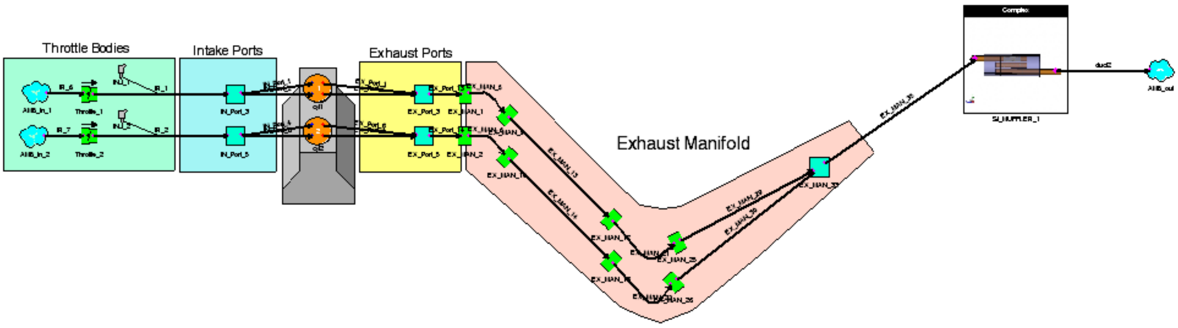


Figure 43 N/A simulation in Ricardo Wave (Latif & Allport, 2018)

Figure 43 shows the model. Ambient air enters two individual throttle bodies, on the left of the image, into the engine and out of the exhaust through a silencer and into the atmosphere. Several aspects of the powertrain were not modelled such as the air cleaner, the catalytic converter and the Honda OEM silencer. These three items were not modelled into the simulation because of lack of information regarding the air cleaner, OEM silencer and catalytic converter.

Nevertheless, the main restriction on an engine is the catalytic converter, so an error margin of 5% was given with the results meaning these results were trustworthy enough to be used as a benchmark for the model to be turbocharged. Furthermore, the silencer used in Ricardo's 600cc engine model was used, as this is a similar engine style to the Honda CB500x.

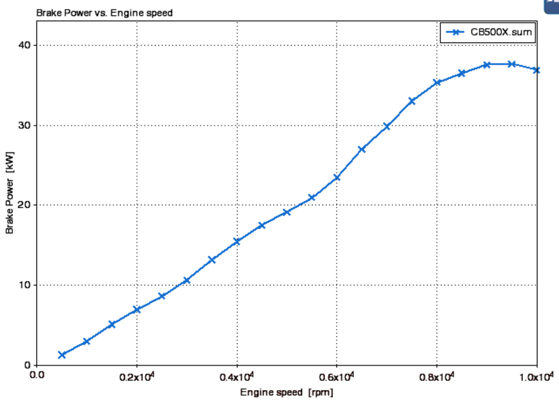


Figure 44 Brake Power results from N/A Ricardo Wave simulation (Latif & Allport, 2018)

Figure 44 and above indicate the power in kW produced by the simulation. The Honda engine produces 35 kW at 8500 RPM whereas the simulation produces 37 kW at 9000 RPM. The error between the simulation and the dynamometer data were analysed and the results show a 5% error between the speed range and power (Latif & Allport, 2018).

However, the dynamometer results can vary due to ambient air temperature as the Ricardo simulation intake temperature is set to 20°C, thus the difference between the model and testing can vary and the dynamometer data can also change due to ambient temperature meaning the N/A model can be relied upon and used to create a turbocharged model.

Turbocharging the N/A model

When engines which are naturally asperated (N/A) are turbochargered the engine itself is not usually rebuilt. Therefore the engine itself within the simulation will stay the same. However the exhaust manifold and intake manifold will change and also an effective cooling method will need to be added to dissipate the heat energy of the added boosted pressure.

The simulation requires several inputs; the first being a compresser map. This can be seen on Figure 45. Several inputs were taken from the compressor map. These inputs were the speed, pressure ratio, mass flow and efficiency. The compressor map is the turbocharger's characteristics map; it can tell a user if the turbocharger will fit the engine aswell as at show speeds at which the turbocharger will spin and the adequate compresser efficiency. Additionally the compressor map will show the mass flow produced by the turbocharger and its corresponding pressure ratio.

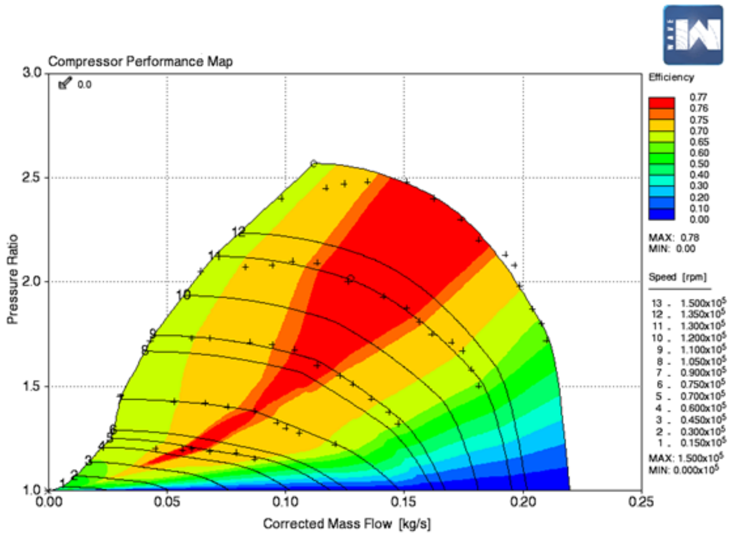


Figure 45 Compressor map added to Ricardo Wave (Latif & Allport, 2018)

Once placed into the simulation, the simulation interpolated the inputs of the compressor map data and adds in the choke margin of the compressor map. This can be seen on Figure 39 with the efficiency regions in blue.

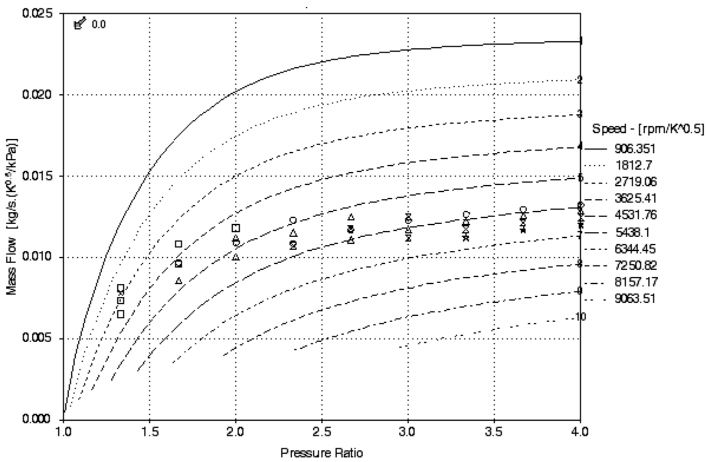


Figure 46 Turbine Performance Map

Figure 46 represents the turbines map. As the data from Garrett was very minimal for the turbine an adequate turbine from Ricardo's turbocharger library was used. However, the inputs for the turbines map were adjusted to correspond with the Garrett GT2056 turbine map. This is because the data from Garrett showed the gas turbine flow against the pressure ratio in addition to the maximum efficiency.

As the corresponding speed is required for the simulation the speed results were interpolated from several turbine maps to produce a turbine map with the correct graph curvature along with a maximum efficiency of 65%.

This turbine map is not the ideal case to use in a simulation as the results will vary from the Garrett map however the simulation will produce an accurate idea of the power and torque that the powertrain could produce.

What can be expected

Turbocharged engines mostly produce torque earlier, compared to their corresponding naturally aspirated version. This means that if an engine before it was turbocharged, produced its maximum torque at 6000rpm due to forced induction; after being turbocharged the engine will see maximum torque earlier on in the engine's speed range. Additionally, the maximum torque will stay constant throughout the engine's speed range until the engine reaches the redline.

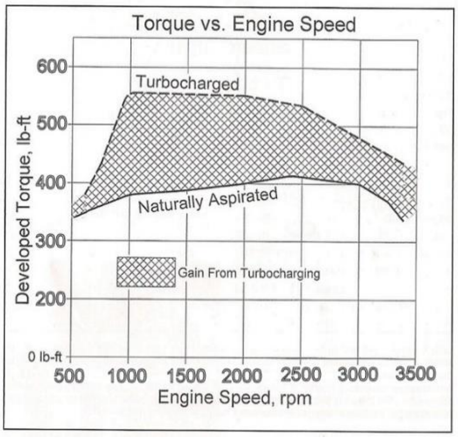


Figure 47 N/A torque against Turbocharged expectations (turbollc, 2012)

Figure 47 clearly indicates the effects that boost pressure has on an engines capability to produce torque. The reasons for this is that naturally aspirated (N/A) engines volumetric efficiency is around 60 to 80%; this means that the cylinders are not filled totally with air. Hence, as the engine starts to gain engine speed the volumetric efficiency rises to around 80%. At this region the engine will produce its maximum torque.

Boosted engines have air forced into the cylinders. Usually turbocharged engines have an initial volumetric efficiency of around 80% at engine idle. As boost pressure is produced the volumetric efficiency rises to over 100% to around 120%. This means that the engine can add more fuel into the air mixture in the cylinders and produce larger amounts of torque.

PID controller

A PID controler stands for 'Proportional Integral Differential;' it is a effective feedback controller. The PID controller in this cause is used to set a target boost pressure. It does this by comparing the current system output with the defined target set point to calculate an error signal.

The magnitude and sign of the error is used to generate a restoring signal. This is then used to drive the system to correct the error. For example, the PID controller is targeting a maximum boost pressure of 0.5bar therefore the simulation uses a sensor to pick up pressure within the runners of the plenum/intercooler then sends that information to the PID controller which regulates the wastegate to target 0.5bar of boost pressure.

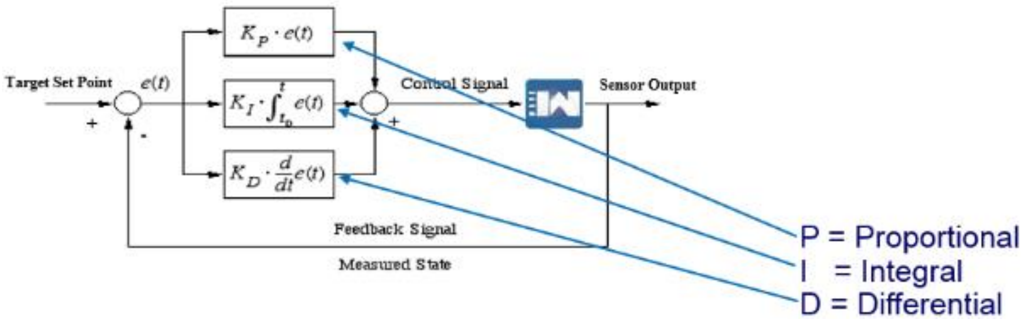


Figure 48 PID controller in Ricardo Wave

Wastegate pressure

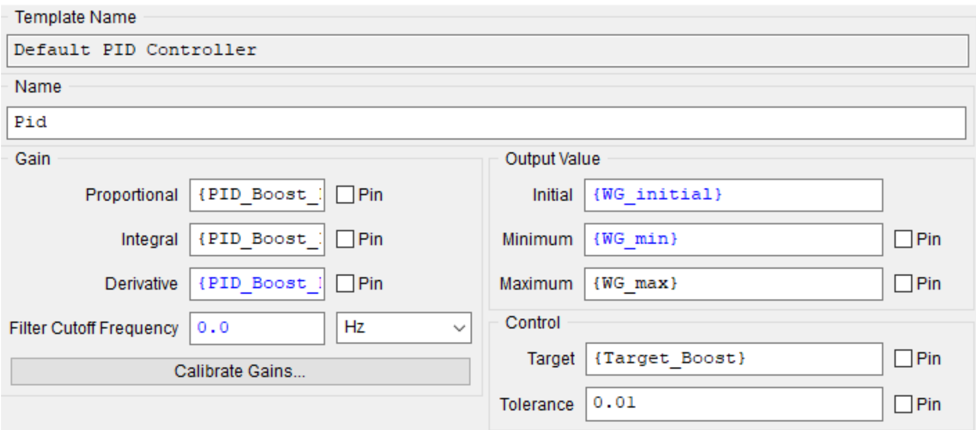


Figure 49 Target Boost and wastegate in Ricardo Wave

The PID output values regulate the wastegate's pressure to 'open area' in mm^2 this means that the outlet boost pressure can be regulated through changing the simulations wastegate open area. Therefore, three inputs were placed into the output values 'WG_initial' is the wastegate open area that the simulation abides by to reach target boost pressure. However, if the target boost is overreached the simulation interpolates from the minimum and maximum output values which were set to 800 mm^2 and 0 mm^2.

Target Boost Pressure

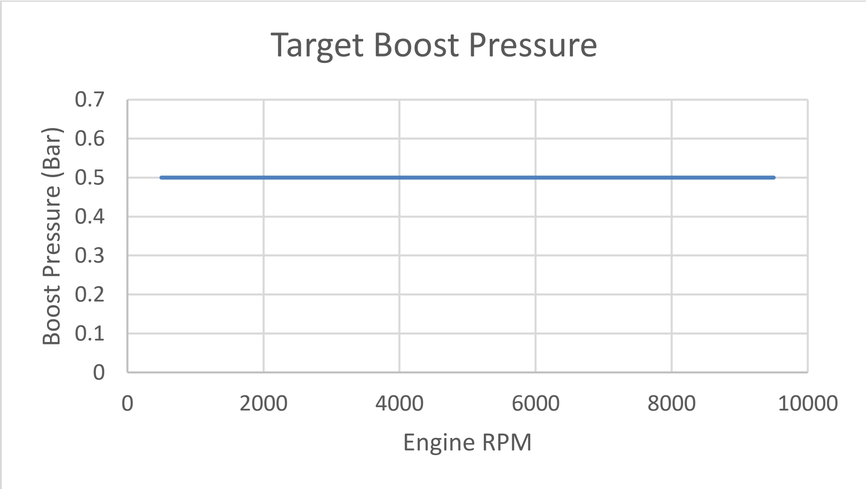


Figure 50 Target boost pressure set in Ricardo Wave

Target boost pressure was set to 0.5Bar of boost pressure throughout the rev range. The simulation will target that boost value and work alongside the wastegate open area values and work towards having a target boost pressure of 0.5Bar.

Compressor speed

To adequately determine compressor speed, a turbocharger speed sensor will need to be added to the turbocharger through its dynamometered period. However the simulation inputs this data to proceed into analysing the effects of the turbocharger.
Therefore Figure 51 shows how a turbocharger compressor map this map has two lines drawn upon the map data. The first line referencing close to the surge margin is the max torque line. The full speed line indicated the highest speed the turbocharger will reach when the engine reaches maximum power.

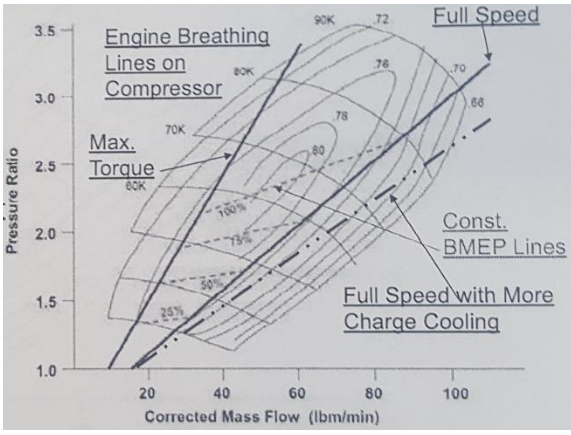


Figure 51 compressor speed theory (Shahed & Tuteja, 2018)

From understanding the effects of a turbocharger on the engine's ability to produce torque and power an engine will first produce its maximum torque at the phase through the maximum efficiency points towards the full speed margin. However, as the turbocharger is much larger in size compared to the engine displacement the maximum speed capable should be around 125krpm from the turbocharger sizing calculations.

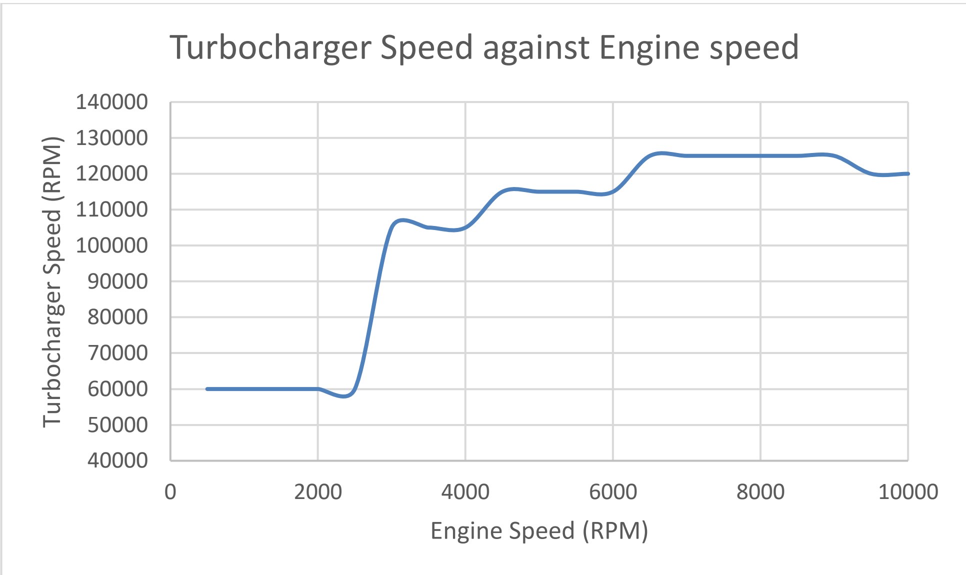


Figure 52 Turbocharger speed initially placed into Ricardo Wave

Abiding by the principle that the max speed capable fo the turbocharger should be 125krpm. The simulation input for maximum speed was set to this value between 6500-9000 rpm. This is where the maximum speed of the compressor should occure.

Turbocharger idle was set to 60krpm this was set high compared to a lower 40krpm idel speed this was done because the simulation would use these reference speeds as a target. The simulations turbocharger will work to 'AUTO' this means that the rotational speed and outlet temperatures aswell as other data will be calculated by the simulation.

Exhaust manifold and downpipe

Figure 53 Exhaust Manifold used in Testing

The lengths and diameter were placed into Ricardo Wave so that the software modelled the exact manifold that will be used for the engine dynamometer testing. This manifold was designed around one Y-junction on Ricardo using the bend where the 2-1 meet up incorporates the bend angles into the software.

The downpipe was a simply designed short 90° bend with a 450mm length tube to open atmosphere. A silencer or resonator will not be used. The reason for the exhaust being designed in this manner was to keep the design simple. Also, using the same design in the simulation and on the dyno should theoretically reduce error.

Resonators and silencers require data which makes it difficult to interpolate the data into Ricardo wave. However, the simulation can be simplified by adding a 3” resonator if the decibel reading on the dyno is too high. Therefore, a simple 3” resonator may be added to the simulation later.

Exhaust manifold design

For performance purposes the manifold design is not adequate, due to the tight 90° bend after the 2-1 collector. This tight bend restricts exhaust flow. Figure 54 shows a simple single cylinder designed exhaust manifold, which has been analysed using Computational Fluid Dynamics (CFD). Where the exhaust manifold angles to the engine flange the CFS’s simulation changes colour. Where the simulation shows red flow streamlines around the inside of the 90° bend, it means that the inside of the bend has a low-pressure zone therefore, the exhaust gas efficiency to run freely through the exhaust is restricted due to the tight bend.

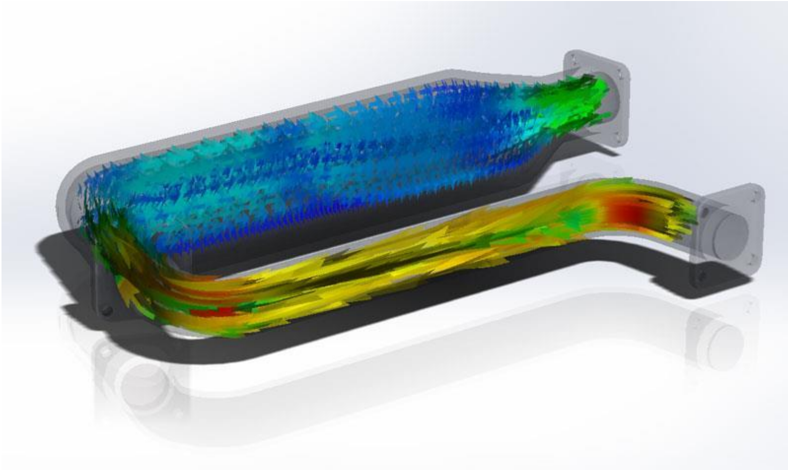


Figure 54 Exhaust manifold design theory (Solidsolutions, 2018)

Intercooling/plenum

Figure 55 Intercooler/Plenum chosen for testing

For the intercooler and plenum, a water to air intercooler was used in the setup. (Figure 55) The three-litre cubic capacity of the intercooler is large enough to accommodate boost pressure, as well as keep the boosted air charge as dense as possible for maximum performance.

Naturally aspirated vehicles have certain criteria. When designing a performance engine runner length, volume and plenum volume all play a part in tuning a naturally aspirated vehicle for performance. The air waves within a plenum and the runners can be tuned for achieving maximum efficiency in power and torque. However, for turbocharged engines these three criteria for plenum volume and runner length and volume do not affect the performance of the engine.

Pressurised air within the intercooler/plenum cannot be tuned therefore for a forced induction application all the aspects of tuning the air waves are irrelevant.

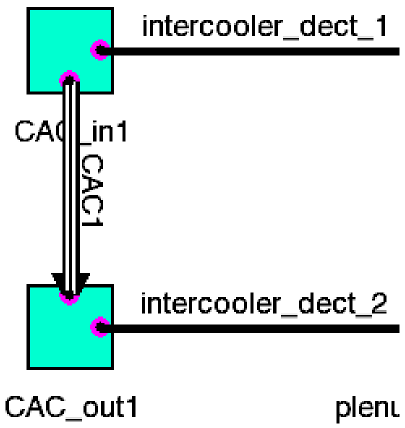


Figure 56 Ricardo Wave intercooler/plenum setup

Figure 56 above shows the intercooler/plenum in the Ricardo wave model; the simulation makes it difficult to simulate a water to air intercooler. However, an air to air intercooler can be simulated to act like a water to air intercooler. The difference between both charged air coolers is, one is cooled via water and the other is cooled by air, therefore the simulation can be manipulated to act like an air to water intercooler.

Engine temperatures

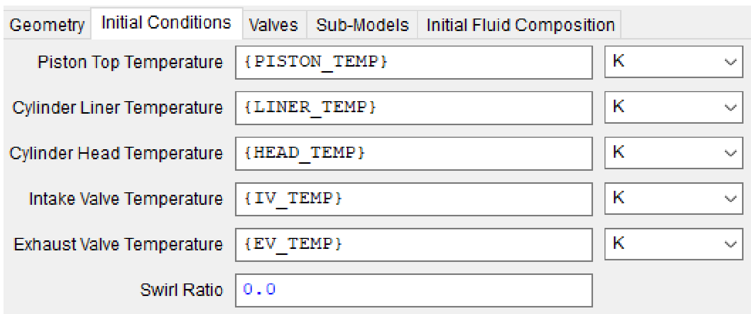


Figure 57 Ricardo Wave engine temperatures parameters

Figure 57 above shows the engine temperature parameters. These temperature parameters will be kept the same as the naturally aspirated model for now. After the engine is dyno-tuned and tested a cold start pull will be made. The data from the dyno test will be placed into the Ricardo perimeters to minimise the errors between the simulation and the test.

Simulation errors

Several errors can be noted from the simulation in advance, the three points below clearly explain the errors that the simulation may have.

Measurement – the simulation requires several measurements such as the diameter of the piping as well as the lengths and bend angles. This was done using a vernier calliper and rulers, the 3D printed components such as the intercooler to cylinder ports were measured using CAD. These measurements are critical in the simulation as the simulation can verify the pressure drop from the compressor discharge to the cylinder ports.
Materials – the material properties such as the stainless-steel exhaust manifold and downpipe were taken from material properties online however, there are several stainless-steel types, and this can affect the results outcome by a small margin, this can be down to heat dissipation.
Turbocharger maps – both the turbine and compressor maps were interpolated from their map images and placed into the simulation however, the 100% data was not used; this is because Garrett has not published the maps publicly. The data from both maps can affect the result outcome in many ways.

Turbocharged simulation

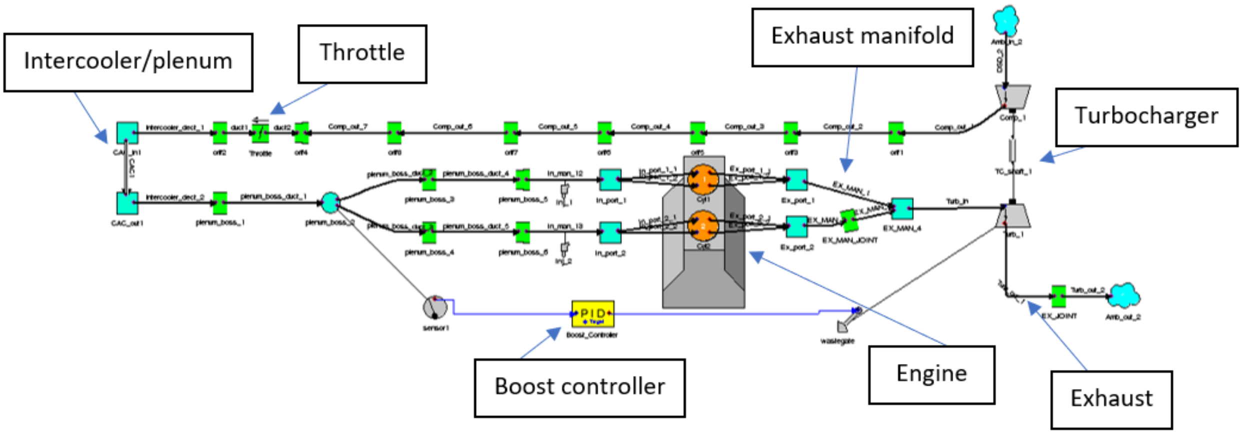


Figure 58 Honda CB500x Turbocharged Simulation (Latif & Allport, 2018)

Figure 58 shows the engine simulation without the electrically assisted turbocharger. The blue cloud at the top right hand of the Figure 58 is the ambient temperature and pressure. These values in the simulation can change. The compressor sits behind the ambient cloud. The black line from the compressor to the throttle in the turbocharger's piping. This piping was measured from the powertrain assembly and plotted into the simulation. The throttle is a 30 mm throttle body.
The rest of the simulation to the intake ports just after the injectors is all measured from the powertrain assembly. The injectors are set to 'AUTO;' this is because the injector flow rates can be matched to the calculations and from this data appropriate injectors can be setup.

After the engine exhaust ports the manifold is sized and measured from the powertrain assembly till the exhaust to the outlet blue ambient cloud. The boost controller from the simulation has a pressure sensor set up to the intake runners and is connected to the wastegate via the PID controller which regulates the boost pressure.

ETC Simulation

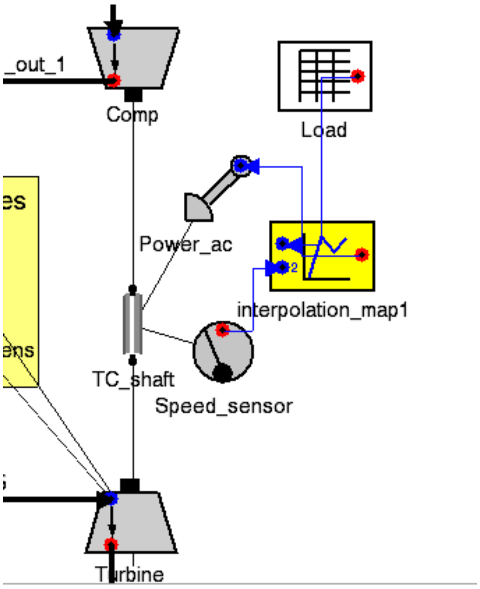


Figure 59 ETC added to Ricardo Wave turbocharged model

The ETC was modelled using four extra components these components were a speed sensor, a 'power_ac' which is the electric motor an interpolation map and a load input constant.
The interpolation map holds the key information of the motor's specification such as power and speed. From this the load input constant which is designed, so it works in a binary code with 1 being on and 0 being off. Therefore, if at engine idle 1000rpm the ETC had a load input constant of 1 it will send the 9 kW of power to the turbocharger.

Chapter 6 Design

Electric Motor Capabilities

Calculating the required power and torque needed to overcome the inertia of the turbocharger requires accommodation of the losses. Having a motor that spins at the required rpm – 70 krpm in this scenario – with the required torque and power, also a motor that allows the turbo to sit at idle at the required rpm would be ideal.

The electromagnetic torque T_e is key; this can be done by using the equation below. This equation accounts for the total inertia of the system including the turbo motor and is represented by $J\frac{d\omega}{dt}$. Also, the power consumed by the turbocharger $Tr\omega$ as well as other losses such as shaft misalignment, motor losses, windage and other losses which is represented by $\beta\omega$.

$$T_e = J\frac{d\omega}{dt} + \beta\omega + Tr\omega \text{ (Schofield, 2018)}$$

Several assumptions have been made, such as the motor's rotor has a diameter of 20 mm and a length of 80 mm with a density of 7400 kgm^3 (Sales, 2000). The next assumption made is that it will only take 1 second to boost at engine idle; this is the time from engine idle that the motor will be working. Furthermore, 70 krpm was used as the rotational velocity; this is the minimal efficiency speed of the compressor.

The calculations can be done in steps firstly calculating the total inertia of the system which came to 3.8299e-05 kgm^2 with the motor having 9.2991e-06 kgm^2 inertia. From this calculation the total power and torque for the rotational inertia can be calculated.

$$torque = J\frac{d\omega}{dt} = 3.8299\text{e} - 05\ (\text{kgm}^2)\text{x}\ \frac{70\ (krpm)\ \text{x}\ {}^{2\pi}\!/_{60}\left({}^{rad}\!/_{s}\right)}{5\ (sec)} = 0.0561\ (Nm)$$

$$power = \left(0.0561\ 70\ (krpm)\ \text{x}\ {}^{2\pi}\!/_{60}\left({}^{rad}\!/_{s}\right)\right) x\ 10^{-3} = 0.4116\ (kW)$$

From this calculation it is required that 2.0580 kW of power is required however, this is not accommodating for losses within the system. Adding 2 kW of absorbed energy of the compressor can be added this gives us 4.0580 kW of power required and 0.5536 Nm of torque. Nevertheless, to accommodate for the $\beta\omega$ other losses a motor with 4.5 kW and 0.6 Nm of torque is required. Though, this might change due to other motor specifications a MATLAB script was made for any changes.

MATLAB Code

```
clear all
clc
% Author R S Latif
% Date: 07/04/2017
% this code is to calculate the required power needed in a motor.
%% calculations
Jt=2.9*10^-5; % turbochargers inertia (kgm^2)
Dr=20*10^-3; %motor rotor diameter (m)
Lr=80*10^-3; %motor rotor length (m)
den=7400; %density of magnets in motor (kgm^3)
Jmot=((pi*(Dr)^4)/32)*Lr*den; %inertia of motor
Tine=Jt+Jmot; % total inertia (kgm^2)
RAD=70000*(2*pi/60); %rotational speed needed in (rad/s)
s=1; % time taken to boost
%% Power and Torque
T1=Tine*(RAD/s); %initial torque without other losses (Nm)
P1=(T1*RAD)*10^-3; %initial torque without other losses (kW)
T1
P1
%% Power and Torque needed including losses
TEA=2; %power absorbed by turbo (kW)
P2=P1+TEA; % total power needed (kW)
T2=(P2*10^3/RAD); % total torque needed (Nm)
P2
T2
%% end code
```

Motor specifications

Model No.	T	KV	Picture	Max Amps (A)	Max Volt (V)	Continious(W)	Max Power (W)@10seconds	Max RPM@10seconds	Rm	NO-Load Current	Shaft Diameter (mm)	Diameter*Length (mm)	Output shaft Length(mm)	Bolt Pattern(mm)	Weight (g)	Poles
TP4070-CM	3D	3200		382	23.5	5000	9000	75K	0.0019	12/7.5	5mm	V1:40*107	18mm	4-M4-25	640	4P
	4D	2200		264	34				0.0024	9.8/7.5V						

Figure 60 Electric motor specifications (TPPower, 2012)

Above are the motor specifications however, we need a motor which is 4.5 kW capable with 0.6 Nm of torque. Therefore, calculations can be made to see if the motor is suitable for this application, as well as calculate its maximum deliverable torque.

$$\omega_r = 75000\,rpm = 7853.98\ {}^{rad}/_{s}$$

$$E = 23.5\,V$$

$$R = 0.0019\ \Omega$$

$$A = 328\,A$$

$$V = 23.5V$$

Using the five variables above the motor stall torque can be calculated.

$$K_\emptyset = {}^{E}/_{\omega_r} = 2.992 \times 10^{-3}\ {}^{Vs}/_{rad}$$

$$T = \frac{K_\emptyset}{R_a}V = \frac{2.992 \times 10^{-3}}{0.0019} \times 23.5 = 37Nm$$

$$23.5V = I_a \times 0.0019 = 12368.4\,A$$

Stall torque is calculated using the voltage divided by resistance, however the power electronics will not be able to handle 12368.4A, the amperage would be limited by the power electronics. In practice the power electronics limit motor current to 328A the motors actual limit current, thus preventing operations at stall. Consequently, the maximum voltage that the motor can draw is 23.5V due to battery voltage. Below are the equations used to calculate motor specifications, with $K_\emptyset$ being a constant.

$$V = 382A \times 0.0019\Omega = 0.7258V$$

Voltage is equal to motor amperage multiplied by the resistance across the motor.

$$torque = \frac{K_\emptyset}{R_a}V = \frac{2.992 \times 10^{-3}}{0.0019} \times 0.7258 = 1.143Nm$$

Once torque is calculated the rotational velocity is calculated.

$$\omega_r = \frac{V_a - (R \times A)}{K_\emptyset} = \frac{23.5 - (0.0019 \times 382)}{2.992 \times 10^{-3}} = 7611.7\ {}^{rad}/_{s}$$

$$7611.7 \times \frac{60}{2\pi} = 72{,}686\ rpm$$

Finally, the motor power is calculated at 8.7kW.

$$power = 7611.7\ {}^{rad}/_{s} \times 1.143Nm = 8.7kW$$

Recalculating the voltage that the motor will draw the actual torque is calculated as a constant of 1.143 Nm. Correspondingly, it produces 8.7 kW at 72 krpm though, the motor can only produce 1.143 Nm of torque for less than 10 seconds. This is because copper loss is proportional to $P_{cu\theta} = 3k_\theta\, R\, I_{RMS}^2$ (Andrada, Torrent, Perat, & Blanqué, 2004) where θ is the temperature R is the resistance in Ohms and I_{RMS}^2 is the current. As the torque is proportional to the current (I), the greater the torque the greater the temperature, as the torque increases so does the temperature, this in translation means more copper losses therefore the electronic throttle control will limit the motors torque to a time base to preserver the motors life expectancy.

However, looking at the Figure 61 the TP4070-CM motor is capable for the electronic assisted turbocharger. This is because the 4.5 kW of power required and 0.6 Nm of torque that is required falls below the area under the curves of the graph.

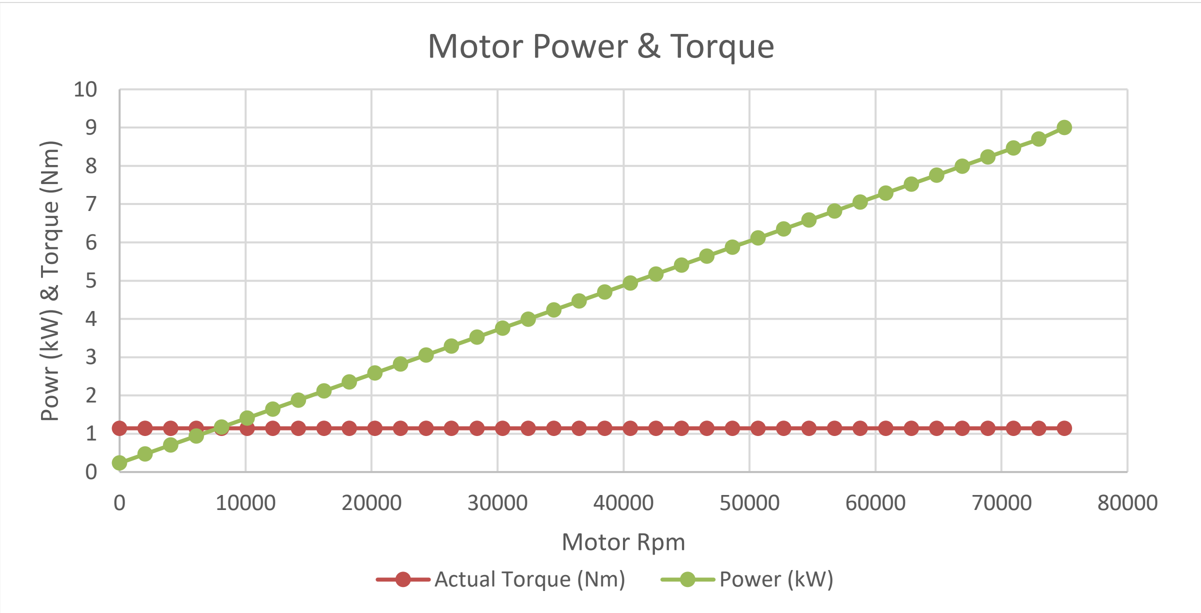


Figure 61 Electric motor calculated power curve

The design criteria calculated that a 4.5kW and 0.6Nm motor was needed however, the specifications dictated that a 70krpm+ motor was required to allow the turbo-compressor to make adequate boost pressure. Therefore, looking at suitable motors which were available from the motor supplier which had a power capability of 4.5kW and 70krpm+ speed the TP4070-CM motor was chosen. Although overrated the motor allows for a margin of design for test purposes.

Components

Number	Component	Name
1	Battery	Alien 6S 16000mah 35C Lipo Battery flat configuration
2	ESC	Alien 300A 3-8S Car ESC BEC
3	motor	TP 4070-CM
4	charger	ICHARGER 106BPlus 1-6S Li-xx 10Amp 250W
5	Controller	Alien Power System 2.4Ghz Electric Skateboard Remote control

Table 5 Components required for ETC

Table 5 shows the components required for the ETC system. The controller is handheld module. However, the ESC can be controlled via laptop with specific programs, this can vary speed, voltage, power and torque. This can be a possibility to use. The battery is a 6s lipo battery this means it has a 22.2 V as 1s is 3.7 V.

The ESC is the motors control module this is a 300 A 3-8s system. The reason this ESC was specified was due to its several programable functions. The two-main being throttle curve (logarithmical, linear, exponential) and acceleration control (Extra soft/Soft/Medium/Hard/Extra hard). Therefore, the testing for the ETC system can be varied for different tests to determine which setting the ESC can give the best performance outcome to the powertrain.

Electrical Circuit

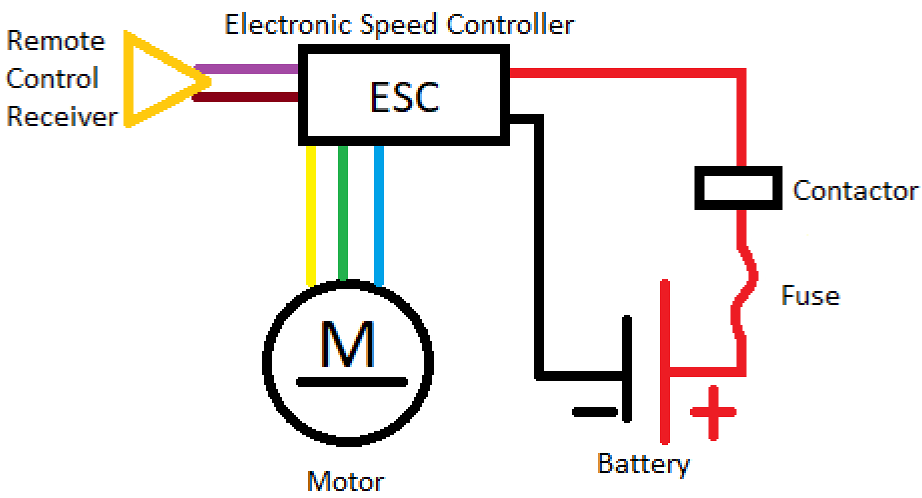


Figure 62 Electric circulate diagram for ETC

The circuit diagram above shows the configuration of the ETC system. The system incorporated a 400 A fuse the reasoning behind this fuse is that the battery is 22.2 V source the motor has a maximum wattage of 9000 W (for 10 second). The motor cannot draw any more voltage than the battery can supply. Then determining the fuse amperage, the watts can be divided by the volts as shown below in Equation 1.

$$\frac{Watts}{Volts} = Fuse\ rating\ [1]$$ (Judge Electrical Limited, 2018)

$$\frac{9000\ (W)}{22.2\ (V)} = 405.41\ (A) \approx 400\ (A)$$

The fuse required is a 405 A fuse however, the next size up form a 405 A is 450 A this would damage the ESC if the motor seized. As when a motor jams it demands the most amount of current as when they are at 0 rpm they have the greatest torque. Therefore, if the ETC system seizes due any form of error the 400 A fuse will blow before the ESC is damaged.

Furthermore, a contractor will be placed into the circuit. Contactors work like relays, they have a coil which must be energies using a 12 V supply. Once 12 V is supplied the coil is energised allowing for the battery to send power around the system, the contactor can handle the higher amperage demanded by the ESC. This will allow the ESC switch to always be on. The use of the contactor will allow for an on and off switch will be connected to the dynamometer battery allowing for greater control of the ETC, and act as a failsafe working along with the fuse (Frank, 1999).

Drive

There are several problems with driving a turbocharger. Firstly, shaft misalignment can cause a large side force on the turbine shaft therefore throwing the balance of the turbocharger. This can cause early failure to a turbocharger. Secondly, acceleration of a turbocharger does not allow any couplers to be adapted into the design due to high rotation velocity of a turbocharger. Most couplers are not designed to speeds over 10krpm.
Therefore, with some research the decision was made to use a Quill shaft to isolate both the two main problems with designing an electrically assisted turbocharger.

Quill shaft

A quill shaft is a specifically designed tube or solid bar which transmits drive. These shafts are designed to be as light as possible. This is to reduce the amount of overhung weight which means that the whole drive system has very little vibration throughout the system.
Commonly, quill shaft generally uses a single hinge coupling. This is because the shafts are designed to bend in a cantilever manner. The hinge is added to the design to the design so that the shaft has misalignment properties. However, designing the shaft to be as this as possible will allow the shaft to bend and move likewise allowing the system to have misalignment.
Isolation of the high-speed turbocharger, because of the torsional thin shaft, results in a lack of generation of frequencies such as 'gear mesh frequencies' and 'driver pulsations' (America, 2014).

Material Properties

Designing the quill shaft to be as light as possible to minimise the inertia. In addition to also take the drive torque of the motor. Bronze SAE 660 was the material chosen this is due to its easy accessibility as well as material properties. Steel was not used due to its high yield strength the shaft would have to be thin (in diameter) shaft which would cause machining problems. Likewise, aluminium 6061 T4 or T6 was not chosen due to either the shaft being too thin which will result in machining problems.
Calculating the maximum thickness that the quill shaft would have to be with a 15% additional torque to allow for an extra factor of safety (FOS) before failure. The quill shaft will be designed to fail if the torque exceeds 0.69Nm. This design consideration was made to allow

the quill shaft to fail if the motor jams or the turbochargers misalignment does cause turbocharger failure.
Bronze SAE 660 has a yield strength of 138MPa which is equal to $138\times10^6 N/m^2$ (Services, 2002). Designing a shaft which as the same shear stress as the yield stress means the quill shaft will have a FOS of 1 if the maximum torque applied is 0.69Nm. Therefore, calculations were made using MATLAB, as these calculations were repeated until the shear stress matched the yield stress, allowing for a FOS of 1. (MATLAB code below) The calculations show that the polar moment of inertia if firstly calculated, then the shear stress.

```
clear all
cl
% Author: R S Latif
% Date: 22/03/2018
% code to calculate shear stress.
%%
Tmax=0.69; %maximum torque
D=2.95;
r=D/2;
%% polar moment of inertia
J1=(r*10^-3)^4; %polar moment of inertia (m^4)
J=((pi*J1)/2);
J
%% shear stress
t=(Tmax*(r*10^-3))/J; %Shear Stress in the Shaft (N/m^2)
t
```

Overall design

With the above calculation the quill shaft will have a centre section diameter of 2.95mm. one side will have a M5x0.8 left hand tapped hole for threading the turbocharger. The other side will have a straight 5mm hole with a depth of 10mm which will be either glued or thread locked to the motors drive shaft. Figure 63 shows the design of the quill drive shaft.

Figure 63 Quill shaft design

The quill shaft will only be connected to the turbocharger and motor. Firstly, no bearing will be used to support the shaft, this is because the motor has two bearings and the turbocharger will have two/three bearings. This is sufficient to support the quill shaft.

After consulting with the machinists, the quill shaft is possible to manufacture however, the centre of the shaft will cause several difficulties. The first being clamping the quill shaft to the CNC lath grips, the diameter is to small and will require new manufactured tools to clamp the quill shaft in the lath. The second problem being boring the holes for tapping, this is due to the material properties and the fact that the centre section is small in diameter and will snap with the force applied by the CNC lath.

Therefore, a new plan for the quill shaft was made. The new design would incorporate a 5mm centre hole through the quill shaft. Therefore, a new shear stress was calculated with a code for the new calculation below. The new shaft would have an external diameter of 5.25mm with an internal diameter of 5mm this intern would give a FOS of 1.

```
clear all
clc
% Author: R S Latif
% Date: 22/03/2018
% code to calculate shear stress.
%%
Tmax=0.69; %maximum torque
d=5;
D=5.25;
%% shear stress
t1=(Tmax*(D*10^-3)*16); %Shear Stress in the Shaft
t1
t2=pi*((D*10^-3)^4-((d*10^-3)^4));
t2
```

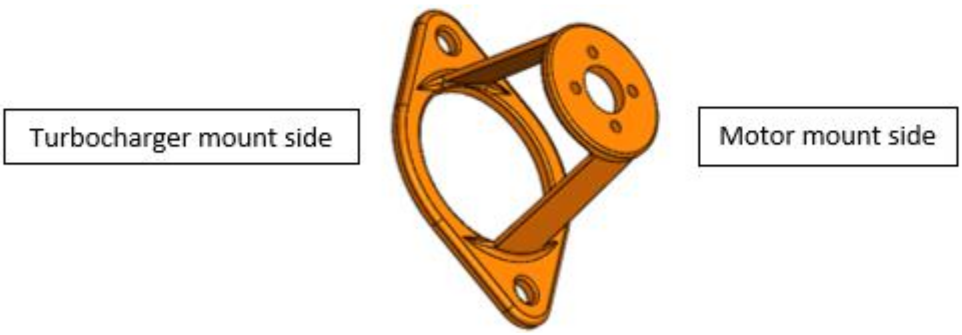


Figure 64 Initial MKI concept design for mounting motor to turbocharger

Figure 64 above shows the MK1 concept design of the support bracketry for the motor to be mounted to the turbocharger. However, the design is floored, this is due to the time needed to machine the component in addition to the complexity of the design. The theory to the design is simple, the flange to the right is designed to mount the motor using four M4 bolts. The flange to the left will mount to the turbocharger using two M8 bolts, the two braces which connect the flanges together are incorporated into the design to centre the motor and the turbocharger to minimise shaft misalignment.

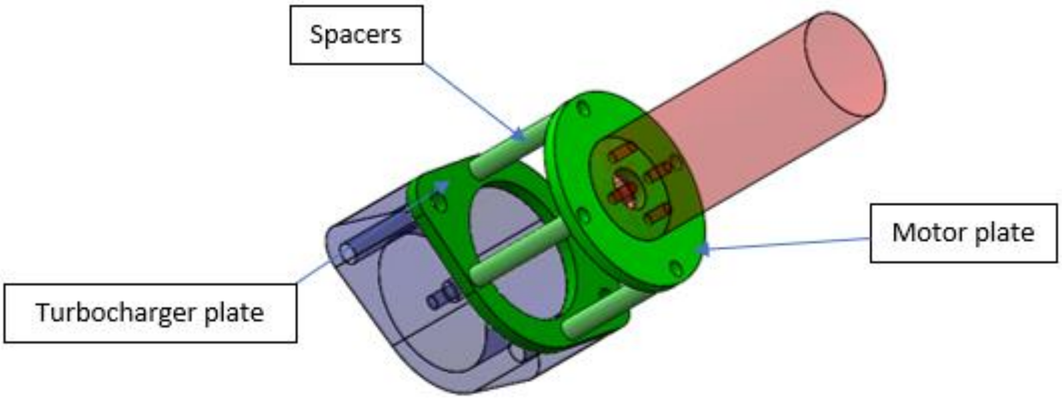


Figure 65 MKII concept design to mount motor to turbocharger

Figure 65 shows the MK2 revised design of the support bracketry. This design was made using four spacers designed to accommodate for M6 bolts. These bolts will tie down the whole system and requires very minimal machining. Despite having more components than the initial design the new revised design cuts down machining time and is cheap to manufacture. The two plates will be made from steel as well as the spacers.

From consulting with machinists to manufacture the components the outline of the plates causes machining problems. The main problem is clamping the inside of the plate to machine the outer contour of the plates. However, if the plates were square or rectangular on the outside the machining time will be cut in half. Therefore, the plates will be re-designed.

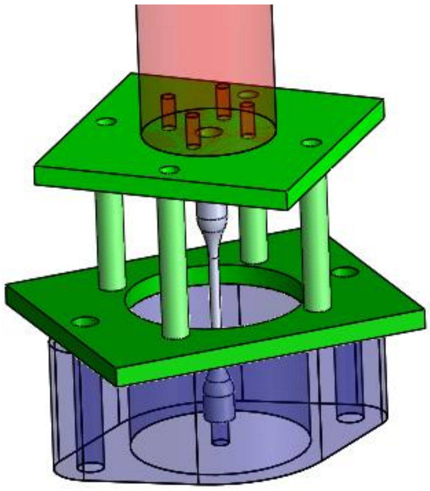


Figure 66 MKIII final concept design

Figure 66 shows the MK3 revised design of the support bracketry. This design simplifies the machining, this is because 6mm steel plate are available. The machinist cut the outline by hand and then used the milling machine to cut the rest. The spacers are also made from steel and will be made using the lath.

Oil pressure

Turbochargers need oil flow to lubricate the bearings, usually this oil flow is tapped from the high-pressure oil chambers within an engine. However, turbochargers do not require pressure to lubricate the bearings, the just need oil flow.

From past tests of turbochargers on motorbike engines that have been previously tested at the university, the turbochargers do not last long. Usually the oil passes through the seals onto the compressor and turbine. From excessive oil loss, the turbocharger will not produce boost pressure. This is because motorbike engines usually produce higher oil pressure compared to car engines.

A 4G63T is a high performance 2.0L engine produced by Mitsubishi. This engine at idle produces 78kPa of oil pressure at higher engine speeds. The engine's oil pump will produce 200kPa or 2bar of oil pressure. The turbocharger that supports this engine can handle the oil pressure produced by the engine without passing large quantities of oil through the system.

On the other hand, a 2006 Honda CBR600 is a 0.6L high performance motorbike bike engine these engines usually produce 100kPa of oil pressure at engine idle. At high engine speeds oil pressure is raised to 500kPa or 5bar of pressure. As motorbike engines have much higher oil pressure then car engines the extra oil pressure produces will be passed through the turbocharger's seals. In return the turbocharger will not produce boost pressure. Symptoms such as oil through the exhaust or either white smoke will be seen if the seals fail.

Oil pressure solutions

There are several solutions to restrict the oil pressure. The easiest will be to incorporate a pressure relief valve can be incorporated into the oil feed from the turbocharger to the engine. A pressure relief valve will regulate the pressure within the feed from the engine to the turbocharger it can be designed so that the extra oil will be returned into the engine's sump. Therefore, a maximum of 2bar oil pressure through the turbocharger will be achieved.
However, as the project will be to test the concept of an ETC the turbocharger will have an external oil feed separate to the engine. However, this system will have to have an oil cooler due to the heat transfer from the turbocharger.

Chapter 7 Dynomometered Results

Naturally Aspirated Results

For any engine to be turbocharged a base map should be designed from the naturally aspirated (N/A) baseline. A N/A baseline map will be designed in a way so that the N/A setup will have the best fuel and ignition timing for maximum torque. Therefore, when the powertrain is turbocharged it will have some changes to ignition and fueling, to accommodate for the additional boosted air pressure from the turbocharger.

There is no base map from Motec for the CB500x therefore, a Honda CBR600 map was used as a base map. However, as it was developed from a superbike there were several changes which had to be made before the map could work on the CB500x which is a commuter-bike engine. Firstly, the map was modified to work with two injectors and coils.

Secondly, the firing order was set to 1-2 with ignition pulses set 180° apart. The injector scaling value was set to 8 milliseconds (msec), this is the time that the injectors will stay open for, if they are running at 100% injector duty. From this the fuel pressure was set to 3.5 bar and the engine was tuned.

The power curve started from 4000 rpm as this is when the engine has a constant oil pressure of 5 bar which will prevent the engine bearing from being starved of engine oil and to prevent knock when full load is applied to the engine by the dynamometer. The engine was also tuned to 8000 rpm, this was due to time restraints along with making a base map before the engine is turbocharged.

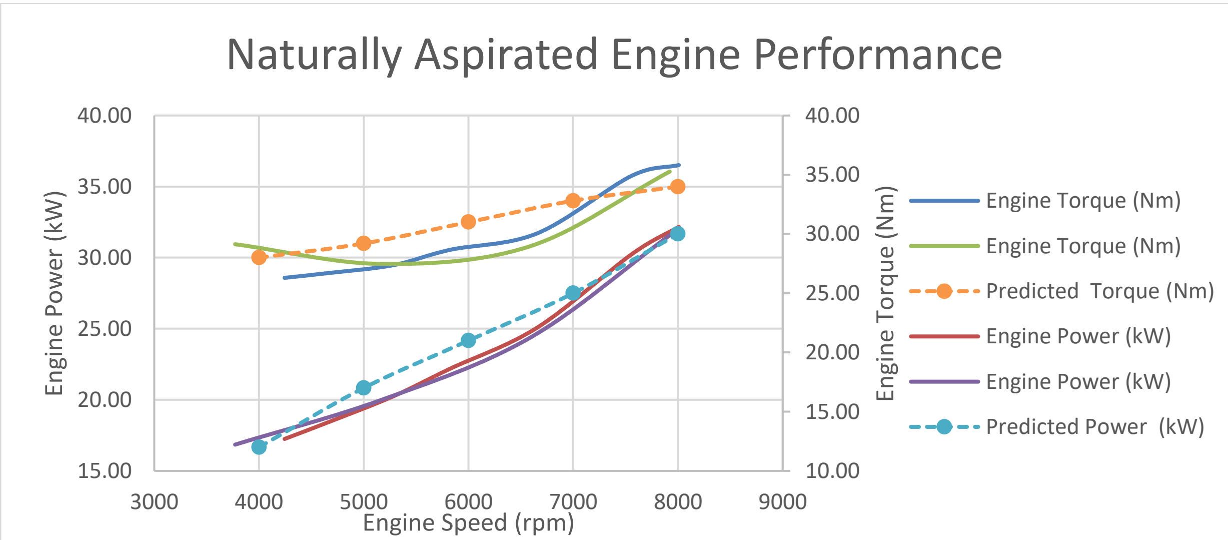


Figure 67 N/A dynamometered results

Figure 67 shows two repeated runs from the naturally aspirated engine to demonstrate repeatability. The two dotted lines are predicted power and torque results, this prediction was made before the engine was tuned and dynamometered. This prediction is still good despite three points that meant that the basis for the estimate did not exactly match the naturally aspirated engine as tested.

The first point was the plenum size and volume was larger than would normally be used. Naturally aspirated engines must have a certain volume in their plenum chamber relative to the engine's displacement volume. The volume of the intake in this case was 3l. This is a very large air chamber for the 471cc engine which therefore struggled to idle at 1500 rpm therefore idle was set to 3000 rpm.

The second point was tuning of the runner volume, runner length, velocity stacks, throttle body and exhaust. The engine was set up to accommodate the turbocharger however, the turbocharger intake components were used in the naturally aspirated engine tuning. This decreased the engines ability to create power and torque, as a naturally aspirated engine must have an exhaust that is tuned to the work with the engines pulses as well as with the tuned runners and velocity stacks to increase the engines intake of air. The original engine has two Individual Throttle Bodies (ITB's) with a diameter of 34 mm each, however the engine in this case was tuned with one 30mm throttle body. This restricted the volume of air able to be drawn by the engine decreasing the Volumetric Efficiency (VE).

The final point regards the Air / Fuel Ratios (AFRs) in the map. As motorbike engines have a small bore and stroke it is difficult trying to keep an AFR value of 12-12.4 or 0.8-0.84 lambda at higher engine speeds. Small changes to the fuel map cause large changes to the AFRs. The engines top end AFRs were around 11.3 AFR which is 0.77 lambda, running this rich decreases the engines overall ability to produce power and torque.

The engine should produce 35 kW (47hp) at 8500 rpm as this is the most efficient running point for the engine due to camshaft lift and profile. However, as the engine was only mapped to 8000 rpm, and considering the three points made above, the engine produced 30.6 kW at 8000 rpm with a maximum of 36.5 Nm of torque at 8000 rpm.

The predictions made for peak power and torque were 30 kW and 35 Nm at 8000 rpm. This prediction was made form the three differences from the Original Equipment Manufactured (OEM) setup and the test as explained above, however another 2-3 kW could possibly be achieved by leaning out the AFR's. This is because N/A engines can run an AFR around 13-13.5 at higher engine rpm, as due to only ambient air entering the engine there is less load being applied onto the pistons.

Tuning for Maximum Torque when Turbocharged

The engine can then be setup to be turbocharged based on the baseline N/A map. Tuning for specific AFR's, the engine should be held at different speeds as well as going through the throttle positions. Varying the load on the engine to keep it at the set speed. Each AFR can then be tuned to achieve Max Best Torque (MBT). Just before boost pressure is reached, the AFR is changed to 12-12.4 to prevent knock at higher speeds.

An engines fuel mixture works hand in hand with its ignition timing, so when the turbocharged engine is tuned, the AFR's and timing will be tuned to MBT. The easiest way to look at mapping a turbocharged engine is to tune for fuel as well as ignition whilst adding boost pressure. Firstly, by increasing the overall amount of fuel injected per stroke by 50% to prevent knock, this can be done by the fuel trims in Motec or inputted manually, then subtracting 2° of timing for every psi of boost pressure added. Next, the over-fuelling is slowly decreased back to 0% until the engine runs smoothly with 0.5 bar of boost pressure.

The engine is not tuned for maximum torque potential at this point, as this is technically a base map with 0.5 bar of boost. For maximum torque the timing will need to be either increased or decreased till the point of maximum torque at any given engine speed seen by the dynamometer, then be retarded by 2°. The reason the timing is backed off is to increase longevity and reliability.

Engine Failure

Going from N/A to Turbocharged brings its difficulties in engine mapping. After results were obtained from the dynamometer the engine was inspected and unfortunately had scoring marks on the cylinders caused by running with high ignition timing. No further testing was therefore possible.

Due to this setback the ETC will not be tested on the dynamometer however, it will be bench tested as well as reviewed on the Ricardo Wave simulation.

Turbocharged Engine results

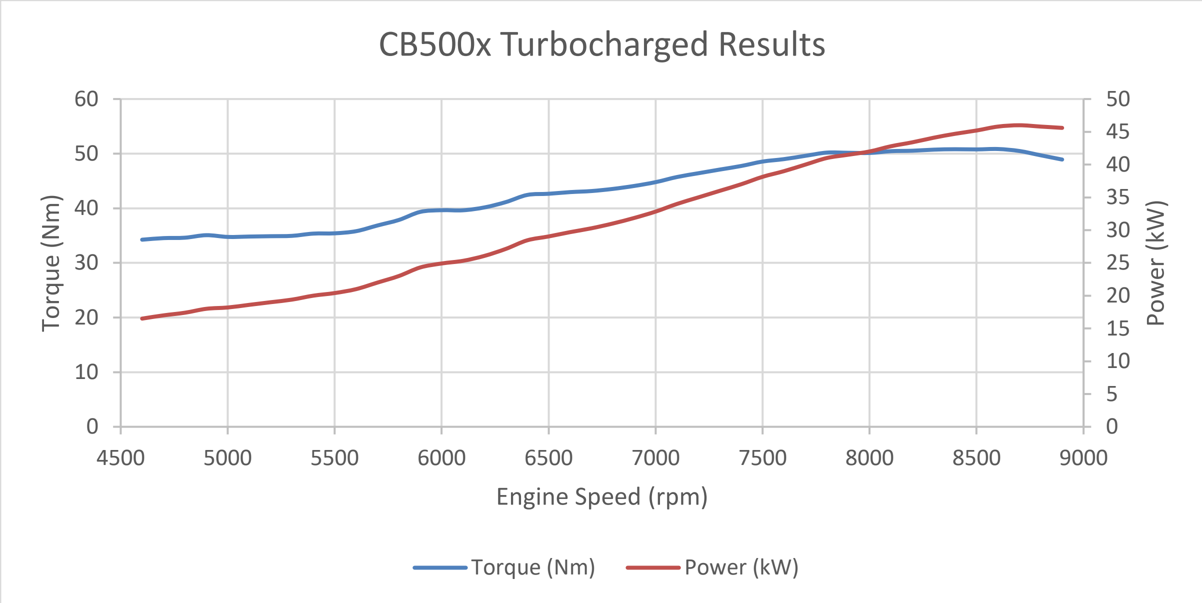


Figure 68 Turbocharged dynamometered results

A prediction was made based on the N/A map that the engine would start to produce boost from 4500-5000 rpm when the engine was limited to 8000 rpm. The turbocharged engine measured results are shown in Figure 68. They show a peak power of 46 kW at 8700 rpm and a maximum torque of 50 Nm from 7800 rpm through to 8800 rpm.

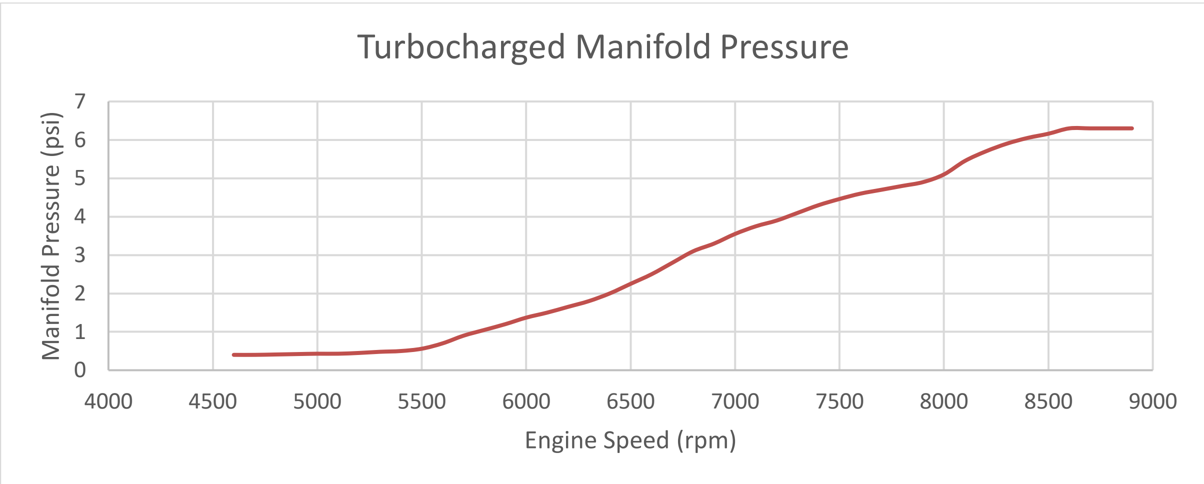


Figure 69 dynamometered results of plenum pressure

Whilst tuning the engine the rpm limit was changed to 8900 rpm this was because the engine stared to produce boost pressure from 5500 rpm represented by Figure 69 showing manifold pressure. At 7000 rpm the boost pressure within the plenum was around 3.6 psi, at this speed range the engine clearly showed a difference in its ability to produce torque. This is due to the fact that the plenum volume needed to be filled with boost pressure has been filled at that point in the speed sweep.

The dynamometer recorded the pressure within the manifold, however a prediction of 1 psi boost pressure drop was made. This is due to the large plenum size with a volume of 3000cc as well as 51 mm diameter piping from the turbocharger to the throttle body.

Plenum size is generally a function of the engine's cubic capacity normally around 50 – 70% (Bell, 1997). Therefore, typically for a 2000cc vehicle the plenum volume could be around 1000 – 1400cc. In testing the CB500x a suitable plenum of 236 – 330cc should have been designed, manufactured and implemented, however, as there was no of the shelf component with this volumetric capacity available, a much larger 3000cc water to air intercooler was used as the plenum.

This plenum was used with the intention of having the shortest piping from the turbocharger outlet to the throttle body, as well as keeping the boosted air pressure charged within the plenum as cool as possible to prevent pre-ignition. However, the time taken to fill the overall volume of the plenum and turbo piping of boost pressure resulted in a large overall turbo lag.

A 50% increase in power is generally expected when an engine is turbocharged with 0.5 bar of boost. From the data obtained, the engine at 8000 rpm has an increase of 42% in power with 0.35 bar of boost pressure, however, the difference in peak power is 53.3%.

Dynamometered Results Comparison

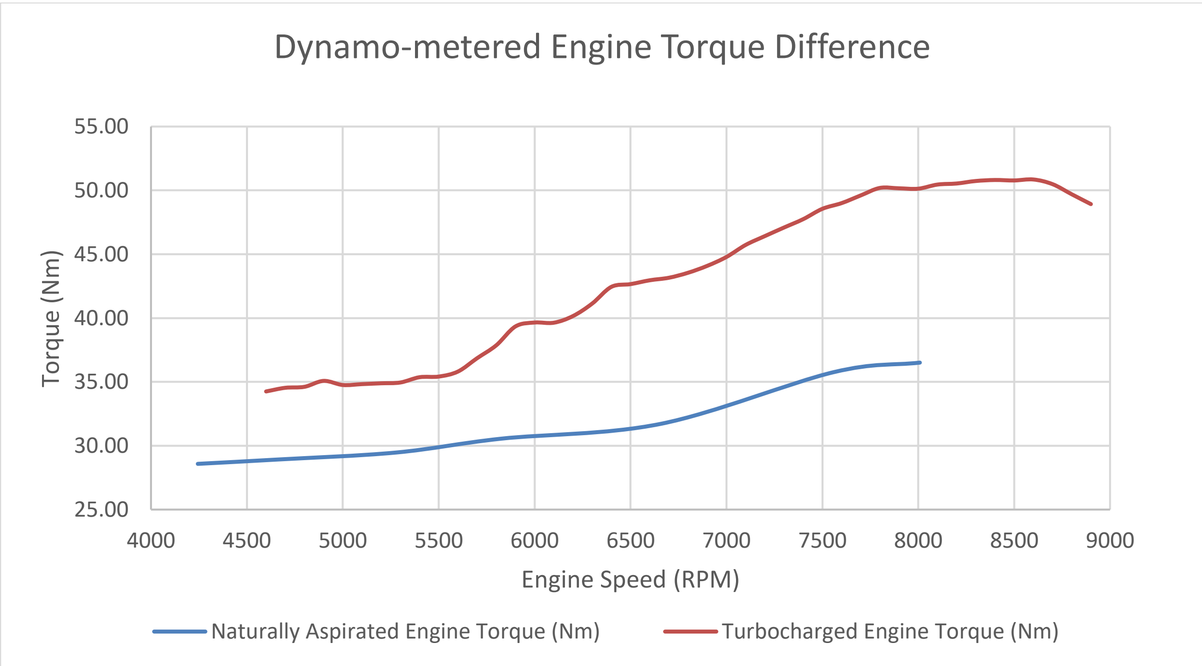


Figure 70 torque difference from N/A to turbochared test results

Figure 70 shows the torque produced from turbocharged and N/A engine on the dynamometer. An increase in torque from 4600-6000 rpm is shown from the turbocharging, however this increase in torque is a result of leaning the AFR to 13.2 compared to the N/A AFR of 12.

A flat torque curve is preferred for greater vehicle control, this is so that the vehicle can be predictable whilst driving. From 5000-8000 rpm the turbocharged engine has a 43% increase in torque whilst the N/A has an increase of 24% within the same rpm range. The percentage difference shows how turbocharged engines have a rise in torque due to boost pressure but, having a greater gradient increase in output torque can make an engine which is unpredictable to drive.

Chapter 8 Simulation, Validation and Results

The simulation was designed assuming the engine would idle at 2000 rpm, as the project looks to improve torque response in the lower engine speed range. Validating the model using the dynamometer power curve, the simulation can be developed further for maximising the engines capabilities to produce torque, power, efficiency, and fuel consumption.

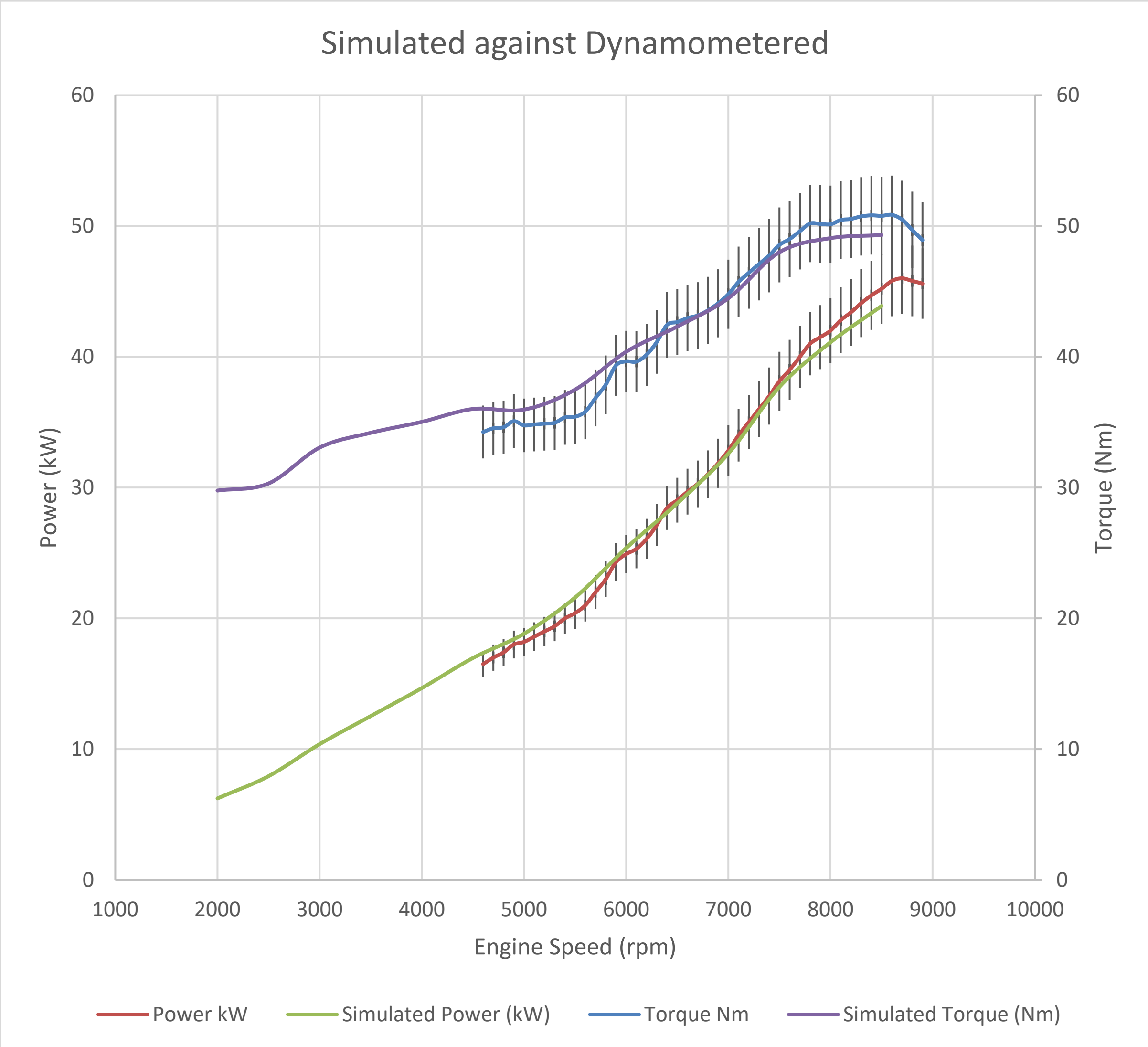


Figure 71 turbocharged simulation results against turbocharged dynamometered results

The simulation can be validated using two measured data parameters. The first being the power curve, the second being the plenum pressure. Figure 71 shows the power and torque from the dynamometered data against the simulated power and torque. Each point has less than 5% error showing the simulation is validated.

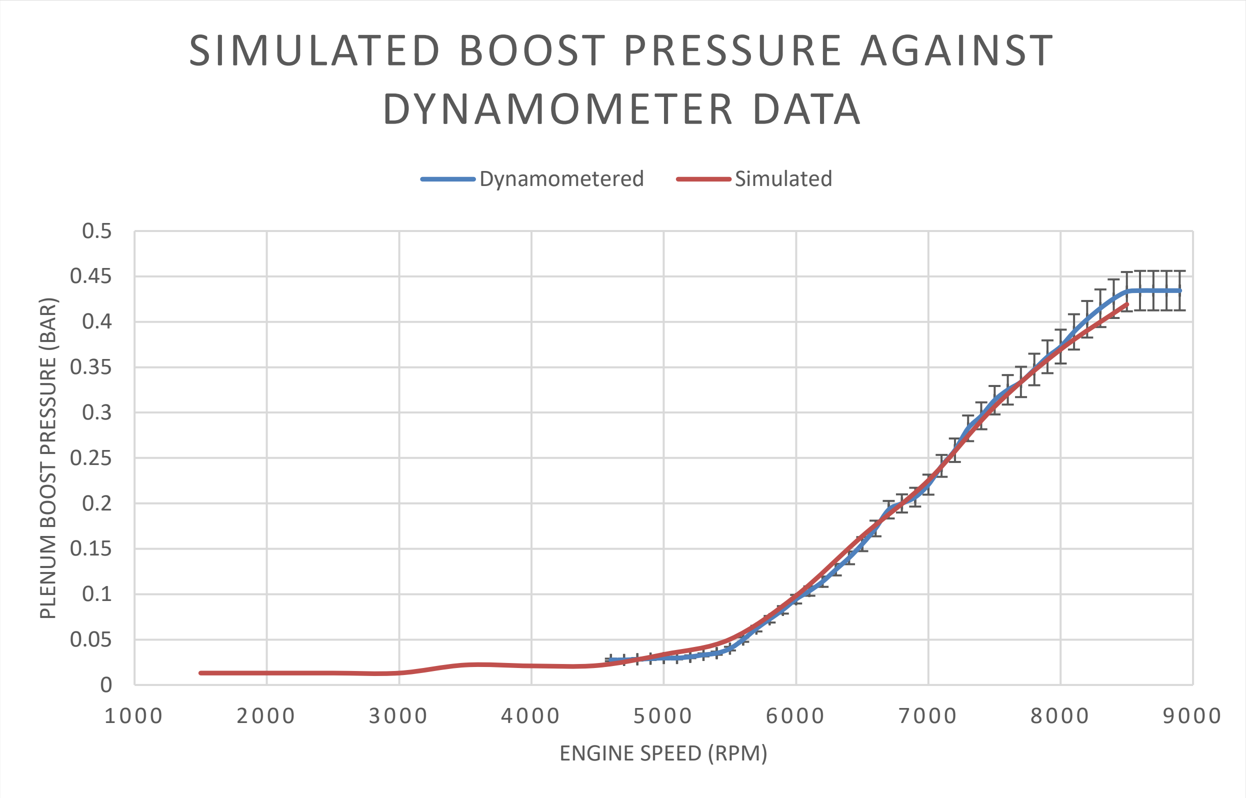


Figure 72 simulated plenum pressure against dynamometered engine plenum pressure

The second measured data parameter was the plenum pressure. Figure 72 represent the simulated boost pressure in the plenum against the dynamometered data of plenum pressure. From simulation and test it cannot be expected to have a 100% match. The error between the simulation and test does shows that most of the data points are within 5%. However, a small between 5100-5500rpm the simulated results are not with in the error margin. This can come down to the turbocharger maps within the simulation as they are idealised and not exact.

Simulation Analysis for Volumetric Efficiency

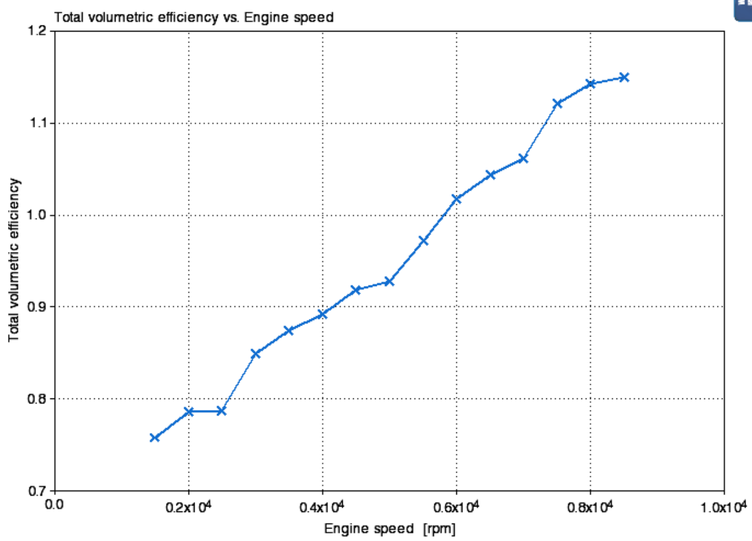


Figure 73 Ricardo Wave Turbocharged Volumetric Efficiency results

One of the most Important parameters of any spark ignition (SI) engine is the volumetric efficiency, as volumetric efficiency has a direct correlation with the output torque that the engine produces (Irimescu, 2010). N/A engines usually have a flat torque curve between a certain range in the power curve, this range is where the camshaft profiles would allow for the maximum air-flow to enter the cylinders via the 'suck' stroke of a four-stroke engine. However, turbocharged engines have more volumetric efficiency than N/A due to the engine being force fed compressed air.

This compressed air has a direct impact on SI engines. Typically, N/A engines have a maximum volumetric efficiency (VE) of 80% of the total volumetric capacity of the engine. Turbocharged engines will pass the 100% VE mark however, this is dictated by the boost pressure fed to the engine. The engine at 8000rpm had a 0.373 bar of boost pressure fed directly into the engine's cylinders, corresponding with the VE, the engine had a total volume of 114% of the total volumetric capacity of the engine. Turbocharging the CB500x gave a 42.5% increase in VE at 8000rpm over the 80% naturally aspirated.

Electric Motor Test and ETC simulation

The Electric motor was attached to the turbocharger and tested. The test was to see the relative mass flow as well as the pressure produced at various throttle positions.

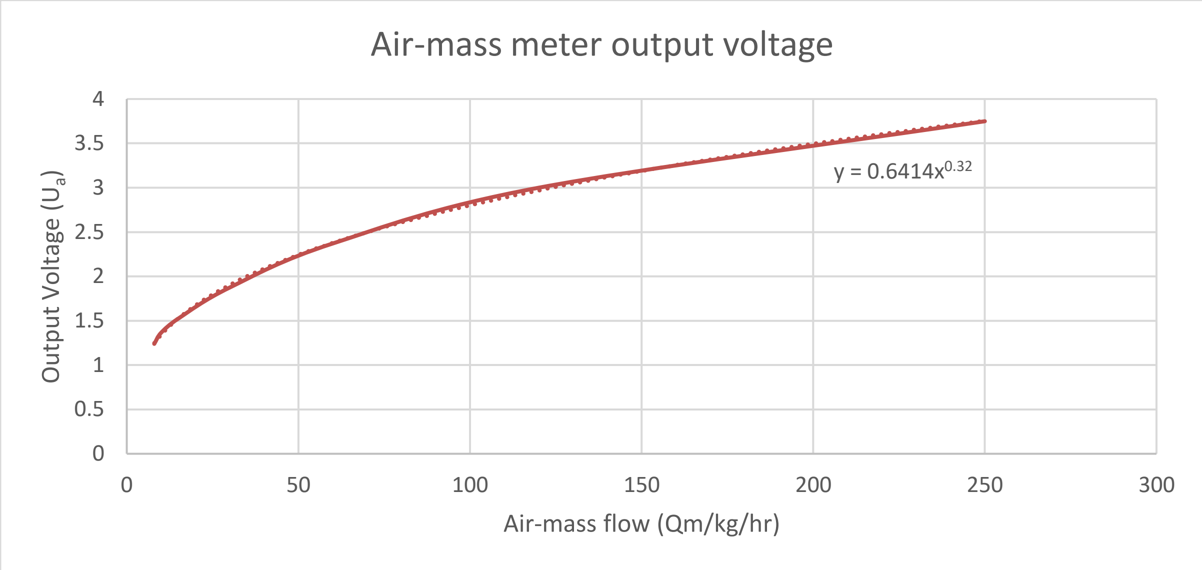


Figure 74 MAF sensor data sheet

Above is the data sheet for the Bosch 0 280 218 019 Mass Air-Flow sensor (MAF). This sensor was used to measure the air-flow of the turbocharger. To detect the mass air-flow, an oscilloscope was used, and the output voltage was read (U_a) and recorded. A pressure transducer was attached to the dynamometer and pressure was also read and recorded.

The results were recorded in voltage from the oscilloscope therefore, an equation was developed in excel from the MAF data sheet. The formula was $y = 0.6414x^{0.32}$ this formula was rearranged to $x = \sqrt[0.32]{y/_{0.6414}}$ from here the voltage readings were placed in to the y of the formula and the x is the air-flow in kg/hr. As the turbocharger compressor map was in pounds per minute the air-flow was converted to these units.

To test the ETC performance a straight pipe was taken from the turbochargers outlet and fitted with the MAF sensor and the throttle body. The throttle body was placed at the end of the two-meter pipe and the MAF sensor was placed a meter into the pipe. This means that if the throttle is opened to 100% the mass flow will read a large number due to the pressurised air passing through the MAF sensor reading its flow rate. However, if the throttle is closed the system therefore act like a pressured container and the turbocharger outlet pipe becomes pressurised.

Throttle position (%)	Air-mass flow (Qm/kg/h)	Air-mass flow (lb/min)	Pressure (Bar)
100	176.820354	6.49703404	0.05
90	176.820354	6.49703404	0.05
80	171.9333968	6.317469153	0.07
70	163.9939307	6.025743791	0.1
60	151.8153538	5.578257815	0.13
50	130.6649215	4.801112674	0.16
40	94.54458982	3.473918045	0.18
30	26.93256463	0.989602075	0.19
20	16.40243886	0.602686293	0.19
10	7.846297664	0.288302007	0.19
0	3.761001445	0.138193109	0.19

Table 6 ETC test results

The data in table 6 sets the range in which the ETC can act in with in the simulation. This means that if the simulation is showing 2.5 bar of gauge boost pressure at 2000 engine rpm the ETC simulation is wrong as the maximum can only be 0.19 bar.

The motor applied full load with a battery voltage of 23.4 V giving the turbocharger a maximum speed of 60,000 rpm however, the turbocharger speed sensor automatically records the data and rounds the speed to the nearest ten thousand rpm, this could mean that the turbocharger could vary from 55-65 krpm.

Losses in the system

The Electric motor without load can reach speeds of up to 75 krpm however, when attached to the turbocharger gave a speed of around 60 krpm due to the inertia of the turbochargers rotating components as well as bearing frictional losses and oil viscosity.

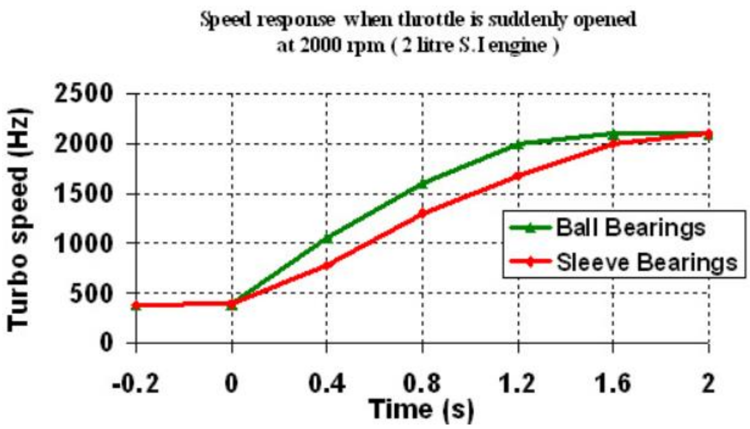


Figure 75 turbocharger bearings time against speed (Honeywell Garrett, 2015)

Firstly, bearings impose frictional losses within the turbochargers system. The turbocharger tested contains journal bearings. Figure 75 shows the difference between ball bearings and conventional turbocharger journal bearings also known as sleeve bearings.
The ball bearing turbocharger here accelerates faster as the ball bearings impose less of a frictional loss compared to sleeve bearing turbochargers. Usually ball bearing turbochargers accelerate up to 15% (Honeywell Garrett, 2015) faster compared to their corresponding sleeve bearing associate.

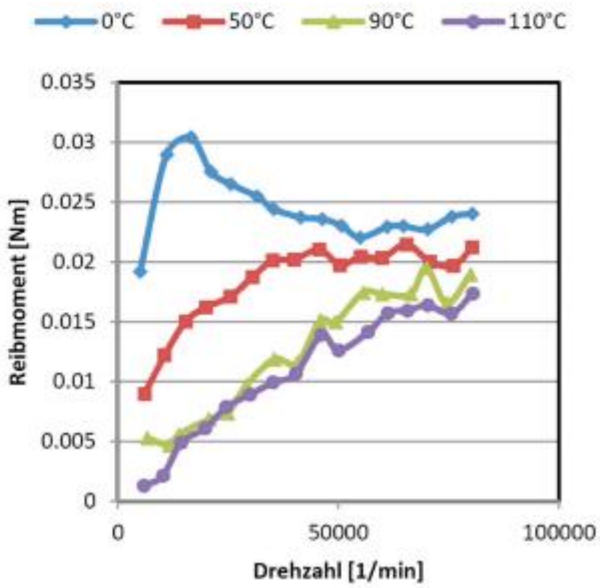


Figure 76 the relationship between oil temperature and frictional torque within turbocharger speed (Vanhaelst, Kheir , & Czajka, 2016)

Figure 76 shows that the oil temperature has a direct impact on friction torque across the turbocharger speed range. The higher the oil temperature, the greater the reduction in frictional torque. At 90$^{\circ c}$ and 110$^{\circ c}$ the temperature has a progressive trendline.
The engine oil in the external system used to supply the turbocharger in the test had an oil temperature at start-up of 25$^{\circ c}$ which was room temperature. As the oil was kept at a very low operating temperature due to a large operating volume, the losses in the system can be dramatic, somewhere between the 0$^{\circ c}$and 50$^{\circ c}$ values in terms of friction torque losses.
As the temperature of the oil used in the turbocharger increases, the viscosity of the oil drops. As the viscosity drops the oil becomes 'thinner' as this happens the oil imposes less of a frictional loss to the system.

ETC Simulated Results

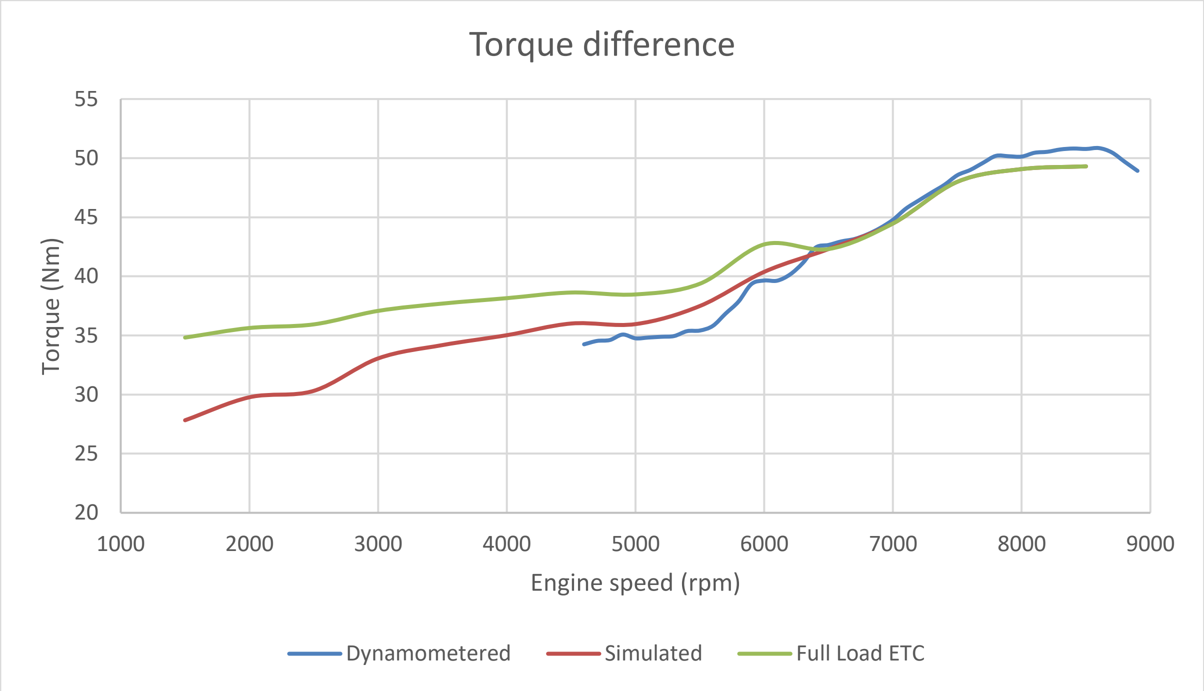


Figure 77 Torque Difference between simulated turbocharged engine, tested turbocharged engine and simulated full load ETC

For analysing the effects that the ETC has upon the engines ability to create torque firstly, the simulated Full Load ETC will be considered as this will have the maximum effect on the turbochargers capacity to produce boost pressure. Figure 77 compares the torque measured in the dynamometered results to the simulated turbocharged results as well as the full load ETC simulated results.

Assuming the engine idles at 1500 rpm the simulated torque value is 27.8 Nm however, the simulated full load ETC torque is 34.8 Nm, that is a 25% increase in torque at engine idle. From the Figure 78 the turbocharger starts to produce boost pressure at 5500 rpm however, the full load ETC produces boost pressure at engine idle. The ETC produces 0.19 bar of boost pressure at engine idle, 1500 rpm. The same amount of pressure is not produced by the turbocharged engine until 6800 rpm.

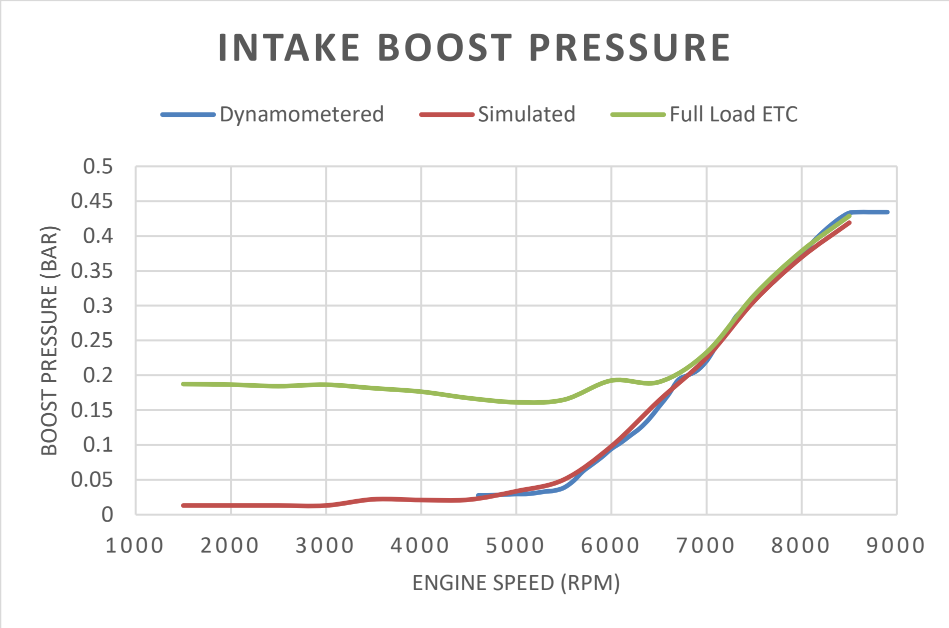


Figure 78 manifold pressure between simulated turbocharged engine, tested turbocharged engine and full load ETC simulation

At 6000 rpm the ETC is switched off, and the engines exhaust gasses take over driving the compressor, hence the torque drops slightly after 6000 rpm. From 3000 rpm to 5500 rpm the torque drops slightly, this is due to the increase in engine pumping.

Figure 79 shows the pressure drop from the turbocharger outlet to the plenum pressure. Initially as the engine is at idle the pressure is balanced from outlet to intake. As the engine speed is increased the ETC continues to produce 1.2 bar of boost pressure however as the pressure is making its way to the engines cylinder head and engine speed is increased, the pressure starts to decrease at the plenum until it reaches 5500 rpm.

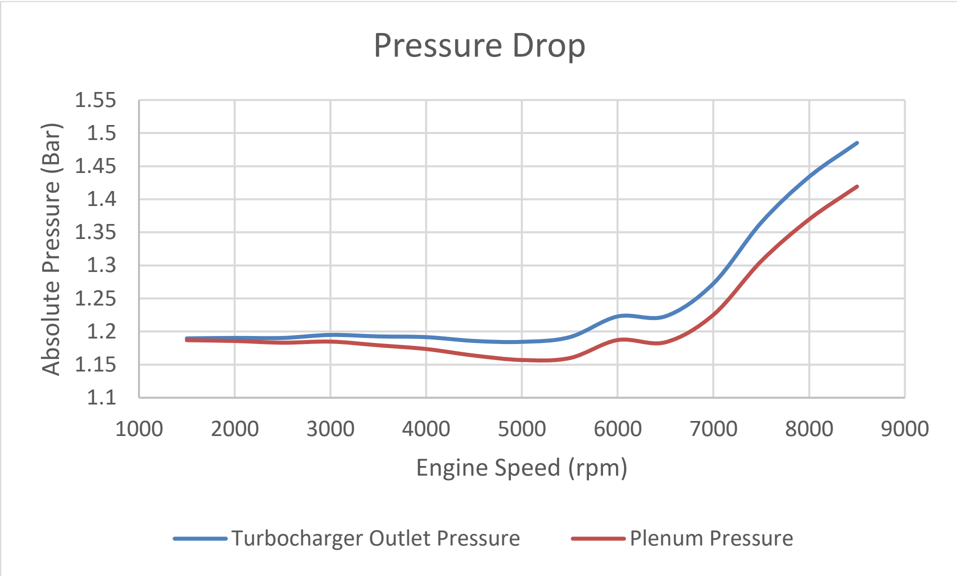


Figure 79 Pressure drop of Full load ETC simulation

At 5500 rpm the engines exhaust gasses take over from the electric motor and the pressure is then increased until 6000 rpm where the ETC is switches off.

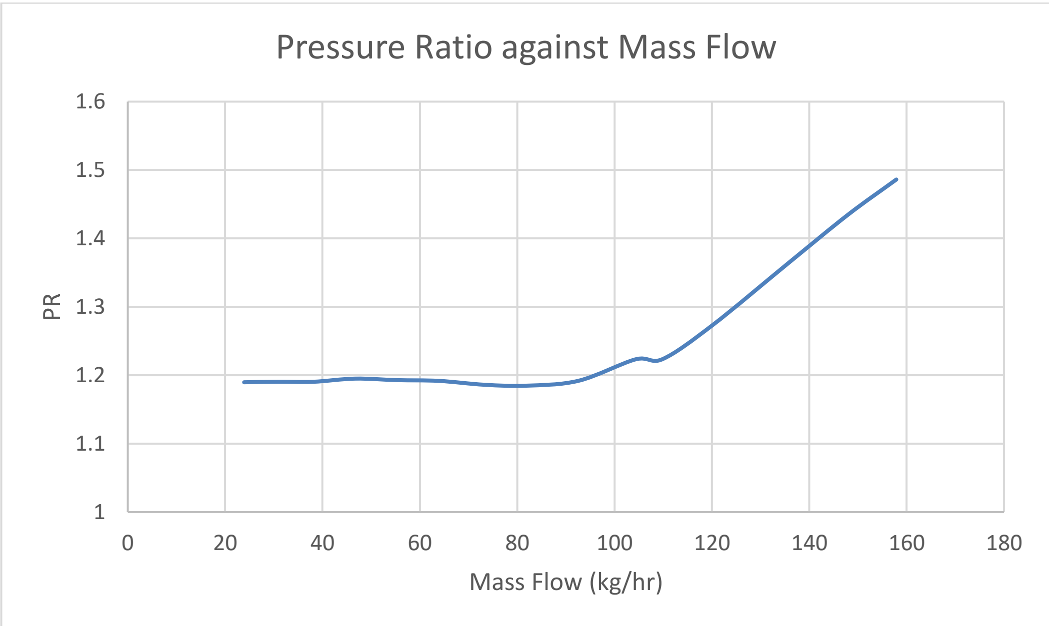


Figure 80 Full Load ETC mass flow and Pressure Ratio simulated results

Figure 80 shows the pressure ratio against the mass flow of the ETC on the simulated CB500x. Above 110 (kg/hr) of mass flow the turbocharger is not assisted by the electric motor. The mass flow and pressure ratio were therefore within the capabilities of the tested ETC.

Torque Comparison

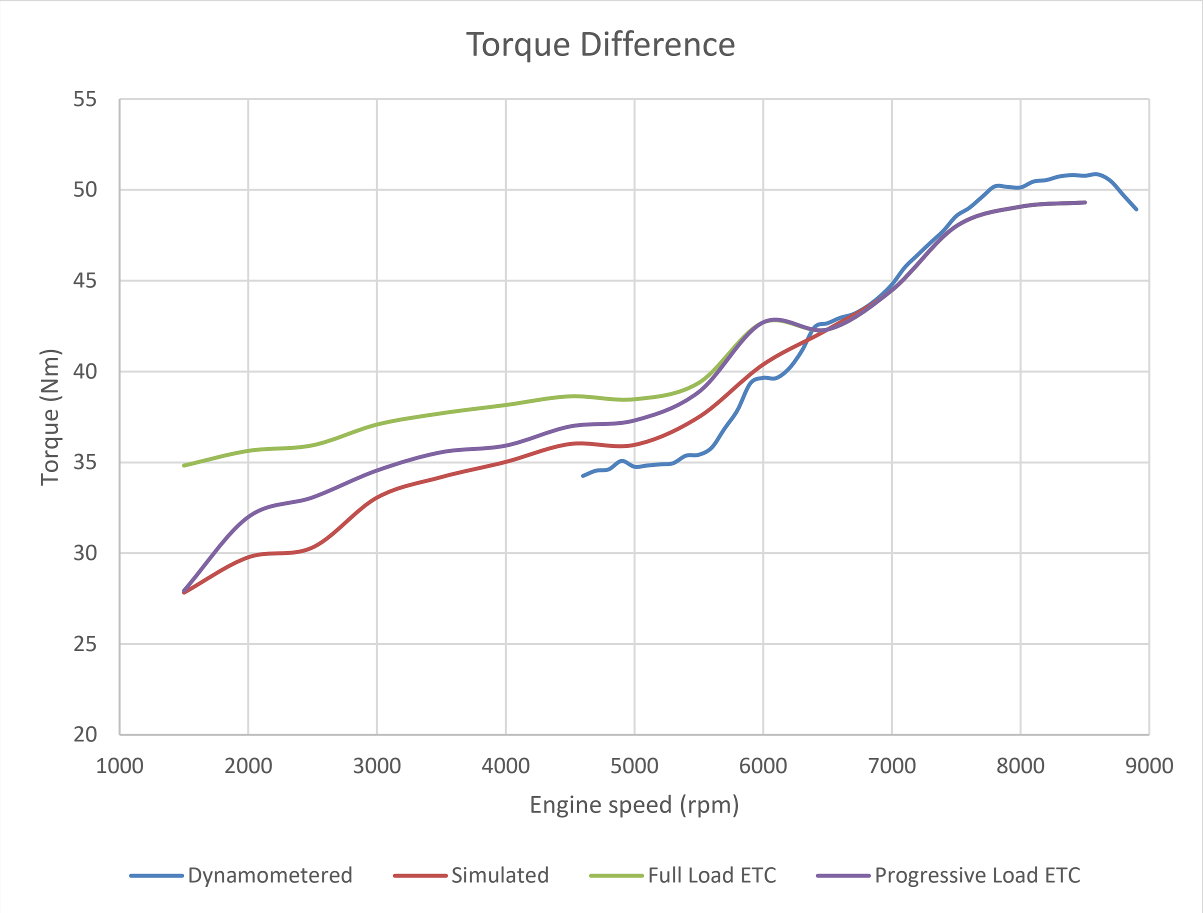


Figure 81 Engine Torque comparison between simulated turbocharged engine, tested turbocharged engine and two forms of ETC load

Figure 81 compares the benefits from the turbocharger through to electric assisted turbocharger with two methods of assist. Predictably, the full load ETC gives the most torque instantly, however the progressive load method acts more similarly to the turbocharged engine at low engine speeds. At 5500 rpm the progressive load ETC behaviour is the same as full load ETC, then turns off at 6000 rpm.

From the torque curves above the most effective is the full load ETC This is due to the turbocharger producing a constant 1.2 bar of absolute pressure at the turbochargers outlet irrespective of engine speed. This increase in boost pressure at the bottom end engine speeds results in greater torque and increased drivability at those speeds.

When the engine is turbocharged the output torque at 3000 rpm is 33 Nm, at 8000 rpm the engine makes 50 Nm, the torque gradient increase over the speed range is 51%. The full load ETC within the same speed range has an increase in torque gradient of only 35%.

This shows that the ETC system can make the turbocharged engine behave more like a N/A engine by flattening the torque curve whilst still giving the top end torque of a turbocharged engine.

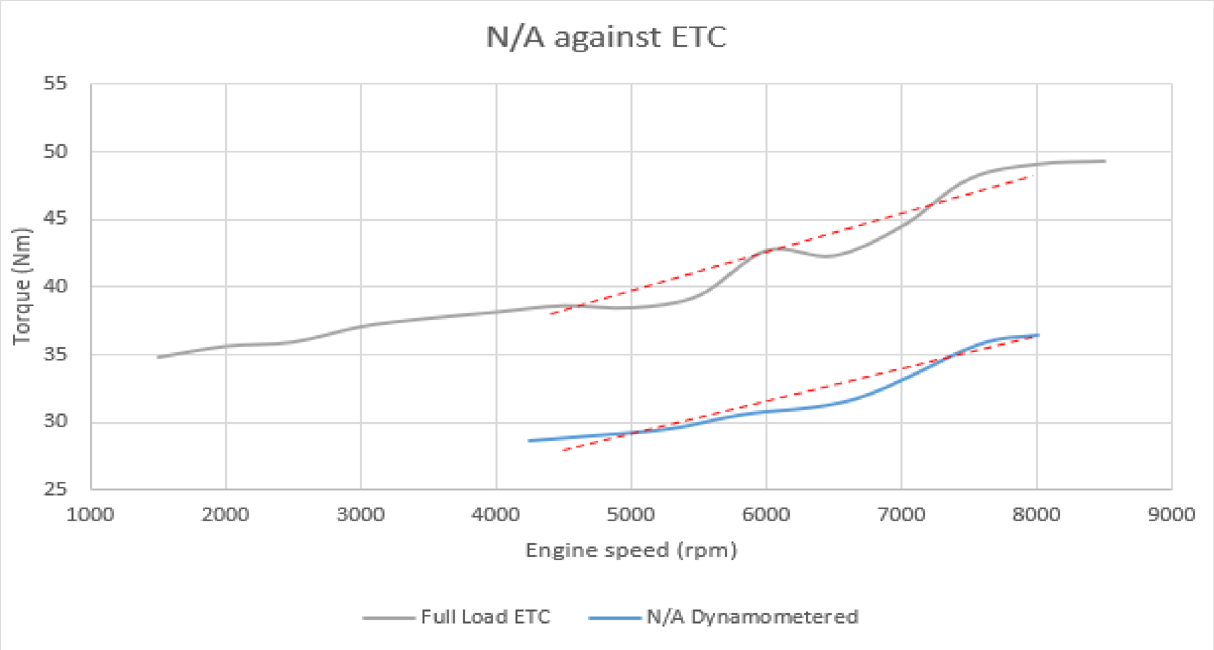


Figure 82 N/A dynamometered engine results against full load ETC simulated results

Figure 82 shows the increase in torque of the full load ETC over the N/A self tuned engine. The increase from N/A to turbocharged is around 24-38%, at 5000 rpm the torque increase is 31% and at 8000 rpm is 34%.
Assuming both the torque curves are straight, represented by the two almost parallel dotted lines, the objective of a turbocharged engine which acts like a N/A engine is achieved. Furthermore, the overall torque has increased from the N/A engine through to the turbocharged engine and upto the ETC Simulated response.

Formula Student restricted 20mm throttle torque response

Assuming the powertrain can potentially be fitted into a Formula Student vehicle, what would be the torque response? To answer this question a 20 mm restrictor is added, simulating the Formula Student powertrain requirement. The turbocharger will have a full load ETC applied till 6000 rpm. The parameters such as AFR and maximum boost pressure will be kept the same as well as the exhaust, intake and piping.

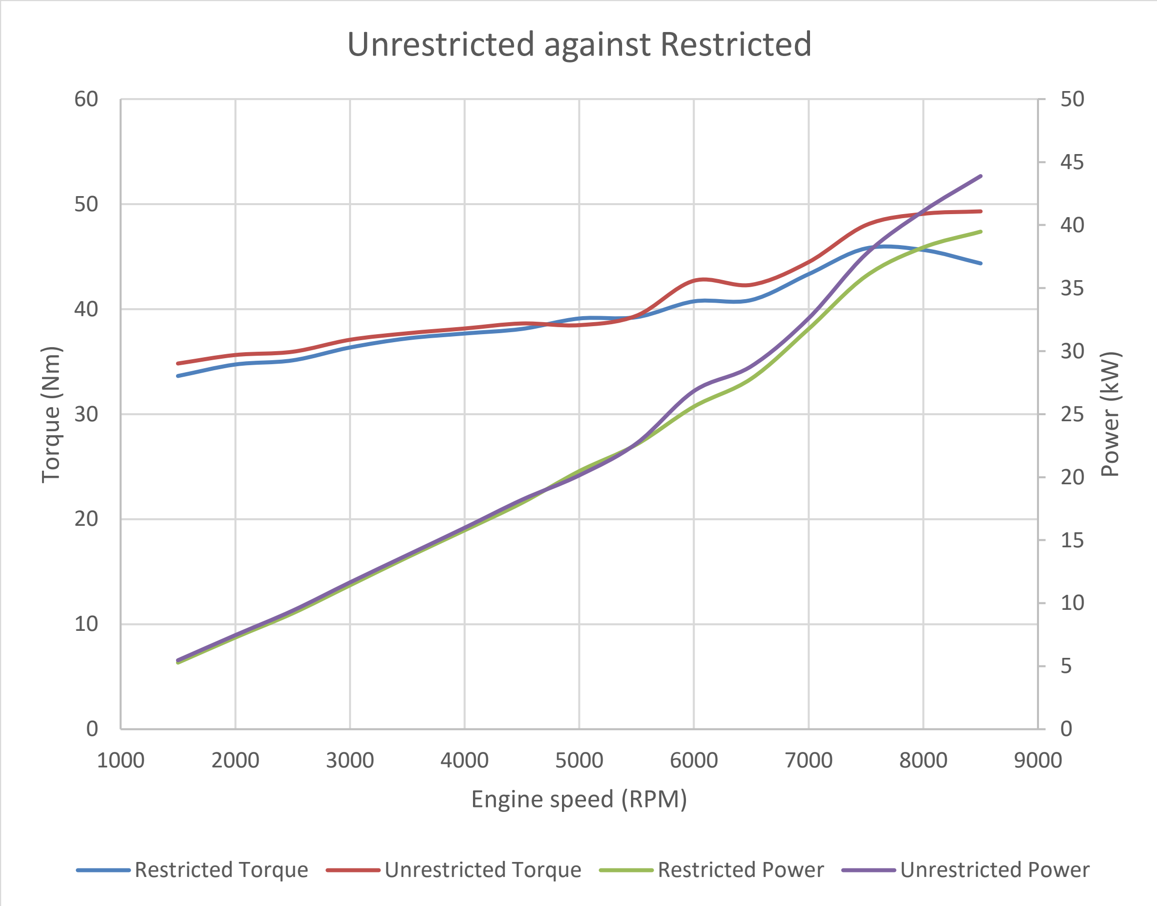


Figure 83 Formula Student 20mm restricted full load ETC performance results

Figure 83 shows the unrestricted against the restricted model. It is clear that the restricted model has a slight disadvantage over the unrestricted model at high engine speeds due to the restrictor 'choking' effectively this is where Mach has reached one. However, a well-tuned turbocharged setup with smaller diameter pipes and a correct sized plenum setup could potentially increase the overall torque at the lower end speed range as well as increase the overall torque delivery and reduce pressure drop within the system.

The restricted response has given a flat torque curve that is desirable in formula student vehicles. A flat torque curve makes a predictable vehicle which is easy to drive. Furthermore, the restricted engine makes a peak power of 40 kW at 8500 rpm and a peak of 45 Nm of torque at 7500 rpm. The output in power and torque is a respectable figure to compete at Formula Student events.

Conclusion

It has been shown through modelling and testing that addition of an ETC to a downsized Formula Student engine can increase the torque delivery of the engine across the speed range, but with the potential advantages of flat and predictable torque curves, whilst also reducing the boost threshold period and turbo-lag periods.

A key aspect within this research is that each test was validated, leading to the simulation results that show less than 5% error between simulation and test.

The aim of this study was 'to design and develop an electrically assisted turbocharger with energy recovery based around the Formula Student rules and regulations, to minimise the boost threshold and increase efficiency.' This has been completed through the use of 1D engine simulation as well as dynamometer performance testing. It was also important to determine if an ETC was viable for small engines, if a correct sized turbocharger is used so that an electric motor can accelerate the turbocharger the engine capacity is not a large factor.

It was important to create a mathematical model of the base powertrain unit to compare improvements against. This was completed through the use of the Ricardo Wave software. The overall measurements which were taken from the engine allowed for a validated engine model which was N/A so that the simulation could be developed further.

The ETC was designed, developed and tested, however the energy recovery aspect was only considered theoretically due to time constraints. This will allow for further development of this work and can be firstly analysed through simulation.

The tested power curves of the dynamometered engine were compared to the simulation, it is feasible to say that the ETC develops more torque low down in the engines speed range.

The added mass flow that is provided through turbocharging has increased the overall torque produced by the N/A engine. This is shown by the Simulations as well as the tested engine. The addition of boosted air at any given engine speed would increase the torque, however other variables have to also be considered such as AFR and ignition timing.

There are benefits of using an ETC however, this study has only considered a small aspect of what an ETC could potentially do. Other studies have shown a reduction in fuel consumption, due to a reduction in engine pumping work. Therefore, the author recommends exploration of further design and performance aspects to have an insightful understanding of an ETC and thereby aiding efficient ICE systems.

Further Work

This report has developed a strong understanding for what an ETC can potentially do to a downsized Formula Student engine however, this research is merely the foundation to the development of an immensely competitive Formula Student powertrain.

Firstly, the simulation should be revised to optimise the intake and exhaust systems, meaning that the plenum, piping and exhaust should be redesigned to maximise the potential of the turbocharged engine. Having a simulation which has maximum error of 5% will also allow investigation of the effect of alternative fuels in order to maximise the torque output of the engine.

Secondly, the powertrain should be redeveloped with the new optimised plenum, piping and exhaust. The simulated results should be matched again and then the ETC can be optimised. The revised ETC should include cooling of the electric motor as when the ETC was tested the motor quickly became very hot and had to have time to cool down before further testing. This issue needs to be overcome for a long term, durable solution to be reached.

Thirdly, a clutch mechanism would need to be designed to allow for the ETC to be disengaged. Small jet engines contain a clutched starter motor which is only engaged when the jet engine needs to be started. However, using the systems design can potentially be adapted with some design changes to be suited for the ETC application.

Fourthly, the Motec m800 ecu can be adapted to work with the electronic speed controller (ETC) that controls the acceleration of the motor. Having an adaptable feature added onto the ecu would allow for the ECU to control the lag of the turbocharger and assist the driver when her or she may need greater driver control or maximum acceleration.

Finally, to see the overall benefits of an ETC the system would have to be tested in an actual Formula Student vehicle against the clock. This would show whether ETC could potentially reduce lap times. Most importantly the benefit of the ETC is to have a turbocharged vehicle which behaves like a normally aspirate one, therefore gathering driver feedback would be essential.

Appendices

Error analysis

When measuring anything there is always an uncertainty known as error. For example, the dynamometer used in the testing was a Sierra CP chassis dynamometer, the accuracy of the dynamometer itself is ±5% (Sierra CP Engineering, 2018) when reading. Therefore, if a known engine is running onto the dynamo and produces 55Nm of torque at 6000RPM but the dynamometer reads 56.3Nm of torque at 6000RPM the error can be calculated.

$$55(Nm) \times 1.05(\%) = 57.75(Nm)\ upper\ limit$$

$$55(Nm) \times 0.95(\%) = 52.25(Nm)\ lower\ limit$$

If the recorded torque of 56.3Nm it falls within the upper and lower limit of ±5% that is the known accuracy. Hence the engine which is ran on the dynamo is classed as pass as it falls within the error range.

This project aims to test the powertrain on the dynamo therefore, the given error of the system is ±5% this value will be used when comparing the simulation to the test.

Test Plan

Introduction and overview

The powertrain will be tested in two stages. The purpose of testing the powertrain is mainly focused on the development of torque produced with a turbocharger before and after ETC assist. This is because the objectives of the project are to visually see the difference in torque with and without the ETC system, looking at the development of torque and its increase with the ETC if there is any.

Stage one, the engine will be tuned to produce 0.5 bar of boost pressure from the turbocharger. Then the powertrain will be set to closed looped so that the AFR will be easily imported into the engine simulation. Recorded data will be saved such as power produced, torque produced, compressor speed, engine temperatures and air temperatures from the compressor to after cooling. Furthermore, the test results will be placed into the Ricardo model and simulated to verify the simulation

Stage two, the second stage of this project will be to see in the ETC will make a difference to the transient response of the turbocharger and to visually represent the difference in torque produced with the ETC. Furthermore, data from the testing shown in stage one will be compared. If possible, the recorded data will also be placed into the Ricardo model to simulate the response from the ETC with a Formula Student restrictor.

Approach, Strategy and Test deliverables

The first tests will consist of tuning the powertrain without the ETC. Therefore, the first stage to testing will consist of three repeated runs for consistent dynamometer results with closed loop on. These results will be taken from a five-minute warm up period of the engine. The warm up period will consist of having the powertrain at engine idle with 5% open throttle before the powertrain is loaded for a power run.

The repeated runs will be to allow the powertrain to be at a stabile temperature range before the powertrain is full load dyno pulled for results. Then the powertrain will be shut off to cool down before the test is redone. This in turn would give consistent dynamometer results as well as consistent temperatures, this will ensure consistency across all test runs. Additionally, the test bed will be used to cool the engines temperatures as well as the boost temperature within the air to water intercooler/plenum.

As the motor can only be on top speed and power for ten seconds without losses, there will have to be several test methods to test the benefits behind the ETC. The ECU will be set to close loop for the ETC testing. The reasoning behind this is due to the turbocharger possibly producing boost further down the RPM range the AFR will alter.

Close looping the ECU will alter the fuelling and ignition maps to keep a steady AFR so that the powertrain is not damaged due to running lean. This in turn will shorten the testing operations of the project as the engine will only have to be tuned once. Additionally, the top RPM of the powertrain will be limited to a determined max rpm this will be to ensure the motor is not over speeded this will intern damage the motor as it will start to produce electricity. The following two tests are for stage two of the test plan.

Test 1) This test will be to full load the electronic motor at idle before the engine is dyno pulled, the term 'dyno pull' refers to the engine being on the dynamometer so that it can producing a full power curve, this test will be to visually represent data of the compressor speed throughout the rpm range of the engine to compare with the powertrain without the ETC.

Test 2) The second test will be to slowly introduce the load onto the motor as it is being dyno pulled this will be to compare against test one as, the test data will show the power and torque produced by the electronic motor being on full load at engine idle.

Both test results will be compared to one another this will be to see the advantages and disadvantages of both tests. Results and visually representing the difference in torque developed and torque efficiency by both tests will sufficiently represent the superior test due to its performance.

Test environment

The testing will take place within the university engine test labs however the open dynamometer will be used therefore for the second stage of testing will require a new risk assessment for the prototype ETC turbocharger.

Remaining test task

The final stage of testing is to simulate both turbocharged and ETC recorded data into the Ricardo simulation. The simulation can have a 5% error margin to pass however, any more than 5% will be classed as a failure to this stage of testing.

Schedule

Week 1-2 – In this week time the powertrain will be checked. These checks will review TDC timing and several other parameters on the ECU, when the parameters checks have been made the powertrain will be 'broken in' this break in process is known as a mechanical run-in it ensures the new engines internal components have been heated and work correctly. Then the engine will be tuned. After tuning the powertrain will be set to close loop and several runs will be made for data purposes.

Week 3 – In this week the turbocharger will be modified to accommodate for the electronic motor. Several safety features will be added to the powertrain, corresponding risk assessments will be placed in.

Week 4 – The first testing of the ETC system will take place with this week. Then the powertrain will be set to close loop and several runs will be made for data purposes.

Week 5 – The second stage of the ETC system will be taken trough out this week. Then the powertrain will be set to close loop and several runs will be made for data purposes.

Risks and Contingencies

The risk assessments for the dynamometer are in the appendix of this report.

Approval

Approval from John Allport (Project Supervisor) and Martin Gargett (Technician) for the test schedule and risk assessments will allow the powertrain to be tested.

ECU Setup

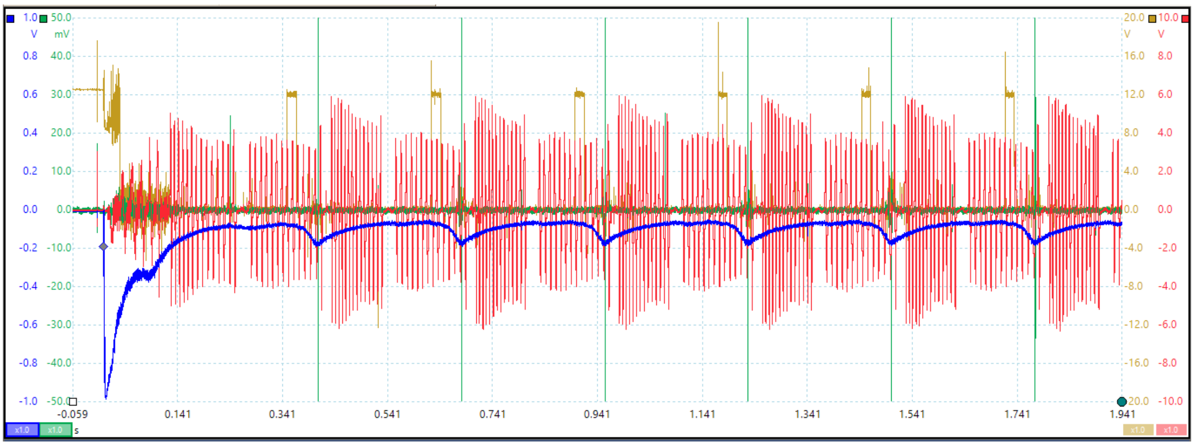


Figure 84 Pico scope results to find TDC

Above is a Pico scope reading shown in Figure 84, a Pico scope is an instrumentation tool used in this case to find the angle of TDC for cylinder 1. This was done by firstly setting the ignition map to 0° throughout the map. As well as removing the sparkplug and coil pack from cylinder 2, this was so that the reading for the current drawn by the starter motor was only for cylinder 1. As the firing order for the engine is 180° apart. To set the parameters into the ECU only the TDC for cylinder 1 in required. Hence, the Crank Index Position (CRIP) value could be placed into the Motec ECU REF Sync setup, so that the ECU would send spark at the exact TDC mark.

The blue line indicates the current drawn from the starter motor showing peaks when cylinder number 1 is at TDC, the red lines shows the tooth count from the crank teeth via a magnetic sensor. The yellow lines indicate the camshaft sensor position using a hall sensor and finally the green lines indicate the coil on plugs sending the spark to the spark plugs.

As the ignition map was set to 0° it would send spark at the value that was set by the CRIP. As the Motec ECU could not pick up on 16 teeth with 3 missing on the crankshaft sensor an additional tooth was added, making 16 teeth with two missing. However, as there are 16 teeth the resolution was low, the ECU was not able to synchronise with the teeth count. Consequently, a hall sensor was added to the camshaft this can be placed between ±180° from TDC to allow for the ECU to synchronise with the sensors and teeth count.

At this point the ECU synchronised and sent spark however the CRIP value was incorrect, the ECU send spark at the wrong time therefore, assumptions can be made to allow for a closer value for the CRIP. Assuming TDC for cylinder 1 occurs around the 12^{th} tooth from the starter motors current drawn peek, after the missing tooth from the red line trace. 360°/ 16 teeth = 22.5° this calculation gives the spacing for the individual teeth for the crankshaft sensor, 16 teeth all 22.5° evenly spaced from each other's rising edge.

Therefore, if the peek current of the starter motor is at the 12th tooth 12 x 22.5 = 270° from the index tooth or first tooth to the 12th tooth, as the Honda is a four-stroke engine a cycle is 720° therefore 360° is added to the 270° for the 12th tooth count to achieve a CRIP value of 630°, however this is an estimation based on assumptions. Therefore, using the Pico scope and trial and error the actual value for the CRIP is 637°.

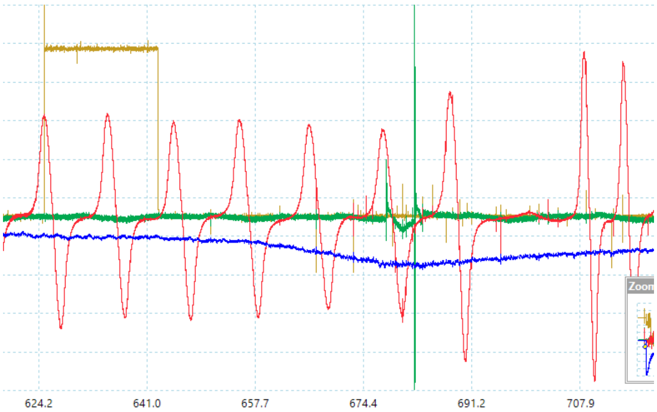


Figure 85 calculated CRIP value of 637° showing TDC for Cylinder 1

This is a close-up screen shot of the four parameters. Showing when the peak current is drawn from the starter motor for cylinder 1 the ECU sends spark at exactly TDC.

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