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Final Report

Group Project - M5X00400

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Abstract

This project is on the development of the Ford 1500cc EcoBoost engine that will be fitted to a ZenTrack chassis which John Avery is manufacturing at Performance Car Club. John Avery's plan is to offer the Ford 1500cc EcoBoost ICE into both his road going and racing vehicles. All the units in ZenTrack's vehicles are going to have to be compliant with the Euro 6 emission standard in order to be road worthy. Given that the automotive world is always looking to move forward with engine efficiency and emissions control, resulting in the older generation engines becoming obsolete, John Avery decided as the company is new, they should develop alongside new technology.

With this in mind, we knew the engine enhancements would have to be reliable and safe, efficient and repeatable, but most importantly economical and emission friendly, all the while achieving a consistent power output of 185Kw (250BPH). Given that little information is known about development of this engine, it was decided that the group will investigate the following areas to achieve what was required; Cylinder-Head development, charge air cooling enhancements and development of the fuelling system in conjunction with ECU tuning.

The project's success is going to be based on matching or perhaps even exceeding this target.

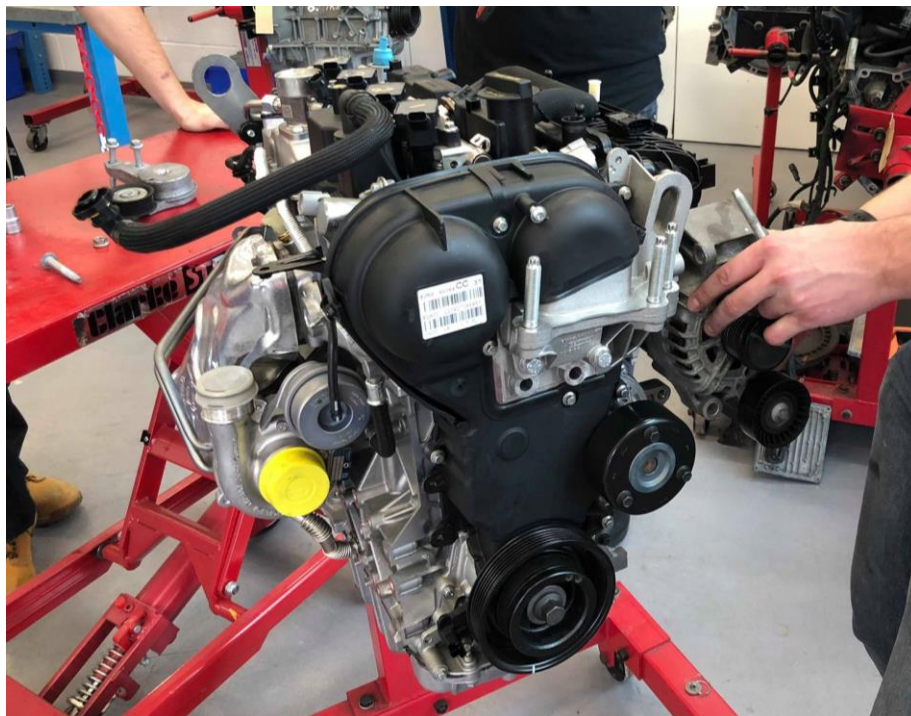


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1. Introduction

The EcoBoost internal combustion engine created by Ford Motor Company is their flagship engine for both increased power, efficiency and economy and this engine design has been a forefront in development throughout the motoring industry. The engine was designed due to ever changing emissions regulation becoming more stringent, along with the extremely competitive market it had to be utilising “every drop of fuel” in turn resulting in a reduction of CO₂ and NO_x emissions and hence the name “Eco”. Then for the “Boost” part, gaining great torque ranges and peak power outputs through the utilisation of both BorgWarner and continental turbochargers, helps to recycle waste exhaust gasses, bringing the finished product to an engine specification that had far exceeded that of their competitors.

All of the Ford EcoBoost ICE's are direct injection (GDI) meaning that although the designs have to be more intricate for the inlet manifold and the inlet ports, the rewards in terms of combustion and burn efficiency far exceed that of the costs to manufacture. Direct injection systems allow the control module to be delivering very finely atomised fuel straight into the combustion chamber and because the high-pressure fuel pump is driven off the camshaft, the pressure in the system goes mechanically hand in hand with the engines speeds, which is a brilliant way to control the amount of fuel for optimisation of emissions and power curves. This system joint with a low inertia turbocharger and variable valve timing (VVT), means the engines are able to deliver a “great spread of Torque” throughout the power cycle. Because of this, they've gained diesel like torque figures at lower end engine speeds, which is why the engine is such a desire in the automotive world because their vehicles are ‘driving like diesels’ with the emissions and power of a petrol engine.

The EcoBoost engines are found throughout a variety of vehicular models offered by Ford, and capacity ranges from a small 1.0 litre, three cylinders setup, to a 3.5 litre with six cylinders. Since launching in 2012, the EcoBoost engine has been considered undefeated in its category (Ford Automotive, 2019). Because of this their engines have been used in various track and road cars through the years with a vast range of various companies now offering a variety of Tuning and aftermarket parts. However, little to no developments have been carried out on the 1.5 litre engine that this project is based on because it's considered as the more ‘economical’ sister ICE of the 1.6 litre option that has been rewound in the motorsport world.

The group was approached by the company ZenTrack and asked to help develop and enhance the 1.5 litre engine so it could be offered in the smaller base model offered by the company. As ZenTrack has development underway on the track spec 2.0 litre and 2.3 litre options already, the task was set to engineer a more economical, road worthy and street legal option, giving the

potential buyer the freedom of choice to use the vehicle as a road going 'kit-car', as well as a track day racer.

This meant that the engine had to be run and benchmarked in order to obtain data, and therefore open up the options for potential tuning, as well as showing the restrictions of engine as well.

The outline of this project and paper is to gain an understanding of the 1.5 litre Ford Motor Company 'EcoBoost' ICE, in regard to improving performance and efficiency in a varied range of specified areas, whilst doing so in a reliable and repeatable manor. To do this, the Team was split into four areas, allowing for increased focus and development within them. The four main project areas were decided as; parts acquisition and engine control module calibration, dynamometer trolley and dynamometer fitment, Thermodynamics and cooling, and cylinder head development. For the most part, each of these areas were able to run independently of one another, however there were times that groups had to coordinate with one another appropriately and efficiently.

The aims of this project were:

- To get the engine up and running on the Dynamometer, in order to do this, parts were required, and a trolley and rig were needed in order to attach the engine, and have it running.
- Once running, the engine was to be benchmarked, and the sensors read to gain an understanding into how the engine performed from standard or at least the base calibration of the SCS GDI4 engine control module.
- The second stage was to increase the performance of the engine by manufacturing a tuning 'package', that was to be shared with ZenTrack, whilst running on the dynamometer because there was no complete vehicle to use.
- Developing the cooling and thermodynamic characteristics of the engine and complete cooling system. This was to be achieved by using an externally mounted water-cooled charge cooler, as well as deciding on pipe work placement, material and sizing that would work effective within the chassis.
- Enhancing the cylinder head, via flow bench testing and computational fluid dynamics analysis. From the data gathered, possible development areas would be discussed, including potential upgrades to the exhaust and inlet pipe-work.

2. Project Organisation – (Roles)

Once the main objects to the project had been outlined, it appeared necessary to split the group into four Pairs, with each contributing to a chosen section of the project, and its associated goals. The four pairs generated were;

Table 1 - Project Organisation.

Group	Members
Part Acquisition and Engine calibration	Stuart Harries Danny Richardson
Trolley and Dynamometer Fitment	Mike Anderson Callum Hill
Engine Cooling and Thermodynamics	Joshua Bass Ifan Hopkins
Cylinder Head Development	Carl-Peter Englebrecht Aidan Chapman

Within the group, each pair came up with their own associated goals and objectives, with the main task of increasing performance in their given area. Alongside this, each group member was given their own individual role to perform within the group as a whole, correlating with their chosen team.

Table 2 - Project roles.

Role	Team Member
Project Manager	Danny Richardson
Technician/Meetings co-ordinator	Stuart Harries
Thermal Management	Joshua Bass
Head Researcher	Ifan Hopkins
Head Designer/manufacture specialist	Mike Anderson
Dynamometer technician	Callum Hill
Workshop Co-ordinator	Carl Peter Englebrecht
CFD Technician	Aidan Chapman

These Roles were set, with paired tasks in mind, to help each member deliver the most effective contributions towards the team and the work required. This would allow for increased efficiency in task completion.

3. Summary of Work Completed

Danny Richardson

3.1 Engine Development & Calibration

3.1.1 Work Content

The work discussed here and throughout this document is about how as a group we decided the best way to enhance an engine that was somewhat exclusively designed for road going, low emission purposes and not for high revving, high power applications. But, because the EcoBoost engine is a brilliant platform and is a pioneer of sorts within the automotive industry, it certainly had potential.

The main area of concern for the project was engine calibration and part modification because this is always the go to point for developing an engine with respect to gaining more power and because manufactures now don't necessarily want the 'public' disrupting and or replicating their software, the engine control module wouldn't ever allow enough access to achieve what had been asked of us. So, the project partner John Avery choose to use an ECU that had been created by 'SCS Delta', a company renowned in their field and have worked with companies such as BAC, Power Torque UK and many other racing teams and small brand companies. Therefore, the engine calibration aspect became a very large area of focus for us because we know that it was now a very open and free area of development that could most definitely be exploited, whilst having the help (If required) from SCS themselves.

With the area of development being largely through the use of the DSG Schenck WS230 dynamometer, we needed to understand how to use the equipment and have the capability to build up a series of testing cycles so that we could effectively tune the engine without having the engine being put through real world testing.

3.1.2 Research

With most of this type of work being new to all involved in the project, the use of the dynamometer for testing and development was a key area to understand, but most importantly it was about knowing how to enhance the engine map on the control module to ensure both

safety and protection of the engine while actually achieving the goal. Therefore, looking at how to use SCS's software package [SXTune] was a starting point. Next was taking a mathematical look at the situation for predictions of things like; BMEP/IMEP, peak ranges, peak values, required AFR and flow rates. Finally, it was understanding the impacts of advancements within the tabulated values for; A/F ratio, ignition timing, cylinder trim values and any closed loop system changes would do to the engine.

Regarding the software side of the research, it does pose a difficult task when there is no data or ECU to work with, but SCS do have an impressive help guide to walk the user through the basic process of connecting to the ECU and how to navigate around the programme to return what you need, whilst not potentially damaging the system or the engine from accidental file removal or disconnection. The guidance notes firstly go through how to connect the ECU with the software and how to correctly save files with the right names, file types and destinations. But it also explains how to save original calibrations and loading in a 'pre-made' map. Next is basic calibration of the sensory devices on the engine such as throttle position and wideband lambda sensors. Finally, the base cycle mapping instructions to obtain the maximum extremes of; fuel cuts and correction factors, boost pressures and ignition advances for both open and closed loop system operation. Furthermore, how to then begin inputting values across the rest of the table while being within safe regions of the engine's physical limits.

For the mathematical approach, it was primarily to understand what values we would be looking to achieve and perhaps what would be required to do so.

Table 3 - Known Engine data (1500cc).

Engine specification		
Bore	79.02	mm
Stroke	76.4	mm
Displacement	1499	cm ³
Compression ratio	10.0:1	
Cylinders	4	Inline
Rod Length	134.00	mm
Crank Radius	23.5	mm
Stock	@6000 RPM	

Max Power	134	Kw
Stock	1600-5000 RPM	
Max Torque	240	Nm
Other		
DOHC - DVCT	4 valves per cylinder	
Cylinder Head	Aluminium	
Cylinder Block	Aluminium	
Oil Capacity (With filter)	4.1	Litres
Coolant Capacity (Stock)	5.5	Litres

- Swept Volume (cm³) – $V_s = \frac{\pi \cdot d^2 L}{1000}$ – $V_s = 374.68$ (cm³)
 - V_s – Swept volume
 - d – Diameter (m) [Bore]
 - L – Length (m) [Stroke]
- Cylinder Volume (cm³) - $V_d = V_s + V_c - V = 416.3080$ (cm³)
 - V_d – Cylinder volume
 - V_s – Swept volume
 - V_c – Clearance volume
- Compression ratio – $CR = \frac{V}{V_c}$ – $CR = 10.0:1$
- Clearance volume (cm³) – $V_c = \frac{V}{CR} - V_s = 41.6308$ (cm³)
- Capacity (cm³) - $cc = V_s \cdot C^n - V_s = 1499$ (cm³)
 - C^n – Number of cylinders
- Density - $\rho = \frac{P_a}{R \cdot T_a} - \rho = 2.2423$ (Kg/m³)
 - P_a – Pressure under 15 (psi) of boost
 - R – 287.05 (J/Kg·k)
 - T_a - Temperature assumed at 45°C
- Volume of air in intake port – using SolidWorks to equate a rough estimation of the volume.
 - Volume in one port is 125.4362 (cm³)
 - There are two ports per cylinder

- Mass of air after throttle & before valve – $M_a = \rho \cdot V_a - M_a = 10.4287e^{-3}$ (Kg)
 - Assuming density at 45°C
 - V_1 - Estimated inlet volume from SolidWorks – 125.4362 (cm³)
 - V_2 - Volume within the airbox – 4400 (cm³)
- Mass air flow - $\dot{M} = \frac{M_a \cdot N_e}{n_r} 2879$
 - @6000 RPM = 0.5214 (Kg/s)
 - @5000 RPM = 0.4345 (Kg/s)
 - @4000 RPM = 0.3476 (Kg/s)
 - @3000 RPM = 0.2607 (Kg/s)
 - @2500 RPM = 0.2173 (Kg/s)
 - @2000 RPM = 0.1738 (Kg/s)
 - @1500 RPM = 0.1304 (Kg/s)
 - @1000 RPM = 0.0869 (Kg/s)
 - M_a - Mass of air – 10.4287e⁻³ (Kg/s)
 - N_e - rotational speed (rev/s)
 - n_r - Crankshaft rotations for full cycle – (4 Stroke = 2)
- Ideal Gas law – Absolute pressure at intake port - $P = \frac{m \cdot R \cdot T}{V} - P = 29.68$ (psi)
 - M_a - Mass of air - 10.4287e⁻³ (Kg/s)
 - Temperature assumed to be 45°C
 - R – 287.05 (J/Kg·k)
- Volumetric Efficiency - $\eta_e = \frac{\dot{M} \cdot n_r}{\rho_a \cdot V_d \cdot N_e}$
 - η_e @6000 RPM = 93.09 (%)
 - η_e @5000 RPM = 93.10 (%)
 - η_e @4000 RPM = 93.10 (%)
 - η_e @3000 RPM = 93.11 (%)
 - η_e @2500 RPM = 93.07 (%)
 - η_e @2000 RPM = 93.07 (%)
 - η_e @1500 RPM = 92.73 (%)
 - η_e @1000 RPM = 93.07 (%)
- Fuel flow rate - $F_r = \frac{1}{\frac{14.7}{M}}$
 - Assuming it's at stoichiometric 14.7:1 (A/F)
- Brake Thermal Efficiency (%) – $\eta_{bth} = \frac{Brake\ Power}{Fuel\ Flow \cdot Calorific\ value} = 49.49$ (%)
 - Brake power - 134 (Kw)

- Fuel flow rate $7.0944e^{-4}$ – Kg/s
- Calorific value of petrol 45.8 – MJ/Kg
- Brake Thermal Efficiency (%) – $\eta_{bth} = \frac{Brake\ Power}{Fuel\ Flow \cdot Calorific\ value} = 56.94\%$
 - Brake power - 185 (Kw)
 - Fuel flow rate $7.0944e^{-4}$ – Kg/s
 - Calorific value of petrol 45.8 – MJ/Kg
- Brake Mean Effective Pressure - $BMEP = \frac{P \cdot 60 \cdot n_r \cdot 100}{L \cdot A \cdot N_e}$
 - BMEP @6000 RPM = 20.33 (bar)
 - P – Power (134 Kw)
 - n_r - Crankshaft rotations for full cycle – (4 Stroke = 2)
 - N_e - rotational speed (rev/s)
 - L – Length of stroke (m)
 - A – Area of cylinder (m²) 408.8058
- Brake Mean Effective Pressure - $BMEP = \frac{P \cdot 60 \cdot n_r \cdot 100}{L \cdot A \cdot N_e}$
 - BMEP @6000 RPM = 36.49 (bar)
 - P – Power (185 Kw)
 - n_r - Crankshaft rotations for full cycle – (4 Stroke = 2)
 - N_e - rotational speed (rev/s)
 - L – Length of stroke (m)
 - A – Area of cylinder (m²)

(Equations developed via (APEX INNOVATIONS PVT. LTD., 2012), (X-Engineer, n.d.))

Finally, it was also about understanding what the advancements within the tabulated values for; A/F ratio, ignition timing, cylinder trim values and any closed loop system changes would do to the engine.

With engine tuning it's difficult to do without a running engine or previous practice in the field and because of this research was conducted into safe but effective approaches to advancing the engines ignition, fuelling and boost.

Given that on a road going vehicle it will spend a majority of its life time within a small margin above or under the stoichiometric AFR, coolant temperatures between 80 – 100 °C and although the intake air temperatures can vastly differ, base it around 30 – 40 °C. With this being the case, to pin point the starting point for understanding the ignition advances that are needed this is the perfect place to begin with respect to discovering the most mechanically

efficient point. To achieve this, we'd need to measure the output torque of the engine across various spark angles when at a given engine speed and load.

The key with this testing is maintaining constants of such as lambda, temperature and airflow of the engine whilst only adjusting the sparking angle. When conducting the test, starting with low spark angles is the starting point where the first set of data is recorded and reviewed, then test again having moved the spark angle forward slightly. Repeating the test through a wide range of angles, to return a series of engine data can then be utilised to depict engine torque against spark ignition in graphical form. Doing this should return a "hook" shaped curve highlighting the point at which the engine is the most mechanically efficient. This test can be performed repeatedly across engine speeds and loading to development a large map of the engine's response to ignition timing changes (Banish, 2009).

When performing this test, the user has to be very wary of 'Knock' that's happening within the cylinder due to pinking. This can be heard with cylinder and sounds like a ball bearing inside a tin can. The EcoBoost engine does happen to utilise individual cylinder knock sensory devices and this could certainly aid the tuner when calibrating.

For base line mapping of the fuelling its best to set the conditions at stoichiometric, so lambda of one for all but wide-open throttle conditions. So, when the engine comes up to working at a lower lambda (richer mix) of around 0.85 for high load percentage and on WOT the map needs to be blended out across in a smooth and uniform pattern. The starting place for the enrichment would be when loads are of about 50 – 60% and the aim is to evenly increment the fuel up to the WOT position, whilst keeping a close eye on exhaust and engine temperatures and if or when they get to high when the engine is held in this area of the table, more enrichment can be added. Because the engine concerned is a forced induction engine that uses a turbocharger, the target lambda should remain somewhat consistent within the boosting area. But because the engine, like its sister the 1600cc EcoBoost may have an "overboost" operating area, this can be taken into consideration within the base map by enriching the mixture to aid cooling and attempt to reduce any knocking effect (Banish, 2007).

With the engine having a turbocharger the lower end and idle condition mapping considerations can be regarded as close if not the same as a naturally aspirated engine because of the nature of how a turbo works, utilising waste exhaust gases, as opposed to being a parasitic drain driven by the crankshaft. And with this in mind, because of the operating nature of a turbo charger, they need heat or velocity to work the turbine side of it and so the more there is the more energy there is that can be harnessed, but because the mass flow is characterised by the engine design and hardware there isn't a lot that can be done in

calibration terms to aid this, which is why appropriate sizing is a key factor to try and eliminate the ‘lag’ that could occur, which would in turn make mapping the transient states between minimum, mid and maximum more difficult.

With the turbo being “low inertia” this dramatically reduces the work need to obtain maximum boost and spooling time. So, with this in mind, peak boost pressure will most definitely occur before peak torque and this needs to account for when mapping the ignition, you should reduce the advance when close to where peak cylinder pressures would occur. Also, an added benefit of turbochargers is that they will continue to work near peak efficiency all the way up to the ‘redline’ area of engine speed, whilst not dramatically increasing the intake air temperatures, but with this being said the system does require a form of charge cooling because of this. And when tuning the ignition, the efficiency of said cooler can dictate how drastically it needs to be advanced or retarded to avoid “knock”. When calibrating the boost tables the exhaust temperatures need to be used as a form of safety factor in terms of prolonging the life expectancy of said turbocharger, catalytic converter and valve train. (Banish, 2007)

Because there is a lot of factors that will dictate the mapping style required it’s difficult to give a definite direction. And when doing said calibration, there is so many variables that are involved when making decisions of running richer or leaner, advancing or retarding timing, tabulated figures for transient states and various operating conditions that effect the loading on the engine when at a given throttle position and engine speed (cruising or WOT).

Therefore, with engine calibration, it all really comes from being hands on and using data that is actually collected from a running engine to make an informed decision.

3.1.3 Experiments

Using the dynamometers guidance notes and user manual, alongside the help of Mark Fall the technician. We investigated the required route that would return what we needed, so the biggest task would’ve been stabilising the engine in a cycle so that basic engine calibration could be conducted and or checked. Because the engine management control unit was to be supplied by SCS Delta, they have a vast collective of knowledge and experience with engine calibration, whilst also having dealt with the 1500cc EcoBoost beforehand offering mapping solutions for customers. We would’ve been confident that the basic configuration that would’ve have been on ECU would have been safe and capable to run efficiently and effectively from the off. But this still meant we needed to create a cycle within the dynamometer programme and build up a lot of effective data that could in turn be analysed

for aspects of definite improvements in the tuning or perhaps other areas that also needed attention.

Starting with the creation of the basic test sequence, we create a basic test file within the software and then checking through standardised Pre-test parameters to be sure (Alarms/messages page) i.e. max engine RPM is set, Fuel supply rate is enough, Engine coolant temperature control is correct and idle speeds. Also, after either selecting certain log channels of just leaving the standard channels in for simply data we'd need to check the safety check list to ensure that:

- Fire safety system is active
- Dynamometer test room door and control room doors are shut
- Combustion air system is on
- Exhaust gas extraction fan is on
- Ignition switch is on
- Fuel supply system is on
- Cooling water system is on
- Idle button is selected on the test screen

With all checks completed, to crank the engine the control key (Ctrl) on the keyboard needs to be held when clicking the crank engine button. When the engine is running, do a complete check of the data logs supplied i.e. engine RPM, coolant temperature, air intake temperature but specifically, the lambda readings to be sure that the engine is operating smoothly. After a short while of warming up period, using the control dials and having selected the “speed” button on the screen, raise up the engine RPM up so to check that the controls are working correctly. Then when confident, take the engine to a high operating position as to then test the system is safe.

Once a good operating test has been completed, turn off everything and shut the test down. Then enter the dynamometer room to check connections like; the engine to the trolley bolts, engine to dynamometer via the cardinal shaft to ensure the bolts are shaking loose. Then for leaks, so; fuelling system across the engine and supply feed, coolant system and connections then also the exhaust system because that could lead to invalid data being collected.

With the testing completed and all parties concerned are confident its operating safely and correctly, the building of test sequences used for tuning the engine can be created.

Experimentation was primarily aimed specifically at advancing the engine control module, but there was discussion of looking into testing of the fuelling system as a whole. So, looking

at the capacity of the fuel injectors used on the engine simply because this was thought to be the biggest failing point in achieving the power required due to them not being physically capable of supplying the demand of fuel to coincide with the required air for correct combustion and this will be discussed in [3.1.6 Testing]. And with the other potential “failing” point was to the fuel delivery pumps, both the low pressure and high-pressure sides being able to safely maintain a suitable pressure to feed the injectors with. There is an external company that offers upgraded fuel pumps for the engine, “Puma Speed Performance Tuning”. This suggested in a preliminary search, that again the fuelling system could present a problem for achieving the desired power and efficiency because these parts were the main area within the limited market of the 1500cc EcoBoost.

But because combustion doesn’t require only fuel to take place, the supply of air was also a key aspect that needed to be addressed. So, given that “Every EcoBoost engine features turbochargers for a power that far exceeds engine size” (Ford Motor Company) this gave the impression that the turbo would be able to sufficiently produce enough boost for the necessary air supply. But again, after a discussion with John, he said he believed that it wouldn’t be enough, and SCS also conducted testing on an engine that included a “Hybrid Turbocharger” and it comfortably achieved the desired power output. However, it wasn’t repeatedly tested, so regarding the reliability of the modification, there is yet to be real world/repeat validating tests. Therefore, testing the turbo with the use of the “Super Flow SF-600” flow bench testing equipment would be a useful tool in the validation between the argument of if it was necessary to uprate the turbocharger.

The experiments that would’ve have been carried on the engine dynamometer, had we received the parts on time, would’ve been primarily from a power tuning perspective. So, having the engine run through a series of step tests, the dyno will go into an automatic running sequence of predetermined parameters set by the technician. Parameters like the starting RPM, finishing RPM, step increase (RPM), ramp time then the setting and average time selects. The test will also require you to input values into the abort/end section so it can return to say idle or completely shut the engine off if something were to happen or just simply when the testing ends. This then allows it to control engine speed and hold the throttle for certain RPM’s, to collect exact data of say; throttle position (%), boost pressure (PSI), intake temperatures (°C), lambda banks ($\neq 1$) and various other channels that would be needed to help understand where to change the engine control module configuration. The step tests are for checking power curves but with a clearer evaluation of what’s actually happening at a specific range as opposed to the “sweep test”. The sweeping test style is simply to return the talked about “print out” that shows maximum power with maximum torque and their associated RPM. This test is done through selecting the essentially the same settings as the

step test, but just with a start and finish RPM value, log timing and again the “end of test” settings.

For the purpose of having the engine be suitable for the car and for our work that would’ve been conducted to be useful to those developing the chassis, we would have liked to have conducted tests that put the engine through a drive cycle that is to the worldwide harmonised light vehicle test procedure (WLTP) standards, simply to see how the vehicle will do in regard to emissions for the mapping we would’ve have configured. Because real world testing was somewhat out of the question, developing this test would still be reputable within the programming of the dyno, because we know what a rough weight of the vehicle would’ve been [820Kg (without driver)] and because the idea was to use the 1600cc EcoBoost’s gearbox, we know that the standard gear ratios and final drive would be: 1st 3.727:1, 2nd 2.098:1, 3rd 1.357:1, 4th 1.030:1, 5th 0.821:1, 6th 0.690:1, Reverse 3.818:1 and final drive at 3.82:1 [based on 205/40 R17 wheel] (Ford Motor Company, 2013). John wants to also offer a “Quaife” insert package, but they offer various ratios based on preference, so these could’ve have been accounted for if it was decided upon.

With this we could’ve setup an automatic testing that started with a stabilisation and warm process, that then moved onto a sequence that had the effect of the changing output ratio into the dynamometer as if the car was imitating a driving pattern like on the road. It would’ve included the required “town driving” procedures with various stops and low gear driving at slower speeds, moving onto rural driving for faster speeds and higher gears, and then finally a motorway driving simulation so the car goes through all expected possibilities that they owner would encounter and we can see how the car would respond to the driving style given an aggressive mapping style. But with the SCS delta control module, it has the capability to store two separate “maps” on and with this feature it means would could’ve programmed a sensible, completely emission friendly configuration for when driving in road conditions. Although it would certainly have a good power and torque curve, the euro six standards would be the main area of concern.

This would be very beneficial for getting the car road legal in the eyes of emission control, but with the other configuration available on the ECU it would allow for a “switchable map” situation. Meaning the other could be for more aggressive driving styles/track use giving the vehicle a larger target audience and just simply more versatile. Although the mapping situation would still have to come within emissions regulations, the power curve could be very different in respect to; peak areas of RPM, over-boost mapping and sound of the car.

3.1.4 Designs

Regarding designs, the only task we could foresee needing to be overcome was the exclusion of the air-conditioning pump that was originally driven off of the auxiliary belt because the vehicle was to be open top and didn't require it, along with it being an unnecessary drain on engine power.

So, given the basic construction for the auxiliary belt looked like:

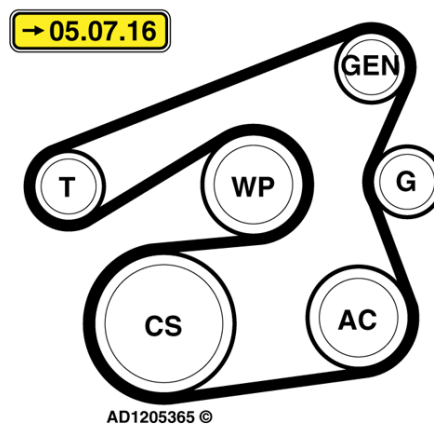


Figure 2 Ford 1500cc EcoBoost Auxiliary Belt (Autodata, n.d.)

It was quite clear that because of the positioning of the air-conditioning pump, when removed it would pose a large problem for trying to easily utilise the standard system. But with that being said, we realised that with the idler pulley (G) in the position that it is, we could quite simply change it out for another of the V-belt style pulley that was currently being used on the tensioner and then measure up for a new length belt that was the same 'V' style (pitch and width) as the original and then it should've been okay.

If it was such as this design didn't work the way we had planned for it to, then the next approach would have been to make 'A-Frame' style bracketry that would work with the original fixing point of the pump, with a new pulley mount to the centre, just like the pump and simply have it free wheel to take up the slack in the belt. But this would be an extra addition to cost and time for reproduction purposes.

3.1.5 Development

With regards to developments, we weren't actually able to make any developments. There were extenuating circumstances that prohibited us from conducting any testing or

experimentation. Therefore, no data was collected, which is why all has been discussed is simply purposed theoretical work.

3.1.6 Testing

Testing procedure of the injectors would've been conducted but there was a limitation of equipment available within vicinity, so outsourcing for a company partnership was investigated and there could've been a list of purposed partners, but it was advised to not contact the companies personally. With this is situation, the testing future looked bleak, but after a meeting with John, he returned with the information that SCS themselves had actually managed to surpass the injection difficulties by discovering that there is an electronic safety feature that stops the injectors and that it isn't a physical problem that would lead to the necessity of larger injectors being sourced, i.e. the 1600cc's injectors. Although it's still not knowledge to us exactly how they surpassed the problem, it clearly shows there would be a definite potential for achieving the required power regarding fuel injection. But, on the other hand, the use of the 1500cc injectors perhaps past their genuinely safe limit may not be a reliable option. Given that it could dramatically reduce the life expectancy of said injectors and for this reason testing should still certainly be conducted to gain a better understanding. Primarily from a safety aspect but also as a small name manufacturer, John has said he wishes to offer a warranty service on his vehicles in order to build a relationship with customers.

For the testing procedure, the main objectives would be:

- Checking the spray pattern of each injector to ensure there is no irregularities along with having the correct distribution and correct atomisation of the fuel.
- Check for leaks from the injectors pintle and this would be done at lower operating pressures to make them visible.
- Run the injectors through flow rate testing so the values can be used to calculate estimations for validation of thought process.

Stuart Harries

3.2 Parts Sourcing & Engine Development

3.2.1 Work Content

The group decided that the first task for the project was to inspect the condition of the Ford 1.5 Ecoboost engine that was going to be used. This was to check for any possible damage to the engine from previous use or from transportation. Also, whilst inspecting the engine, a list of the required parts to get the engine running was produced. This inspection took place on the 9th October 2018 (Figure 71). No visible external damage to the engine was found during the inspection. The group then had meeting with technicians Mark Fall and Barry John in the workshop. This was to check on the history of the engine and to whether any other parts were required that the group may have missed when the inspection took place. During the meeting with Mark Fall and Barry John, the group was informed that the engine arrived direct from the Ford Motor Company's engine plant based in Bridgend, South Wales. So, with this information, we were confident that the risk of any internal faults or damage due to an engine rebuilding processes wouldn't be an issue for us. After clarifying the list of required parts for the engine, a detailed bill of materials list was produced displaying each part, cost and approximate delivery time. This is listed below.

- **Tensioner pulley, V-ribbed belt** – The cost of the component was £34.80 and if purchased directly from Ford the estimated delivery time was between 1-2 working days. (OEM number: BM5Q6A228AA).
- **Alternator** – 2015 Ford Focus 1498cc Petrol FoMoCo 150-amp Alternator. The cost of this component was £49.99, and the estimated delivery time was between 2-3 working days. (Part Number: F1FT-10300-BA).
- **CAT convertor** – Ford Kuga 1.5/1.6 Ecoboost Petrol CAT Convertor. The cost of this was £120, and the estimated delivery time was within 3 working days. (Part number: Cv61 5E211-KE).
- **Starter motor** - 17-18 Ford Kuga 1.5 Ecoboost Petrol Starter Motor. The cost of this component was £44.95, and the estimated delivery time was between 2-3 working days. (Part number: F1FT-11000-MA).
- **Belt** – a belt was required but this could only be purchased after the alternator had been fitted to the engine as an accurate measurement was needed.

- **Engine oil** – 5w20 Ecoboost Magnatac Engine Oil 5 Litres. The cost of this was £24.50. (Product Number: WSS-M2C948-13). This will be purchased direct from Ford.
- **Engine control module** – Ford 1.5 Ecoboost engine management kit. The cost of this was £1400, and the estimated delivery time was between 2-3 weeks. This will be purchased from SCS Delta.

This list was then emailed to the group supervisor (Pedro Monteiro Ferreira) and John Avery.

3.2.2 Research

With the use of Autodata, research was carried out to find the correct torque settings for the bolts across the engine. After having a meeting with Mark Fall, it was recommended to only research the torque settings for the external bolts, as the engine arrived direct from Ford. This meant that the engine had not been stripped down and re-assembled before the development project started. From looking at Autodata, it showed that some of the components needed to follow certain stages for the bolts to be torqued correctly. To torque the flywheel correctly, four steps needed to be followed.

Components	Torque Setting (Nm)
Oil Pump	9
Sump Drain Bolt	28
Clutch Pressure Plate	29
Inlet Manifold Short Bolts	10
Inlet Manifold Long Bolts	18
Water Pump	10
Spark Plugs	13
Fuel Rail	23
Crankshaft Position Sensor	10
Camshaft Position Sensor	9
Engine Coolant Temp Sensor	10
Lambda Sensor	48
Knock Sensor	18
Engine Oil Pressure Switch	18
Oil Filter	15

Table 4- Torque Settings

Research conducted was to find new components that would enhance the performance of the engine. All the components that were found would be purchased from Puma Speed Performance Tuning. A bill of materials list was produced, stating each part, cost and the possible performance upgrades from using that component. This list was emailed to the group supervisor (Pedro Monteiro Ferreira) and John Avery.

- **Hybrid Turbocharger** – 1.5 Ecoboost X-27 280 Billet 7 Blade Hybrid Turbocharger – Ford Focus Mk3 1.5 Ecoboost. The cost of this product will be £789. This offers up to 270 BHP.
- **Hybrid Turbocharger** – 1.5 Ecoboost X-47 320 Billet 7 Blade Hybrid Turbocharger – Ford Focus Mk3 1.5 Ecoboost. This Cost of this part will be £925.99. This offers up to 300 BHP.

Callum-James Hill

3.3 Dyno Fitment – Trolley Fitment

3.3.1 Work Content

The intended outcome for this side of the project is to connect the Ford 1.5 Ecoboost engine to the Dynamometer. To achieve this goal, the engine would first need to be mounted to an old dynamometer trolley that was recovered from the Cadogan building on the old Mount Pleasant campus. The trolley was fitted with an older Ford 1.6 Duratec engine, which needed to be removed and the mounting frame, on which it was situated, would potentially need modifying for our requirements.

3.3.2 Research

Research was carried out on an array of different engine trolleys, to take inspiration and to investigate what modifications would be best, regarding fitment of the 1.5 Ecoboost engine.

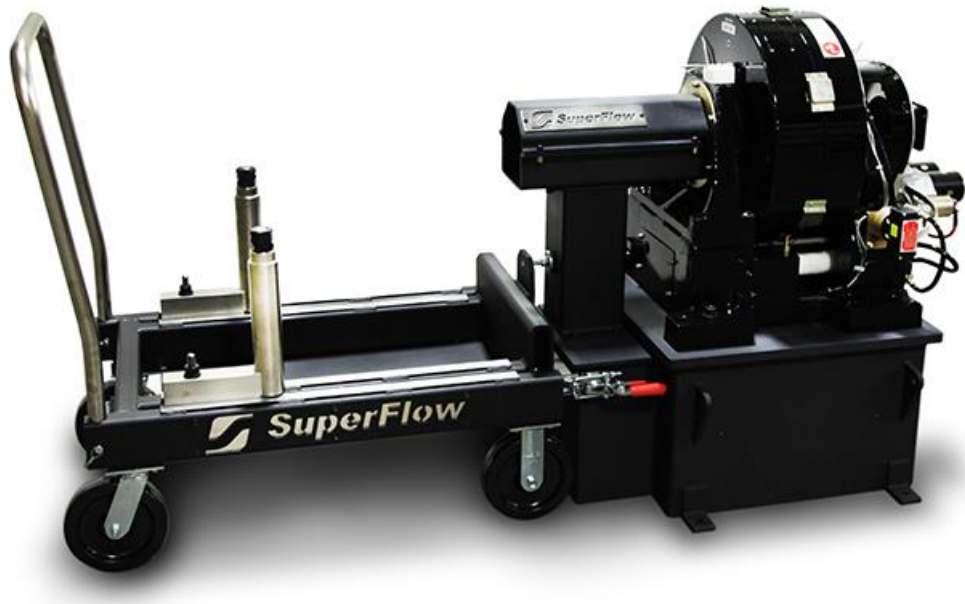


Figure 3-1 - Eddy Current Dynamometer & Trolley (SuperFlow, 2019)

Shown above, is an engine trolley manufactured by SuperFlow which is used on their SF-EC Eddy Current Dynamometer range. They use adjustable engine mounts which allows quick and easy adjustment of the height of the engine on the trolley during docking, to ensure the crankshaft lines up with the dynamometer. As an addition, the SuperFlow dynamometer has clamps on either side which then fasten to the trolley making it more secure and rigid during operation, this was taken into consideration.

3.3.3 Designs

In order to control the engine while it is mounted on the dynamometer from within the control room, a control board was required which needed to be attached to the trolley. In addition to this, to fully understand the concept of how the control board and all its components worked schematic diagrams were created consisting of the unit and everything within it, this process is covered with more depth further on within this report.

Shown in Figure 3-2 below is the first schematic drawing.

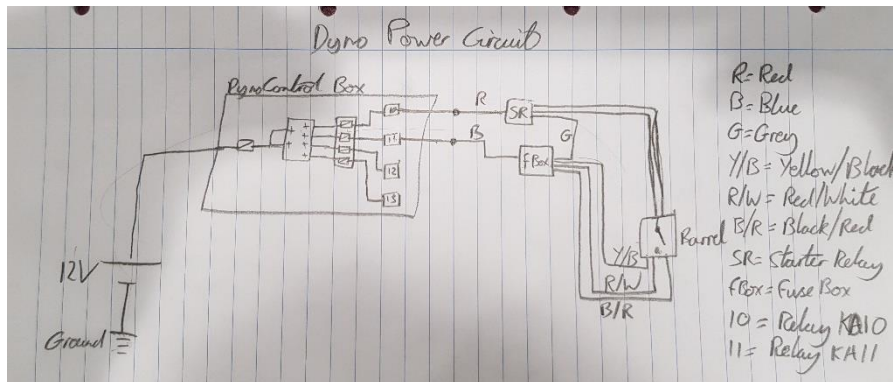


Figure 3-2 - Sketch of Electrical Power Circuit

Furthermore, with the use of SolidWorks electrical, a final schematic design was created as shown below.

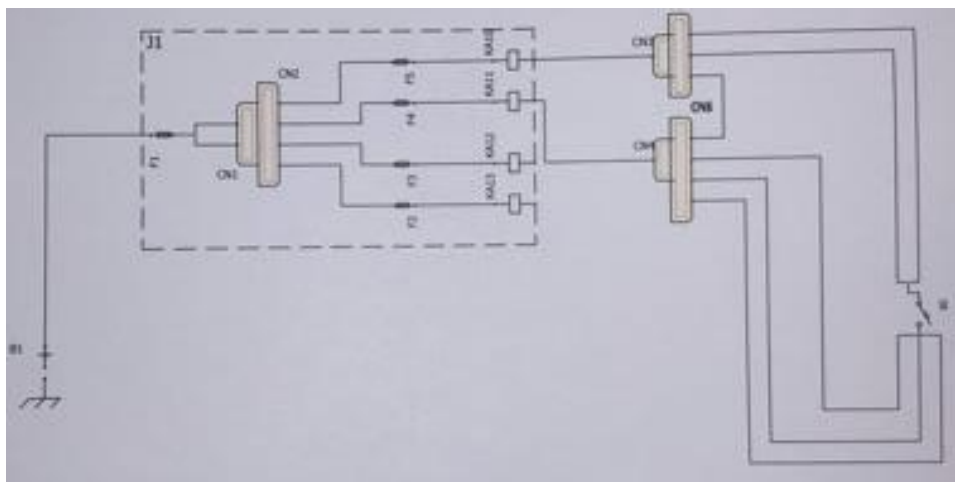


Figure 3-3 - SolidWorks Electrical Schematic

To complete the trolley, a control board would need to be attached consisting of an ECU, fuse box and an ignition switch.

3.3.4 Building

In order to successfully mount the engine to the dynamometer, it first needed to be fitted to a dyno trolley. The trolley that was chosen, was currently in use, a Ford 1.6 Duratec engine was already mounted on it so it needed to be removed. This trolley was chosen because the Duratec engine block was similar to the 1.5 engine block that was required to be mounted, this meant that the mounting brackets attached to the trolley would need little to no modification for the required task. To check if modifications were needed, the bolts that currently held the Duratec engine in place were visually checked for placement in correspondence with the Ecoboost engine. Luckily the placement for each of these were in the same location, but

the Ecoboost engine would provide a problem with one of the mounting arms, as the air conditioning pump would use some of the same holes that the mounting arm would use. The client had already decided to remove the pump, so this was a welcome solution, the holes were measured as accurately as possible to ensure they matched up, which they did.

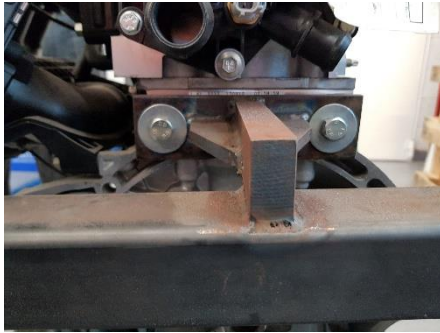


Figure 3-4 - Flywheel Side Mounting Bolts

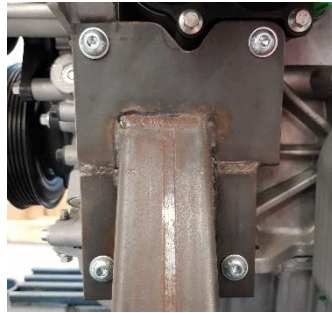


Figure 5 - Intake Side Mounting Bolts

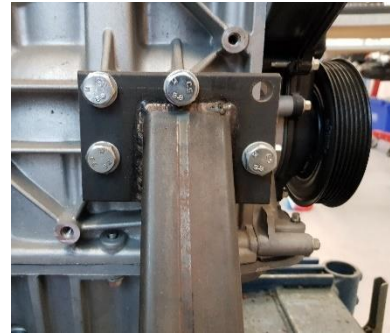


Figure 3-6 - Exhaust Side Mounting Bolts

Using a crane, the 1.6 Duratec engine was removed from the trolley and placed on a pallet at the side of the workshop. The 1.5 Ecoboost engine was then lifted up to an appropriate height, the mounting brackets were removed from the trolley and bolted onto the engine in mid-air, this ensured that there was no external pressure or twisting forces applied to the engine block allowing for adequate fitment.

The engine was then lowered down and bolted down to the trolley so that baseline measurements could be taken from the centre of the crankshaft, to the base of the trolley and then from the base of the trolley to the floor, the required height from the crankshaft to floor needed to be 800mm to allow it to line up with the rotating shaft of the dynamometer. Unfortunately, the measurements added up to 675mm, therefore meaning the engine needed to be raised 125mm on the trolley.



Figure 3-7 Trolley



Figure 3-8 Engine to Trolley Fitment

Initially, to achieve this the plan was to use 50mm steel box sections which would have been modified and welded together until they were the correct height. Fortunately, Workshop technician Mark Fall sourced four adjustable mounts to use instead, which are more conducive and easier to modify if ongoing adjustments needed to be made allowing the height to be conveniently tailored to whatever is required of them.



Figure 3-9 Adjustable mounts

The adjustable mounts required modification to allow them to be bolted to the trolley, and to allow the original trolley mounts to be bolted to the top of them. Firstly, with the use of a Vernier calliper measurements were taken to identify where holes needed to be drilled, upon inspection it became evident that the original mounts were not all the same size ranging in width from 126mm to 120mm. This proved not to be as much of a problem as first anticipated as the original mounts were placed on top of the adjustable mounts and lined up into place. Then with the use of a sharpie markings were made through the existing holes on the original mounts onto the adjustable mounts, this was repeated for each original and adjustable mount, they were then marked up between 1 & 4, to ensure each original mount and adjustable mount that were marked up together would not become mismatched. Therefore, it did not matter that the original mounts were different sizes as each adjustable mount was specifically marked for each original mount. In addition to this, markings were then made on the bottoms of the adjustable mounts allowing them to be bolted to the trolley. When all measurements had been completed, the drilling could begin with the assistance of Workshop Technician Paul Davies, foremost, all the markings were centre punched then drilled with a 3mm centre drill as pilot holes, 6mm, then 9mm holes and finally 12mm.



Figure 3-10 Bottom of Adjustable Mount



Figure 3-11 Top of Adjustable mount pilot holes and 12mm holes

Upon further inspection, it became apparent that when the adjustable and original mounts were bolted together, as they were rotated to reduce their height the thread from the adjustable mounts would begin to drive into the bottom of the original mounts, therefore limiting how much they could reduce in height. The solution was to use a “gas axe” in the form of an oxy-acetylene torch, with the assistance of Paul Davies, to cut out a section in the centre of the bottom of the original mounts allowing enough clearance for the thread of the adjustable mounts to feed through whenever they’re lowered to the appropriate height. Using 8 M12x1.25 x 50mm bolts, the adjustable and original mounts were bolted together.

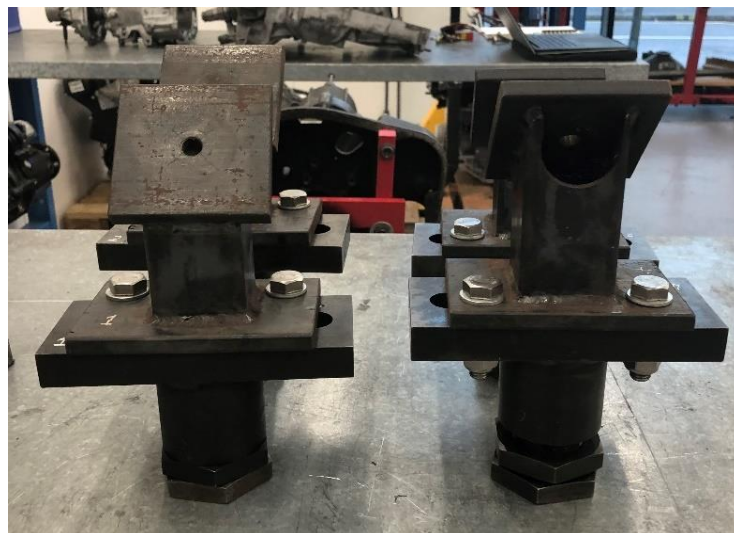


Figure 3-12 Adjustable and original mounts

Before fitting the mounts back to the trolley, the spacers which were situated between the top of the trolley mount and the bottom of the legs that were bolted to the engine, had a pile of spacers between, these were replaced with four rubber bushings as shown below.

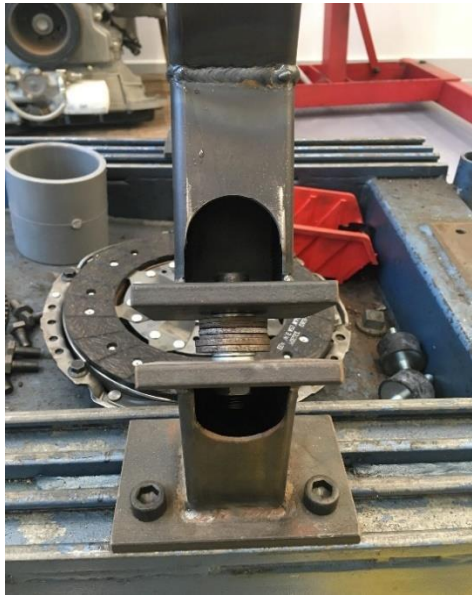


Figure 3-13 Spacers



Figure 3-14 Rubber Bushing

Upon further scrutiny, it became clear that the rubber bushings shown above, may not be able to withstand the stress the engine may put on them due to their small size, this would cause them to deform due to the torque output of the engine. If the engine was perfectly balanced for force and moments it would have no tendency to move or transmit vibration to the frame, unfortunately there is no such thing as a perfectly balanced engine. “As a consequence, flexible mounts are needed for supporting an automobile engine and its peripheral components” (Ramos, n.d.). Research was conducted to see what the most viable option would be to replace them. It was decided to purchase four rubber engine mounts from a UK land rover specialist. A land rover discovery V8 with a maximum torque output of 430NM (JLR, 2019) uses two of the rubber mounts, whereas four were fitted to the trolley and the Ford Ecoboost engine which has a maximum torque output of around 240NM. (Motor Review, 2017).



Figure 3-15 Land Rover Rubber Engine Mount



Figure 3-16 Land Rover Rubber Engine Mount 2

The mounts were bolted to the legs that were fitted to the engine, then with the use of the gantry the engine was raised over the trolley and then gradually lowered down to be bolted on, unfortunately there had been a mistake with the placement of the drilled holes, due to the new thicker, wider Land Rover rubber mounts which resulted in the engine not sitting correctly. To rectify this, the engine was lowered onto the trolley to see where it would sit correctly, a bolt was screwed into the trolley and aligned with where the new markings needed to be made, with a use of a scribe a line was marked across the mount, as shown below.

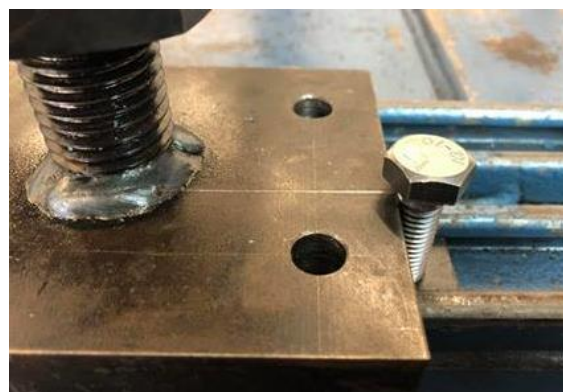


Figure 3-17 Remarking New Holes to be Drilled

The new holes were drilled, firstly with 3mm pilot holes, 6mm, 9mm and then 12mm holes.



Figure 3-18 Bottom of Adjustable Mount Corrected Holes

Once this process was completed the engine was re-fitted to the trolley using 8 M12x1.25 x 40mm bolts and adjusted accordingly until the crankshaft was 800mm from the floor.

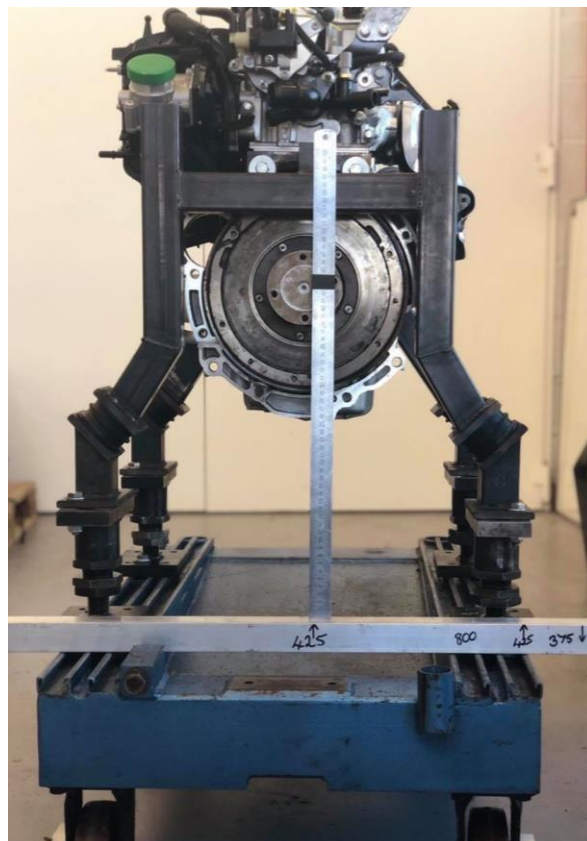


Figure 3-19 Engine on the Trolley at Correct Height

Immediately upon completion of the engine to trolley fitment, a control board needed to be created and fitted to the trolley which would consist of the ECU, a fuse box and an ignition switch. A positive wire runs from the battery to the dynamometer control box into an ignition fuse, which will then run into a junction box, which is then distributed into a small 20amp fuse

box which then feeds into four different relays KA10, 11, 12 and 13. From relay 10 a wire would run over to the ignition relay on the control board and then from the ignition relay to the ignition switch. Furthermore, from KA11 a wire would feed to the crank connection which in turn would then run to into the fuse box on the control board, 3 wires then feed from this, two get relayed back to the ignition switch subsequently providing power to the fuse box which is transmitted to the ignition switch, the third wire then feeds into the starter relay creating a loop within the circuit. Unfortunately, only the control board its self was able to be acquired, all the other parts required to complete the circuit were unattainable due to funding, so a schematic drawing of the circuit was created with the use of SolidWorks Electrical to plan out the circuit, should any funding appear.

An old control board which was sourced from a different trolley was provided.



Figure 3-20 Control Board from Source



Figure 3-21 Control Board Fitted to Trolley

As the control board was significantly looking worn down, it was cleaned with the use of a disc cutter with a wire brush attachment and painted with 'Ronseal No Rust Metal Paint' to make it more aesthetically pleasing. (Ronseal, n.d.)



Figure 3-22 Control Board Ready to be Painted



Figure 3-23 Control Board

Michael Anderson

3.4 Adapter Shaft Manufacture

3.4.1 Work Content

The need arose to physically attach the dynamometer to the engine, with the use of a ‘cardan shaft’ in order to gain necessary data, such as; Power and Torque curves, fuel flow rate, turbo pressure and air flow rates. An adapter shaft needed to be manufactured so that the rotation energy would be transmitted from the Ecoboost engine through to the dynamometer. This would need to be able to withstand the torque transferred through it, without any distortion.

3.4.2 Research

A Masters internship report filed by the University of California Davis was writing to demonstrate how a Honda Prius plug in hybrid engine was mounted onto a dynamometer in order to potentially increase performance and fuel economy with the use of Ethanol (E85). (Dortland, 2007)

Ideas from A.E Dortland’s report about how to manufacture and design the adapter shaft, were taken into consideration for the requirement of transmitting power from the 1.5 Ecoboost and into the dynamometer for analysis of performance. They created a two-piece adapter made from Steel 4340, attaching one end to the engine’s crankshaft, and the other to a cardan shaft connected to the dynamometer, both pieces would slide together using a spline shaft and key connection. Hence, as seen in *Figure 3-40* the shaft was developed to resemble this.

3.4.3 Experiments

By creating basic paper drawings and taking measurements with a Vernier gauge, initial CAD designs were constructed using SolidWorks to suit each individual side of the adapter shaft. These CAD files were converted from ‘.SLDPRT’ into ‘.stl’ files for the 3D model Slicer program ‘Cura’, this program generated a print setup file with various settings, in the form of a ‘.gcode’ that a 3D printer can interpret and build. Using an ‘Ender 3’ 3D printer, these models were formed into a physical 3D prototype that would be used to test fitment and design.



Figure 3-24 - Cardan Shaft End 3D Print

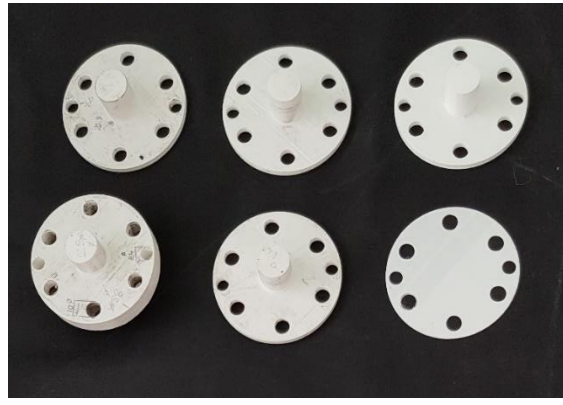


Figure 3-25 - Crankshaft/Flywheel Side 3D Print



Figure 3-26 - 3D Printed Adapter Shaft

Multiple prototypes were created for each individual section of every component. Although many in number, this saved having to print the entire adapter shaft each time which would take at least 18 hours, where the average printing time for a smaller component totalled 2-3 hours. This was beneficial because parts were easier to test for functionality, adjust measurements and saved PLA printing filament until each end was completely correct.

3.4.4 Designs

A basic sketch outlining the design of the adapter shaft was drawn with approximate dimensions as advised by Dynamometer Technician Mark Fall. This was then replicated using SolidWorks to create a 3D model which would be given to Gareth Staddon and Dave Cooper in order to manufacture the require part. There were two ends of the adapter shaft that would need to be created:

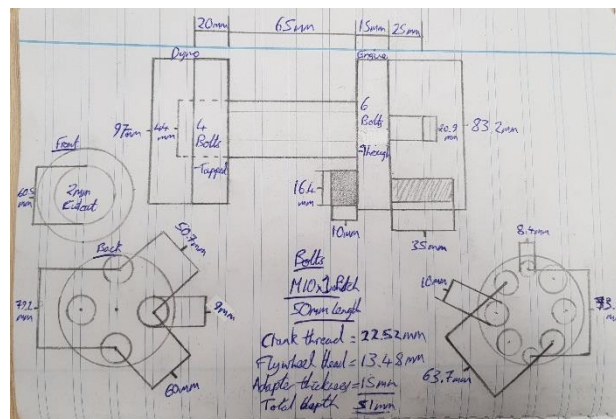


Figure 3-27 - Initial Sketch of the Adapter Shaft

The end that would attach to the Cardan shaft consisted of 4x M10x1mm threaded holes in a rectangular format 61mm x 51mm. The diameter of the base was 97mm to sit flush with the diameter of the Cardan shaft and a thickness of 20mm for torsional resistance. The Cardan shaft was home to an extrusion of 1.8mm, a recess needed to be manufactured so that the adapter would sit flush against the cardan shaft, this recess was made 2mm deep to account for this.



Figure 3-28- Side Profile of Cardan Shaft



Figure 3-29 - Top Profile of Cardan Shaft

The other end would bolt into the crankshaft of the 1.5 Ecoboost engine through the flywheel using 6x M10x1mm bolts, the flywheel was home to two locating rivets which were slightly oval but 8.3mm at their widest part, so the holes were made to a diameter of 8.5mm for enough clearance, using 3D prints to test these dimensions, it was found that 8.4mm was too tight a fit due to the oval shape of the rivets.



Figure 4-30 - Engine Flywheel Profile with Locating Rivets



Figure 4-31 - Engine Crankshaft Profile

The crankshaft side of the adapter shaft had a diameter of 83mm to sit within the 85mm diameter recess of the flywheel, and a thickness of 20mm, to ensure enough strength to withstand any torsional force applied through it, there was an extrusion 25mm long and a diameter of 20.8mm which would sit within a recess inside the crankshaft ensure the adapter was concentric to the crankshaft and to provide a guide for positioning.

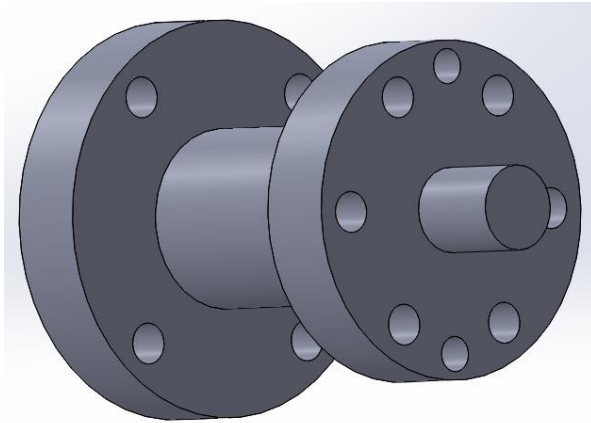


Figure 3-32 - CAD of Crankshaft Side

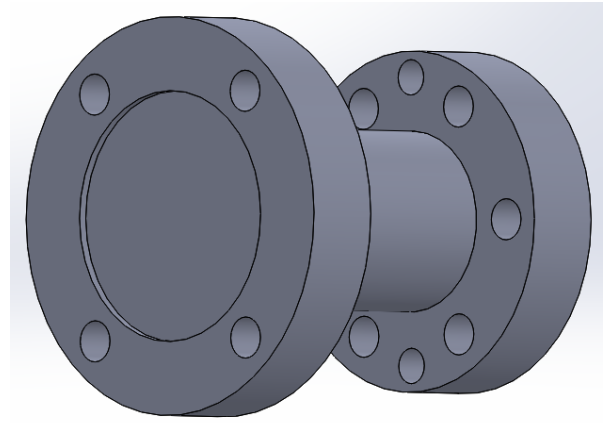


Figure 3-33 - CAD of Cardan Shaft Side

Both ends were joined together by a solid shaft 65mm long and a diameter of 44mm, these were the smallest dimensions possible due to the following restricting factors. The length of the shaft was determined by the length of the bolts that would be inserted through the holes, with enough clearance to enable an Allen key to tighten the required bolts. The diameter of the shaft had to be as wide as possible, but it was limited by the head of the bolts that would be used, the bolt heads were measured to be 15mm, this meant that for enough clearance around the bolt enabling them to rotate smoothly the width was to be 17mm clear of the bolt holes, with the overall diameter of the central shaft at 44mm.

3.4.5 Testing

Once the design and measurements of the adapter shaft were finalised, stress tests were conducted within SolidWorks to be sure that the adapter shaft would withstand the torque being applied through it. Considering the dynamometer has a limit of 1000Nm of torque, and the engine being used for the group project would only be estimated to be 300Nm at most, if the adapter shaft could withstand 1000Nm of torque being applied, it could tolerate the same torque as the dynamometer without deforming. Multiple static studies were conceived, locking one end in place and applying the load force of 1000Nm Torque being applied to the opposing end and vice versa. The intention of this was to simulate what would happen if, with the engine at full throttle, the adapter shaft would withstand a sudden and complete stop, such as; if the dynamometer or engine completely seized due to poor lubrication or mechanical failure.

This resulted in a slight rotational deformation on the rotating end, stemming from the centre of the shaft. The deformation observed through the tests, depended on the material used and the torque applied, but should be efficient enough for what we require as although it will be under the same applied force, it would be free to turn at both ends and therefore should not see the situation that that the static study replicated.

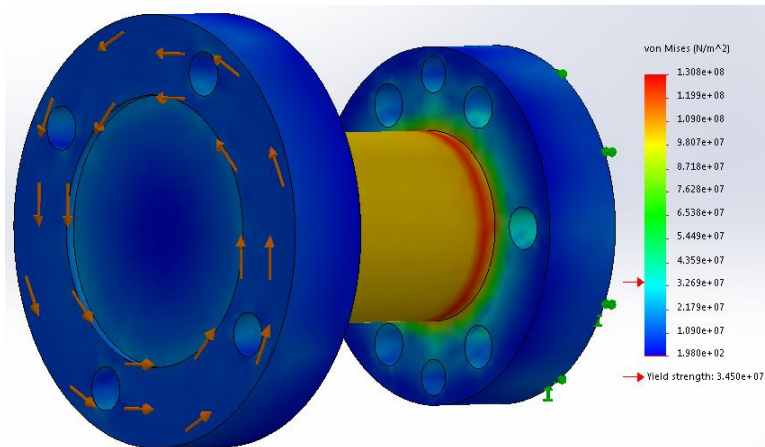


Figure 3-34 - Static Study - Cardan Shaft Side

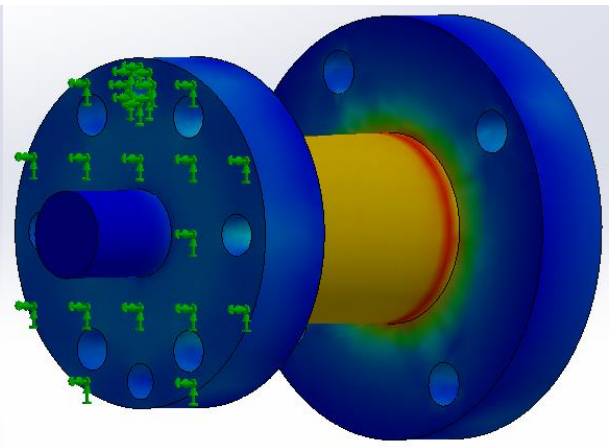


Figure 3-35 - Static Study - Crankshaft Side

3.4.6 Building

The crank shaft of the engine consists of 6x equally spaced M10x1mm threads, which the flywheel bolts onto, along with 2x 8.6mm slightly oval locating rivets to ensure correct placement. The cardan shaft has 4x 10mm holes which form a rectangle, with a width of 51mm and a length of 60mm between the centres of the holes.

The adapter shaft needed to incorporate the two different fixtures. Measurements were taken multiple times of each adjoining end, to ensure accuracy, and from these, a CAD drawing of the adapter shaft was created, with guidance from Dyno technician Mark Fall about how the shaft should look and limitations on dimensions.

With the use of a 3D printer, prototypes were designed to check the dimensional accuracy of both the CAD designs and the initial measurements taken in a physical representation. Adjustments were made to the CAD design several times to provide a meticulous degree of certainty that it would fit and sit correctly once the steel counterpart was produced.

Dyno technician Mark Fall aided us with the sourcing of a 100mm diameter solid billet in the university storage which he assured the group, would be sufficient to tolerate the forces applied through it. Unfortunately, the exact material that was used to manufacture the shaft, could not be identified for the stress test.

The final CAD drawings were given to Gareth Staddon, whose expertise on a lathe, created a beautiful steel version of the adapter shaft, that would tolerate the torsional forces applied from the engine, and sufficiently transfer the torque through to the dynamometer for analysis.



Figure 3-36 - HAAS VF-0 Vertical Machining Centre



Figure 3-37 - The Worktable of the HAAS VF-0 VMC

After the turning process on the lathe had been completed, David Cooper received the CAD design for the adapter shaft and, by using a HAAS VF-0 Vertical Machining Centre, CNC the multiple holes required, along with the recess that sits against the extruded lip of the cardan shaft, ensuring a snug and secured fit.



Figure 3-38 - Crankshaft End of Adapter Shaft



Figure 3-39 - Cardan Shaft End of Adapter Shaft



Figure 3-40 - Complete Adapter Shaft

The 4x 9mm holes that would bolt to the cardan shaft, connected to the Dynamometer, were required to be tapped with an M10x1mm pitch thread, and bolted together using the respective thread 40mm length bolts. The 6x 10mm holes on the engine side were 'through holes', meaning the M10x1mm bolts would sit through the adapter shaft, the flywheel and clutch assembly, and bolt into the thread of the crank shaft using 50mm length bolts.

Once preparation of the adapter shaft was completed, fitment was tested by checking that all holes, bolts and locating rivets sit snugly, and securely bolted together with no play within the components. The only concern that was initially raised would be if the 'cap head' screws connecting the Cardan shaft to the adapter shaft, would limit the mobility and freedom of the cardan shafts' universal joint. Luckily this only proved an issue when bolting together, about half way down the thread the head protruded into the bolts' heads' path by about 3mm unless

the 'UJ' was rotated around, past a certain point the bolt was and once all four were tight, the bolt heads had a 3mm clearance on all sides from the universal joint, ensuring full degree of freedom.

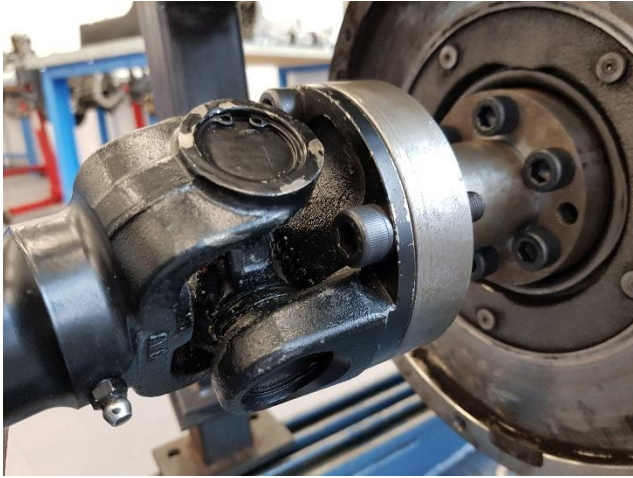


Figure 3-41 - Cardan and Adapter Shaft Bolted Together

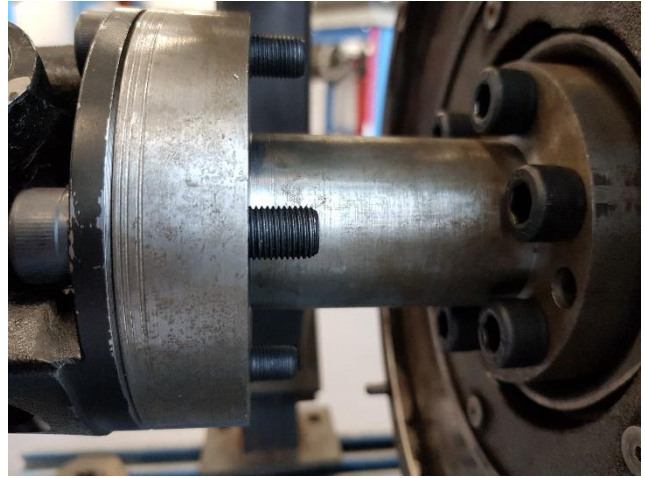


Figure 3-42 - Crankshaft/Flywheel and Adapter Shaft Bolted Together

Callum James-Hill & Michael Anderson

3.5 Engine Fitment to Dynamometer

3.5.1 Work Content

This task would require the Ford 1.5 Ecoboost engine, that was now successfully fitted onto the dyno trolley, to be mounted DSG Schenck WS230 Eddy current Dynamometer. This would enable the group to undertake the operation of running the dynamometer, gaining access to information detailing specific characteristics about the performance of the engine as standard. Throughout the development process this operation would be repeated, allowing the group to assess how performance of the engine has progressed and potentially increased.

3.5.2 Research

Research was conducted into how to successfully mount an engine onto a dynamometer. Although there was nothing specifically on the Ford 1.5 Ecoboost engine for dyno fitment, the process of engine mounting, connections and power transmission would be the same.

Engine Labs published an article detailing what deliberations to acknowledge before taking your engine to a dynamometer centre for measurement of characteristics and analysis. Although some of the details are basic, they are overlooked by a considerable amount of people, as simple as, ensuring all fastenings are correctly torqued, a spare battery, spark plugs fuel and lubrication are brought along by the customer to ensure that there is no delay if something goes wrong. (Engine Labs, 2012)

Research was conducted on the DSG Schenck WS230 Eddy current Dynamometer (apicom DS Group, 2019) in order to acquire technical data about how the dyno operates, the maximum torque it can withstand and at what RPM this occurs at. Eddy current dynamometers were briefly researched as to ascertain their purpose and how they operate based on the principle of magnetic fields to provide counter restraining torque. (Dynamometer World, 2019)

SBD Motorsport have developed a blog on their progress of testing a Ford 1.6 Ecoboost engine and the development of it upon a dynamometer. It details specific parameters and parts they upgraded through their development and provide graphs to show how the performance of the engine was affected through each change, and how to overcome any problems you may encounter through refinements. (SBD Motorsport , 2017)

3.5.3 Building

Once the 1.5L Ford Ecoboost engine had been successfully mounted to the trolley and aligned to the correct height, 800mm from centre of the crank shaft to the floor, it could now be securely mounted to the DSG Schenck WS230 Dynamometer.

On the four wheels of the trolley, Acrylic blocks were attached, which would wedge against the corresponding Acrylic rails bolted to the floor mounts of the Dyno, this will keep the trolley in a consistent position and stop it from being misaligned if the engine is removed and replaced.

The trolley was ratchet strapped to the frame of the Dynamometer, ensuring adequate tension, the secured position made it possible to accurately measure the distance between the Coupling Adapter Plate, which bolted onto the driven rotating shaft of the Dyno, and the adapter shaft, which was bolted to the crankshaft of the engine. This measurement corresponded to the Cardan shaft, which, when fully compressed, measured 500mm in length. In order to compensate for any possible vibration, the Cardan shaft was extended 20mm for a 'cushion of protection' due to the damper within the Cardan shaft, totalling 520mm distance needed between the adapter shaft and the Dynamometer.

The original measurement was 490mm, which meant the engine needed lightly re-positioning on the trolley 30mm away from the Dyno. To accomplish this, the M12x1.25mm bolts securing the adjustable height feet to the trolley were slackened and a copper and hide hammer was used to gently persuade each foot of the engine mounting brackets in turn, re-checking the measurement every rotation until the required 30mm adjustment had been successfully accomplished. A spirit level was then used to ensure each adjustable height foot was not misaligned, then, each bolt securing the foot, previously slackened, was tightened to the correct torque setting 123Nm. (Fuller Fasteners, 2019)

The adapter shaft was then removed from the crank shaft, this proved the easiest way of bolting the Cardan shaft and adapter shaft together because, as before, the universal joint protruded into the bolts path. With the adapter shaft separated from the crank shaft, it was easier to position the Cardan shaft's universal joint so that it was furthest out of the way of each bolt, preventing any further damage to either the bolts or the thread inside the adapter shaft. Each bolt was secured using thread locking paste on advice from Technician Mark Fall, who had experience with previous bolts vibrating themselves free, an event best not recreated. Once dry, the bolts which protruded a total of 10mm out the far side of the adapter shaft which limited the ability to insert the bolts for the crank shaft, these were cut back using a hacksaw, which solved the

problem. The adapter shaft was painted, excluding the connecting faces, to give it a rust protective coating and left to dry.



Figure 3-43 - Cardan Shaft Bolted to the Painted Adapter Shaft

The adapter shaft could now be re-bolted onto the crank shaft using the M10x1mm x 50mm bolts, securing these in place with thread locking paste to again prevent vibrations from loosening during use, with a torque of 70Nm. The other end of the cardan shaft was bolted onto the Dyno's coupling adapter shaft with M8x1.25mm x 20mm bolts, these had a cushion of protection due to the slightly extended cardan shaft, they were also to be removed at a later point, so the thread locking paste should not be necessary until everything was ready to go.

Now the engine was connected, it was visible to see that the engine was not entirely central in the Dyno's cardan shaft's enclosure, this needed to be rectified. To do this, slight adjustments were made to the adjustable feet on the inlet side of the engine mounts to drop them 3mm in height, this was enough to centralise the engine on all sides. The enclosures' lid was placed back on in order to ensure that there was enough clearance all around the cardan and adapter shaft, to not limit or impede operation or cause damage to any components.

Once, the engine was connected to the Dyno via the adapter shaft to the correct distance apart, the trolley now needed to be fixed down in place so that it does not vibrate or move around while the engine is in operation. For this, we devised a plan of using six separate pull action quick release latch toggle clamps, to pull down independently on the front, left and right of the trolley, with two clamps on each side. This should be more than sufficient in fixing the engine trolley onto the frame of the Dyno, by withstanding any unwanted forces, distortion or movement that could potentially occur during operation of both the Dynamometer and the Ford 1.5L Ecoboost engine.

A random cylinder was welded onto the trolley, to the Flywheel side of the trolley. This may have previously been used for mounting purposes originally but would now impede on the toggle clamps' ability to connect to its' opposing latch, therefore, this cylinder would need to be removed. A handheld angle grinder, fitted with a cutting disc, aided the removal task, but due to its' position, required awkward angles to sufficiently clear away the excess metal. The remaining surface was then grinded using a flap disc, smoothing it back to the surrounding level of the trolley.

Initially, clamps were positioned on the Dynamometer end of the engine trolley, in suitable positions with enough clearance from the cardan shaft enclosures' supporting arm. Each clamp was placed into position on the top face of the trolleys' frame, marking the holes using a centre punch, then drilled through using two different drill bits, the 1st was a 7.6mm to start the initial hole 1mm deep, this helped to centre up the 2nd 6.7mm drill bit finishing the hole. Once all eight holes, four per clamp, were drilled through, an M8x1.25mm tap created a thread, in which, corresponding 8x M8x1.25mm 25mm length bolts would secure the two clamps into position.



Figure 3-44 - Random Cylinder Welded to Trolley That Was Removed



Figure 3-45 - Welded and Painted Buffers



Figure 3-46 - Clamps Attached to Buffers



Figure 3-47 - Fully Fitted Clamps and Buffers

The clamps were too short to reach the 500mm distance between the engine trolley and the mounting frame of the Dynamometer. To overcome this, two mounting arms ("buffers") needed to be created by cutting 2x 275mm lengths of 75mm x 50mm rectangular box section steel and 4x 80mm x 50mm rectangles out of 10mm thick steel plate which were to be welded to each end of the box section. Two of these steel plates were drilled through the centre point on the flat face using a 20mm drill bit, with a M20x1.75mm threaded nut welded onto the inside of the plate. Two hook latches would be welded in place at the end furthest from the Dyno, that the toggle clamps would fasten to. While these were being welded by workshop technician Paul Davies, the corresponding holes could be marked up and drilled into the frame of the Dynamometer for the brackets to be bolted to, using 2x M20x1.75mm 40mm length bolts.

The 20mm holes were marked out by using a 50cm long ruler, aligned to the centre line of the clamps' arm, and flat against the frame of the Dyno, a set square helped accurately draw a vertical line down with a permanent marker pen. On the frame of the Dyno, close to where the holes would be situated, a cut-out on either side, used as access already existing mounting bolts, provided us with a reference point of where the bottom of the webbing inside the frame, was located. Drilling through the webbing would weaken the structure, so using the 50cm ruler, aligned level to the bottom of the cut-outs, aided with drawing an accurate horizontal line.

Both vertical and horizontal lines would enable the correct placement for the 20mm holes that required drilling, which would secure the two mounting brackets to the frame of the Dynamometer. The end plates which had the 20mm holes drilled through, were aligned against the marker pen lines, the horizontal line against the bottom of the end plate, while the vertical line determined where the centre of the hole should be located, the hole's circumference was marked using the sharpie pen, while a small ruler measured the direct centre, a centre punch left a distinguished indication of the exact centre of the circle so that could they both be drilled accurately, as the point on the tip of the drill bit, would sit within the centre punches' mark, reducing any chance of the drill bit jumping or slipping on the polished and painted surface of the Dyno frame.

Both holes were drilled using 5 different drill bits, increasing in size, to reduce the strain applied to each drill bit during its' operation. A 3mm drill bit created a pilot hole to start the process, followed by a 6mm, a 10mm, 14mm, and a 20mm drill to gradually widen the hole step by step, until the required size was achieved. The 2x M20x1.75mm screws bolted through the frame of the Dynamometer and into the fabricated bump stops, enabling the engine trolley to sit against them firmly and latch correctly and securely to the horizontally mounted toggle clamps previously fitted onto the engine trolley.

The four remaining clamps were mounted vertically on the left and right sides of the engine trolley. On the front end of the trolley, closest to the Dyno, on the left and right side of the trolley, were four existing tapped holes for M6 bolts, unfortunately, their position didn't correlate to either of the brackets that would be used for this project, so new ones would have to be made far enough away from these holes to safeguard against any structural weakness within the trolleys' frame.

The first bracket was positioned freely, as close to the front end of the trolley as possible, ensuring it was level and central against the top, bottom and edges of each face, this avoided the possibility of accidentally drilling too close to a fold behind the surface, which would block and impeded the full length of any bolt that would hold the clamp in place. The inside of each 7.8mm mounting hole on the clamp was marked using a sharpie pen, the centre of the top left

hole was then made clear using a centre punch, assessing the punch mark for accuracy before removing the clamp. An initial 7.6mm drill bit was used to start the cut 1mm deep, this helped the second 6.7mm drill bit cut cleanly into the frame of the trolley. This 6.7mm hole was then threaded using two M8x1.25mm taps, an initial taper tap and a second tap, for the corresponding M8x1.25 25mm length bolt to secure inside.

The clamp was repositioned in accordance to the pen marked dots and the hole drilled a moment ago, the correct bolt, and spring washer was inserted and slightly tightened in order to briefly hold the clamp in position. The top right hole was now assessed, it determined how level the clamp would be to the frame of the Dyno and each edge of the trolley face, adjusting as required. A centre punch was used to mark the centre on the hole, assessing and rectifying if essential, then repeating the same drilling process as the previous hole. Repeating the tapping process using the two M8x1.25mm taps and inserting the correct bolt, then hand tightened. This now meant that the 3rd and 4th holes at the bottom of the bracket were fully aligned and would not move position, the marker pen dots on the trolley were assessed against where the holes now sat for level. Both holes were marked using a centre punch, drilled using both 6.7mm and 7.6mm drill bits, tapped using the two previous M8x1.25mm taper and second taps, then bolted up securely using the same M8x1.25 25mm length bolts, the same process applied to the previous two holes. The bracket was then judged on stability and security. The remaining 3 toggle clamps followed the same exact process.



Figure 3-48 - Holes Drilled for Clamps



Figure 3-49 - Drilling Holes for Clamps



Figure 3-50 - Finalised Side Mounted Clamp

Four hook latches would be positioned and bolted onto the frame of the Dyno, in a location easily reachable for the pull action quick release toggle clamps now bolted onto the engine trolley. They would be mounted at a 45° angle to both the frame of the Dyno and the toggle clamps, this would resist forces both vertically and horizontally on the 'x' and 'y' axis relative to the respective plane ensuring extra security and protection. In order to mount these, the same 75mm x 50mm box section material that was used for the bump stops, was marked up provide

a prototype for placement and plausibility of concept, this was then cut using a band saw, ensuring the long face that would sit against the frame of the Dyno, did not have the crease/seam from where the manufacturer of the steel has joined the edges together, as this prevented the face from sitting completely flat.



Figure 3-51 - Rear View of Floor Mounts



Figure 3-52 - Front View of Floor Mounts

The cut steel featured a tapered side at 45° , the bottom face had a length of 70mm with a 12mm hole drilled centrally in the face for an M12x1.25mm bolt to screw into the bottom frame of the Dynamometer. Both the tapered box section and the hook latch were loosely positioned, hooking onto the toggle clamp to determine the best position for them to be situated and that the mechanism would work as hoped. The box section steel was then cut into three more 45° tapered, 70mm bottom surface lengths, and drilled using a 12mm drill bit in the centre of the bottom face to cut two 12mm holes, the four hook latches were then welded onto the slanted face of the box section steel, along with a plate of 10mm steel shaped to fit the bottom of the inside of the box section, welded into place, which will ensure that it will not deform when bolted down to the correct torque setting (120Nm).

The engine and trolley were located correctly into position on the dyno frame, which helped align the welded hook latch box sections into place, relative to the clamps that would determine their position. The holes that required drilling, were marked out onto the dyno frame using a pen to mark the outside rim, and centre punched the hole. The four holes were then widened to size using small increments of drill bit size, increasing by approximately 3mm each, starting from 3mm gradually rising to 10.5mm which would be sufficient in size to accompany the thread that will be carved into the hole for the M12x1.25 thread.

Once the holes were tapped, the completed brackets were bolted into place to ensure correct fitment and positioning against the dyno. Everything matched perfectly and provided more than enough security between the dyno and the engine trolley.

This concluded the engine to dyno fitment of this group project, the only thing left to do, as recommended by Mark Fall, was to paint the engine mounting brackets fitted to the trolley with 'Ronseal No Rust Metal Paint' which did enhance the aesthetics of the engine mounts quite significantly, along with the frame, the 4 constructed hook latches and box section were painted using the same no rust paint as the engine mounts and trolley frame bringing the old and dated trolley a new lease of life. (Ronseal, n.d.)



Figure 3-53 - Completed Fitment of 1.5 Ecoboost Engine to Dynamometer

Josh Bass

3.6 Cooling R&D

3.6.1 Work content & Research

The 1.5 eco-boost engine has a water-cooled charge air cooler, which means it is an air to liquid heat exchanger. The phrase ‘charge air cooler’ means it “cools the turbo’s charged air before it is routed into the engine” (Vestas Aircoil, 2017). The charge air cooler is to be used on a hybrid turbo “internal combustion engine to cool down the hot compressed air coming from the turbocharger” (Thermal Systems: wikidot, 2007). During this project, my role was to research into the thermodynamic properties already incorporated and develop a more efficient cooling system for the 1.5 Ecoboost engine. However, there was many gateways as the group did not have the parts needed to run the engine and also the ECU to develop more BHP to the engine. Therefore, what could have been done, in relation to the cooling system during this project was proposed entirely on theory.

Firstly, research was done into heat transfer equations for heat exchangers and the first law of thermodynamics. Heat can be transferred by three methods, these methods include conduction, convection and radiation; conduction and radiation would be considered as negligible.

Convection is a thermodynamic process “where heat is transferred by one part of a fluid to another by movement of the fluid itself” (A Dictionary of Physics (6 ed.), 2009).

In a heat exchanger, the rate of heat transfer $\dot{Q}$, is where the hot fluid to the cold fluid, is considered to obey the Principle of Conservation of Energy. This is where the total energy remained constant within the entire heat exchanger. The equation below, states convection on the charge air cooler, where in convection U is considered as the overall heat transfer coefficient; A is considered as the heat transfer area and ΔT is considered as the change in temperature.

$$\dot{Q} = U \cdot A \cdot \Delta T$$

Heat exchangers

Types of heat exchangers, such as a tube and shell, or plate, are components that are used to displace the heat from one fluid (or gas) to another. Fluids and or gases do not mix, and heat flows from hot to cold.

Heat exchangers can transfer heat between a liquid and a gas or two gases, in our case air to water. This is where the radiator cools the water heat coming from the engine. There are three types of flow arrangement in a heat exchanger such as parallel, counter-flow and crossflow.

Types of Heat Exchangers

There are two main types of heat exchangers, firstly there is one called shell and tube. This type of heat exchanger is where one fluid flows in the tube while the other flows around the tube which is contained around a surrounding shell (Atkins & Escudier, 2013, A), where the shell and tube heat exchanger contains a mix flow between the three; parallel, counter-flow and crossflow.

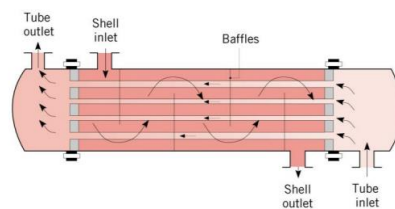


Figure 9: Shell tube heat exchanger (Queenssu, 2019).

However, the charge air cooler is a crossflow plate-fin heat exchanger used on a turbocharged internal combustion engine referred in Figure 12. The heat exchanger is used to “improve the volumetric efficiency by increasing intake air-charge density through isochoric cooling” (Vestas Aircoil, 2017). However, according to (Vestas Aircoil, 2017), if there was a decrease in the air intake temperature this will provide a denser intake charge to the engine which will allow more air and fuel to be combusted per engine cycle, which will increase the output of the engine.

Heat exchanger flow

There are three main types of flow in a heat exchanger, firstly there is parallel flow. According to (Atkins & Escudier, 2013, B) is where two fluids at each end enters separately, then flows together in the same direction.

Parallel Flow

$$Q = UA \cdot LMTD$$

$$LMTD = \frac{\Delta T_{out} - \Delta T_{in}}{\ln \left(\frac{\Delta T_{out}}{\Delta T_{in}} \right)}$$

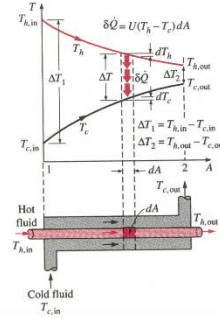


Figure 10: Parallel Flow (Queenssu, 2019)

Counter-flow is where two fluids at each end enters separately but moves in the opposing direction (Atkins & Escudier, 2013, C).

Counter Flow

$$Q = UA \cdot LMTD$$

$$LMTD = \frac{\Delta T_{out} - \Delta T_{in}}{\ln \left(\frac{\Delta T_{out}}{\Delta T_{in}} \right)}$$

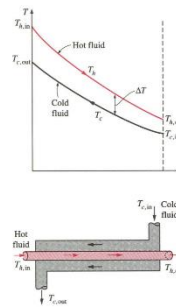


Figure 11: Counter-Flow (Queenssu, 2019)

A crossflow heat exchanger, which is used in a plate-fin heat exchanger, is “made of flat metal sheets separated by corrugated sheets to create longitudinal channels with the sheets being sealed along the edges parallel to the corrugations. Separate hot and cold fluid streams flow through alternate channels” (Atkins & Escudier, 2013, D).

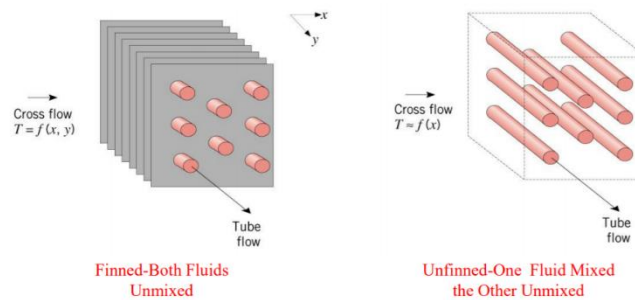


Figure 12: Crossflow Heat Exchanger (Queenssu, 2019)

Background research on the charge air cooler

Figure 13 shows the circuit of a water-cooled charge air cooler, using a fluid as coolant, being distributed through a circuit which contains a radiator and pump. The fluid, after passing through the radiator, is then passed through the water charge air cooler, which results in removing heat from the air. This method of using a water charge air cooler will have several benefits over a charge air cooler (Devi Sankar & Kishore, 2016).

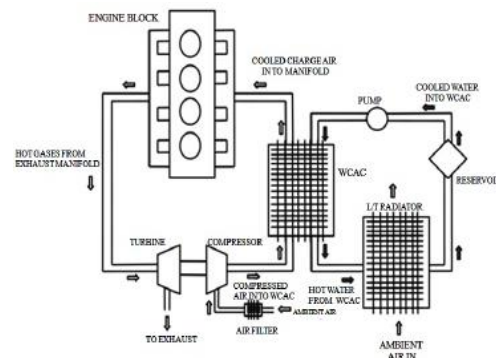


Figure 13: Water-Cooled Charge air cooler in turbocharge diesel engine (Devi Sankar & Kishore, 2016)

According to Vestas Aircoil (2017), the main process for the charge air cooler is to return the engine air to a lower temperature as it passes through the turbocharger, but before it enters the engine for the most “optimum power from the combustion process within the engine” (Vestas Aircoil, 2017).

The charge air cooler is intended to cool the charged air temperature from 180 – 220 degrees (T_2) down to 40 degrees (T_1), the charge air cooler accomplishes this by using cooling water.

According to GrundfosX (2019), a transitional crossflow heat exchanger the thermal efficiency across a square cross section is around 40-65%. In addition to this, a counter-flow or dual crossflow heat exchanger are normally used when a larger thermal efficiency is needed such as 75-85%.

The charge air cooler has a total amount of 16 fins, this is “to increase the heat transfer from the surface” (Spakovszky, et al., 2019). In addition, the charge air cooler has a thermal conductivity of 238 (W/m.K) and the fins’ “shape and thickness are undulated, 0.5mm” (Sandu, 2015). The charge air cooler would be operated as a crossflow heat exchanger, where the fins are visible to the flowing fluid which cools or heats it by the high thermal conductivity which allows heat to be increased while being conducted from the wall through the fin.

Thermal Analysis of the charge air cooler

“The first law of thermodynamics relates changes internal energy to heat added to the system and the work done on the system” (physics.bu.edu, 2019).

Systems that involve energy in the form of heat and work is the study of thermodynamics.

Therefore, theoretically developing a thermodynamic system in Microsoft Excel to show the gas restrained by the piston in a cylinder could have been executed. However, further theoretical work could have been done to show the changes in temperature from a standard 181hp 1.5 eco-boost engine from when where the group could have mapped the ECU to give the engine 250hp. This would show the change in temperature the engine will produce and how much cooling would have been needed for the charge air cooler.

By creating a PV diagram graph, as shown in Figure 14, will show the pressure given from the hybrid turbo at each stage of degree. According to (Ring, 2017), if there is a higher load of pressure this will mean there will be a higher intake of temperature given.

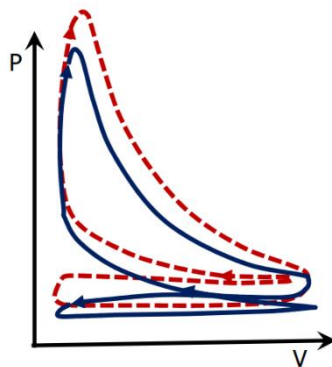


Figure 14: P-V Diagram graph (Ring, 2017)

However, according to (Ring, 2017), if the compression ratio and efficiency was increased, this will allow more cooling to the charge air.

This virtual representative design in Figure 15 shows all relevant cooling features for thermal management; such as “split cooling, integrated exhaust manifold, engine oil cooler and the integrated thermal management module” (Ring, 2017).

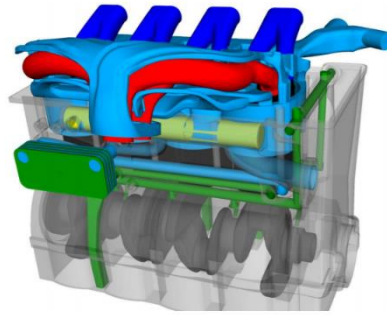


Figure 15: Thermal Management on the engine (Ring, 2017)

In Figure 16, shows the “water cooled charge air integrated with the intake manifold” (Ring, 2017), which shows the heating within the ports, and how using crossflow can give more efficiency.

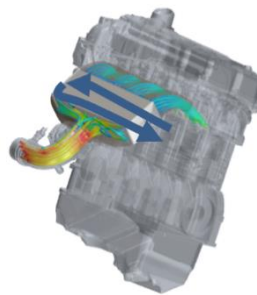


Figure 16: Water cooled charge air integrated with the intake manifold (Ring, 2017)

Theoretically, showing how the charge air cooler is operated in the Zentrack chassis could have been done. However, Figure 17 shows that while operating the charge air cooler temperature is up to around 120 degrees on a virtual test track.

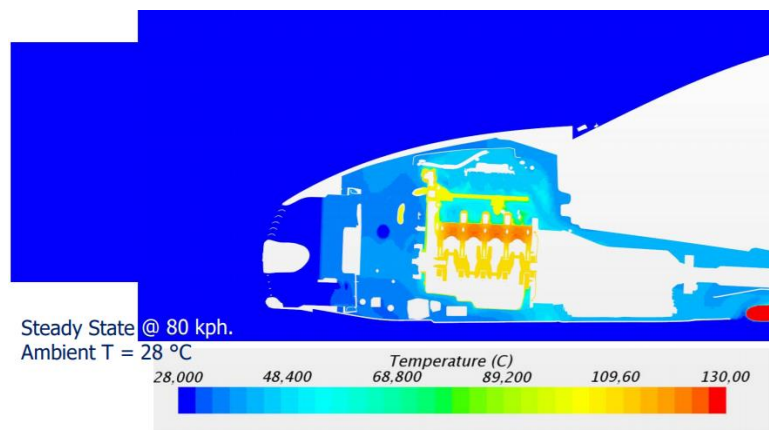


Figure 17: CFD, charge air cooler temperature (Ring, 2017)

Thermal Design and Development of Intercooler

According to Pawar (2016), the theory behind the design of a heat exchanger involves the heat transfer between the two fluids with a mechanical pump which is expected to overcome the fluid friction and allow the fluids to move through the heat exchanger. By increasing the fluid-flow velocity, this can create a higher rate of heat transfer per unit of surface area; friction power also increases with flow velocity.

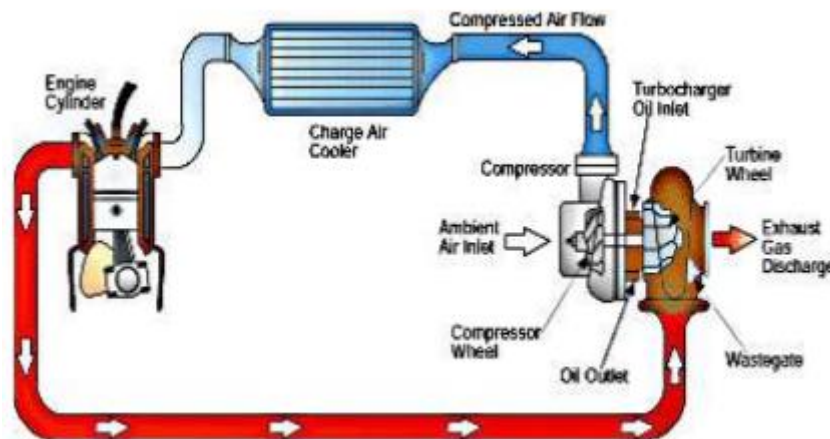


Figure 18: Design of Heat exchanger (Pawar, 2016)

Figure 18 shows a design concept of the turbocharger operation and where the charge air cooler is located. According to Pawar (2016), as the air is delivered to the induction-manifold, from the turbocharger, creates turbulence which causes a high result of heat. The engine power depends on the mass of air in the cylinder per cycle, which will increase the charge pressure ratio. If the charge temperature is relatively low at a constant pressure, the volume will be lower; hence, there will be more air charge entering the cylinder. The turbocharger will produce temperatures of around 120 degrees and remains uncooled before entering the engine, this means there will be a power loss which could be as high as 38%. The intercooler will reduce the temperature down to 60 degrees and raise the engine power to 20%, with a boost-pressure ratio of 1:6:1 or more. “The function of an intercooler is to transfer heat from the compressed charge to another source which is at a lower temperature, such as the engine’s cooling system, or directly to surrounding atmosphere” (Pawar, 2016).

3.6.2 Analysis

Due to many gateways, the group did not have the parts needed to run the engine and also the ECU to develop more BHP to the engine. Therefore, no results are available; however, by theoretically researching what could have been the outcome, if the group was able to run the engine on the dynamometer.

The results to follow are from Devi Sankar & Kishore's (2016) journal article entitled *Thermal Analysis of Water Cooled Charge Air Cooler in Turbo Charged Diesel Engine*.

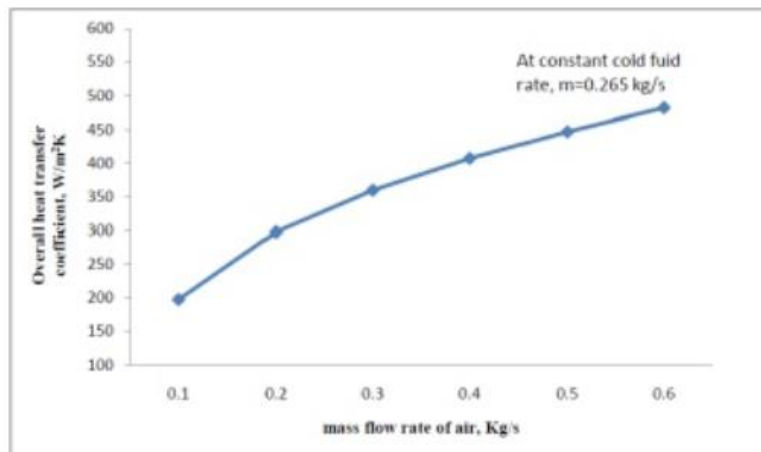


Figure 19: Heat transfer Coefficient ($\text{W/m}^2\cdot\text{K}^{-1}$) vs Mass flow rate (kg/s) (Devi Sankar & Kishore, 2016)

Figure 19 shows the overall heat transfer coefficient ($\text{W/m}^2\cdot\text{K}^{-1}$) along the mass flow rate of the air (kg/s). As can be seen from the graph, as the mass flow rate of the air increases, so does the overall heat transfer coefficient ($\text{W/m}^2\cdot\text{K}^{-1}$). According to Devi Sankar & Kishore (2016), the overall heat transfer will depend on both hot and cold fluids and surface areas from the heat transfer coefficient.

Generally heat transfer coefficient is small on air side. With the increase in mass flow rate of air, the Nusselt number increases, resulting in higher heat transfer coefficients. So with increased in mass flow rate of air from 0.1 to 0.6 kg/s the heat transfer coefficients increases, resulting in increase of overall heat transfer coefficient (Devi Sankar & Kishore, 2016, p. 196).

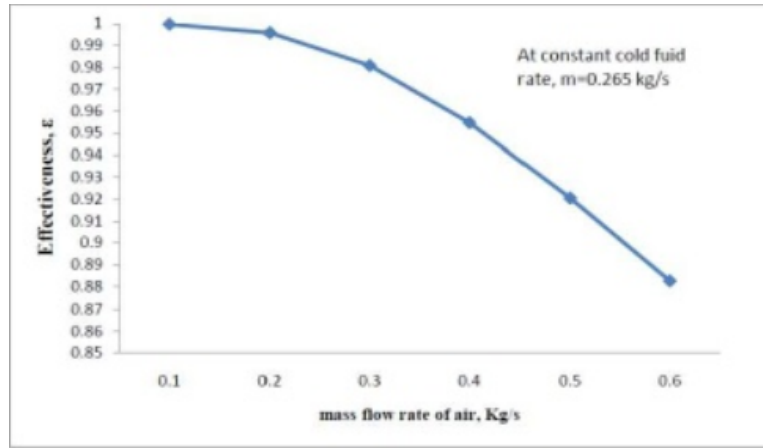


Figure 20: Mass flow rate vs Effectiveness (Devi Sankar & Kishore, 2016)

Figure 20 shows the mass flow rate of air in a charge air cooler over the effectiveness of the heat exchanger. According to Devi Sankar and Kishore (2016), they show as the mass flow rate of the air increases from 0.2kg.s to 0.6kg.s the effectiveness of the heat exchanger will rapidly decrease.

The results which follows are from Carl-Ziess-Ring (2017), document from web page entitled *Thermal influence on engine intake air*.

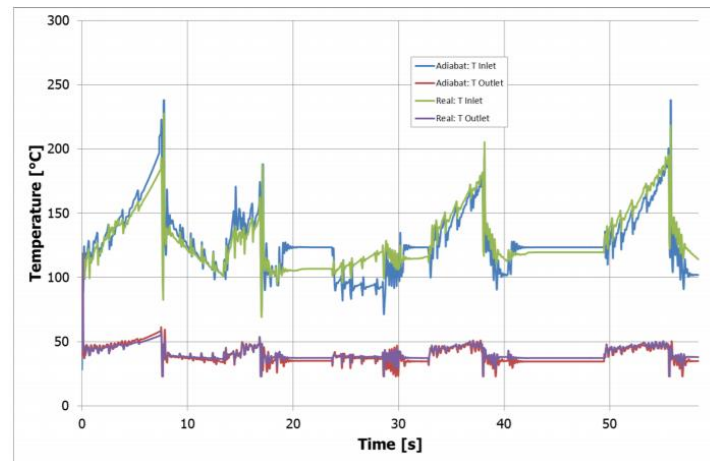


Figure 21: Temperature against Time (Ring, 2017)

Figure 21 shows the ambient T inlet and outlet, and the real T inlet and outlet, at a constant coolant volume flow through the water pump as this is a water to air charge air cooler tested via a virtual track. As can be seen from the graph, both trends of ambient and real inlet and outlets match, which demonstrates what temperatures we can expect to see the rate of the charge cooler from an adiabatic simulation and a conductive simulation.

The results to follow are from Salmon, et al., (2017), journal article entitled *Numerical Investigation on Various Heat Exchanger Performances to Determine an Optimum Configuration for Charge Air Cooler, Oil and Water Radiators in F1 Sidepods*.

As follows, these results correspond with the F1 charge air cooler, oil and water radiators, but from a theoretical point of view, these are some results the group could expect or to compare. Salmon, et al., (2017) uses CFD study to show the rate of mass flow against pressure drop and the rate of heat transfer.

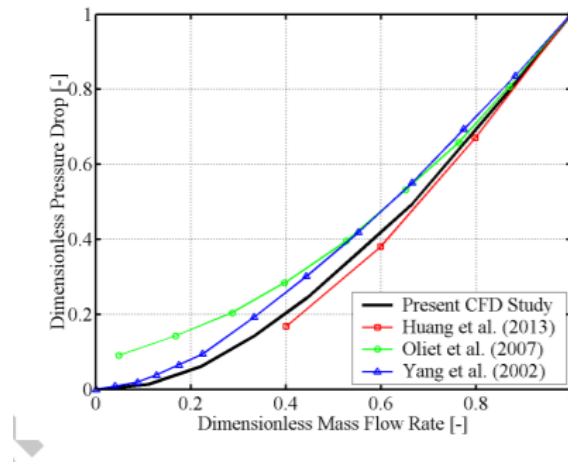


Figure 22: Dimensionless, Pressure vs Mass flow rate (Salmon, et al., 2017)

Figure 22 shows, as the dimensionless mass flow rate increases, so does the dimensionless pressure drop. These results all compare from three different people and also the present CFD study; however, all the trends seem to follow each result.

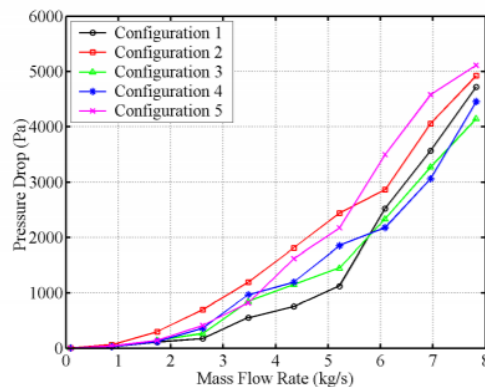


Figure 23: Actual Pressure vs Mass Flow Rate (Salmon, et al., 2017)

Figure 23 shows the increase in pressure drop over the mass flow rate, when compared to Figure 22, the dimensionless graph, pressure v mass flow rate reacts differently. According to Salmon, et al., (2017) the reason for the pressure drop to react differently is “a result of the time-dependant and sparadic seperations that occurs on the rearward facing surfaces of the heat exchanger cores in the laminar-to-turbulant transitional flow regime”. Configuration 4 shows the most reliable trend in this graph, (Fig.23), as the values do not vary as much as the other configurations and is a more reliable result to consider.

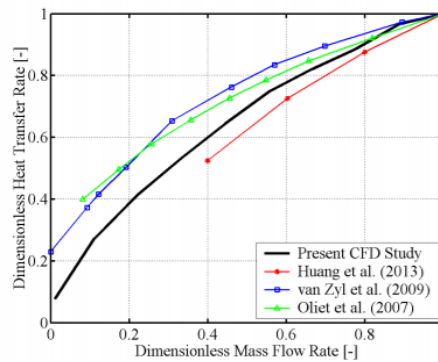


Figure 24: Dimensionless Heat Transfer vs Mass Flow Rate (Salmon, et al., 2017)

Figure 24 shows the dimensionless heat transfer increasing as the mass flow increases. Salmon, et al., (2017) states the main performance metric for a heat exchanger is when it can reject the thermal energy emmited per unit of time for the cooling air mass flow rate. During motorsport applications, a higher heat rejection from the heat exchanger per unit of volume is important as this will allow the smaller heat exchangers and tigher packaging to the “rear body-work to the centre line of the car, which improve the rear-end aerodynamic efficiency of the vehicle” (Salmon, et al., 2017).

Therefore, the best heat exchanger configuration is the one which rejects the most thermal energy over each unit of cooling air mass flow rate and the least pressure drop.

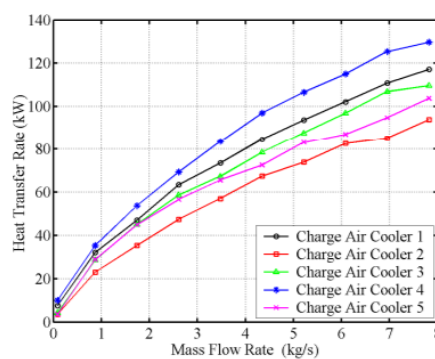


Figure 25: Actual Heat transfer vs Mass Flow Rate (Salmon, et al., 2017)

Figure 25 shows the same trendlines as the dimensionless values (Fig.24). As stated in Figure 3-33, charge air cooler 4 also shows the highest heat transfer rate per mass flow rate; this is because the charge air cooler is “placed upstream and it receives the cooling air at its nominal temperature by having a thinner section but larger projected face area”.

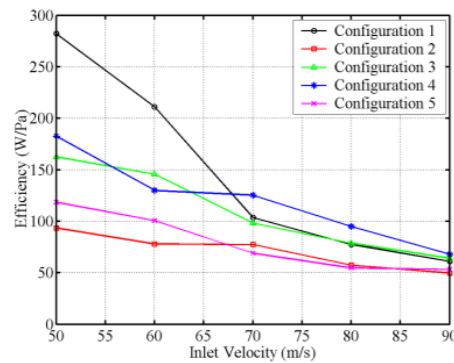


Figure 26: Efficiency vs Inlet Velocity (Salmon, et al., 2017)

Shown in the graphs above, configuration 1 and 4 has to be the best out of all 5 configurations. This is because they have the highest efficiency, highest heat transfer rate and a relatively low pressure drop. This is a good example to compare each charge air cooler for the most optimum heat exchanger.

Ifan Hopkins

3.7 Cooling R&D

3.7.1 Work content

	④	Name	Duration	Start	Finish	Predeces...	Resource Names
180		Research	20.125 days	08/10/18 09:00	05/11/18 10:00		
181		Stock charge air cooler	5 days	08/10/18 09:00	15/10/18 09:00		Ifan Hopkins[50%];Josh Bass[50%]
182		Stock water pump and thermostat	5 days	15/10/18 09:00	22/10/18 09:00		Ifan Hopkins
183		Limiting factors of the stock cooling system	5 days	22/10/18 09:00	29/10/18 09:00		Ifan Hopkins[50%];Josh Bass[50%]
184		Thermostatic oil cooler	5 days	29/10/18 10:00	05/11/18 10:00		Ifan Hopkins[50%];Josh Bass[50%]
185		Cooling upgrades	47 days	28/01/19 09:00	03/04/19 09:00		
186		Packaging of upgrades	12 days	28/01/19 09:00	13/02/19 09:00	175	Ifan Hopkins[50%];Josh Bass[50%]
187		Charge air cooler	15 days	13/02/19 09:00	06/03/19 09:00	95	Ifan Hopkins
188		Thermostat	5 days	06/03/19 09:00	13/03/19 09:00		Ifan Hopkins
189		Customise computer controlled water pump	5 days	20/03/19 09:00	27/03/19 09:00		Ifan Hopkins
190		Increase cooling ducts	5 days	27/03/19 09:00	03/04/19 09:00		Ifan Hopkins
191		Integrating the engine on the dyno	13 days	10/12/18 09:00	17/01/19 09:00		
192		Research into making an engine dyno cycle	4.875 days	10/12/18 09:00	14/12/18 17:00		Ifan Hopkins[50%];Josh Bass[50%]
193		Tutorial session with Mark on how to read data from dyno	0.547 days	11/01/19 09:00	11/01/19 14:22	31	Ifan Hopkins[20%];Josh Bass[20%];Danny Richardson[20%];Stuart Harries[20%];Mark Falls[20%]
194		Create engine dyno cycle	2 days	15/01/19 09:00	17/01/19 09:00	193	Ifan Hopkins[50%];Josh Bass[50%]
195		Pre dyno run checks	0.063 days	14/01/19 09:00	14/01/19 09:30	141	
196		Cooling circulation system	0.063 days	14/01/19 09:00	14/01/19 09:30		Ifan Hopkins
197		Exhaust extraction system	0.063 days	14/01/19 09:00	14/01/19 09:29		Ifan Hopkins
198		Cooling fan test	0.063 days	14/01/19 09:00	14/01/19 09:30		Ifan Hopkins
199		Calculations	40.125 days	04/02/19 08:00	01/04/19 09:00		
200		Collect data from all sensors (Temperatures/Pressures/Volumes)	2 days	04/02/19 08:00	05/02/19 17:00		Ifan Hopkins[50%];Josh Bass[50%]
201		Thermodynamic calculations for engine cooling	12 days	06/02/19 08:00	21/02/19 17:00	200	Ifan Hopkins[50%];Josh Bass[50%]
202		Thermodynamic calculations for intake cooling	12 days	22/02/19 08:00	11/03/19 17:00	200	Ifan Hopkins[50%];Josh Bass[50%]
203		Analyse dyno data	15 days	11/03/19 09:00	01/04/19 09:00		Ifan Hopkins

Figure 27 - Initial Gantt chart of personal tasks

Figure 27 shows the initial plan for the cooling research and design. Unfortunately, since the engine was missing various parts, it did not get to a stage where it could run, therefore no calculations were performed. Had the engine been running temperature data could have been collected and compared with the engine running the OEM map and the tuned map.

Parts needed

Mark Falls recommended researching which coolant pipes were missing from the engine, but after extensive research it was difficult to find replacement hoses. This is likely due to the engine being relatively new (built in 2017), therefore parts are not in abundance. A parts list for the M9DB engine was not found, therefore exactly which hoses were missing was hard to ascertain. In addition to the hoses found in Figure 28 and Figure 29 the coolant hoses for the turbo are also needed. Table 5 shows how much coolant is needed and its cost, and the price of an auxiliary belt (to run the water pump).

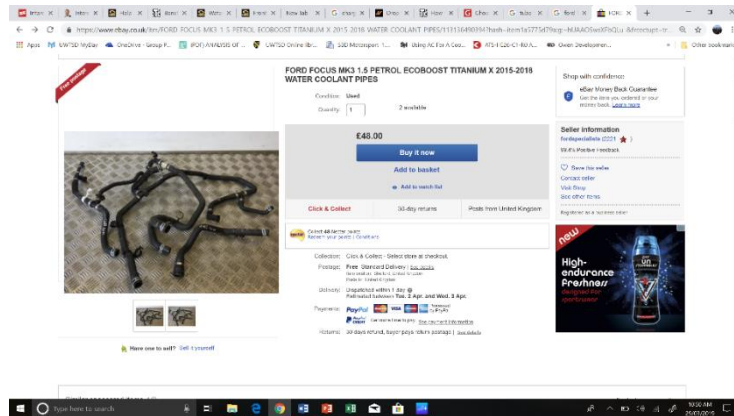


Figure 28 - Ford coolant hoses needed

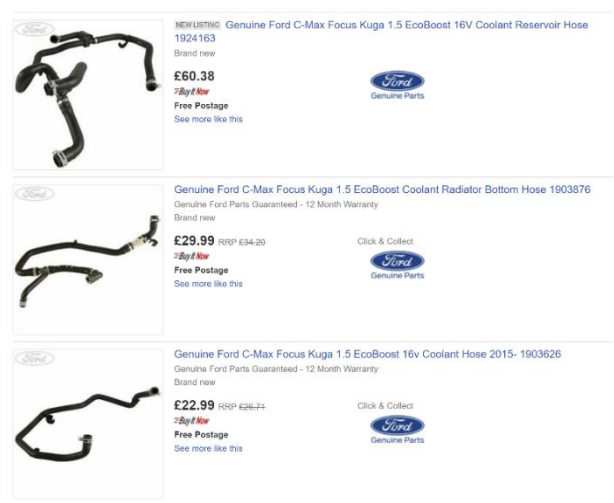


Figure 29 - Assortment of coolant hoses needed

Table 5 - Cooling parts needed to get the engine running

Ford Super Plus Premium Coolant (6.3l needed)	5l bottle - £22.75 2x 1l bottle - 2x £5.04 (www.fordpartsuk.com) Total - £27.79 Lead times 2 months for the university
Auxiliary drive belt	£22.25 (www.fordpartsuk.com) Lead times 2 months for the university

3.7.2 Background Research

The 1.5L EcoBoost engine was developed as a downsized, more efficient version of the 1.6L EcoBoost. It is a turbocharged direct injection engine, with 16 valves and twin independent variable camshaft timing. The 1.5L engine is rated Euro 6 European emission standard as opposed to the Euro 5 rating of the 1.6L. Both engines have a bore of 79mm, with the stroke being reduced from 81.4mm to 76.4mm for the 1.5L. The compression ratio is slightly lower for the 1.5L at 10:1 as opposed to the 10.1:1 for the 1.6.

‘It also boasts an oversquare design, with a 3.11-inch bore and a 3.01-inch stroke as opposed to the 1.6-liter’s undersquare 3.11-inch bore and 3.20-inch stroke; this likely explains why the 1.5-liter’s slightly lower peak torque hits earlier in the rev range.’ (Stoklosa, 2013)

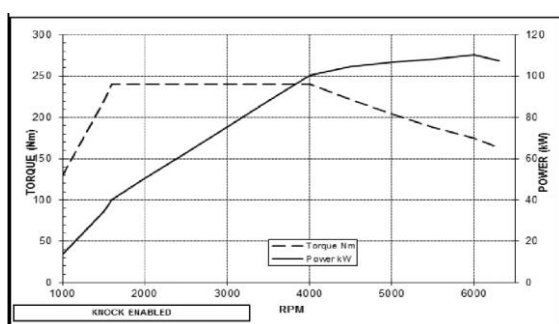


Figure 31 - 1.5 EcoBoost 110kW power and torque (found at <https://www.hendypower.co.uk/engine-sales/automotive/ford>)

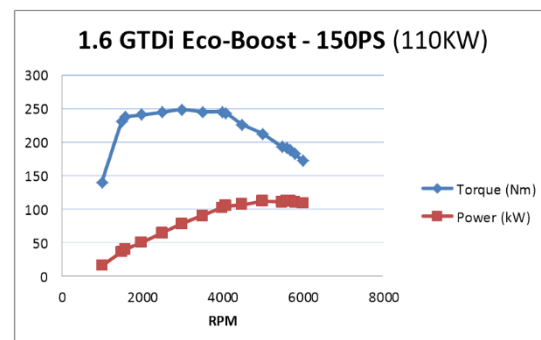


Figure 30 - 1.6 EcoBoost 110kW power and torque (found at <https://www.hendypower.co.uk/engine-sales/automotive/ford>)

The 1.5L uses a low inertia Borg Warner B0BG turbo with a vacuum actuated wastegate.

‘The turbos being so close to the combustion chambers allow them to reach maximum speed more quickly. This, along with the low-inertia rotors that drive the turbine, allows for quick turbo reaction and virtually no turbo lag.’ (Ripple, 2017)

Initially, research was conducted on the engine specifications, and the cooling system. It was found that the cooling system is very sophisticated. The aluminium engine block is an open deck design for efficient cooling, and features cast iron sleeves. The engine features a computer-controlled clutch on the belt driven water pump, which allows the engine to reach operating temperature after start-up much quicker. It also features an integrated exhaust

manifold, which helps the engine reach operating temperature after start-up much quicker and also prolongs the life of the turbo. It also increases the torque from the engine.

The stock charge air cooler is an indirect charge air cooler integrated into the intake manifold. According to Mahle this design has improved packaging, lower pressure loss and greater thermal inertia. Mann+Hummel explain how an intake manifold with a fully integrated charge air cooler ‘...offers a reduced charge air volume and quicker build-up of boost pressure which improves the response characteristic of the vehicle.’ (Mann-Hummel, n.d.)

Coolant is circulated through the CAC by an electric coolant pump. The auxiliary cooling system is filled through the same degas bottle as the engine cooling system, and a bleeder valve is provided above the cooler. Coolant flow to the turbo is supplied through a line that branches off from the engine cooling system.

3.7.3 Charge air cooler CAD model



Figure 32 - Measuring the CAC for the CAD model



Figure 33 - Measuring the CAC for the CAD model



Figure 34 - Measuring the CAC for the CAD model



Figure 35 - CAC integrated into the intake manifold

A CAD model of the charge air cooler was created using SolidWorks in order to run a flow simulation (see *Figure 36*). The aim was to design a duct to increase airflow through the charge air cooler. It soon came clear the routing of the coolant pipes inside the charge air cooler would need to be known to produce any useful results. The charge air cooler was measured (see *Figure 32, 33 and 34*), so an accurate scale model could be created. Unfortunately, without knowing the coolant pipe routing inside the CAC, those elements couldn't be modelled. The model was created despite knowing that only the outside of the model would be to scale.

Various features were used in SolidWorks to create the model. Initially the linear sketch pattern feature was used to create the external turbulators. The amount of external turbulators on each channel of the CAC was approximated for the model. The next challenge was creating the piping on the top of the CAC for the coolant, as it featured a complex pattern. The 'Sketch Picture' function was used in SolidWorks in the 'Sketch tools' dropdown and this tool allowed the image to be traced. The sweep and loft function were used in order to create the coolant piping on the top of the CAC. The revolve boss/base and revolved cut were used to create elements of the coolant inlet and outlet pipes. Various skills were learned from creating the charge air cooler, as there were some challenging elements to create.

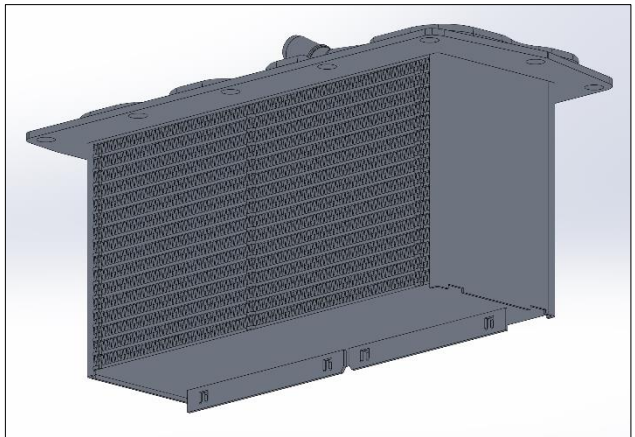
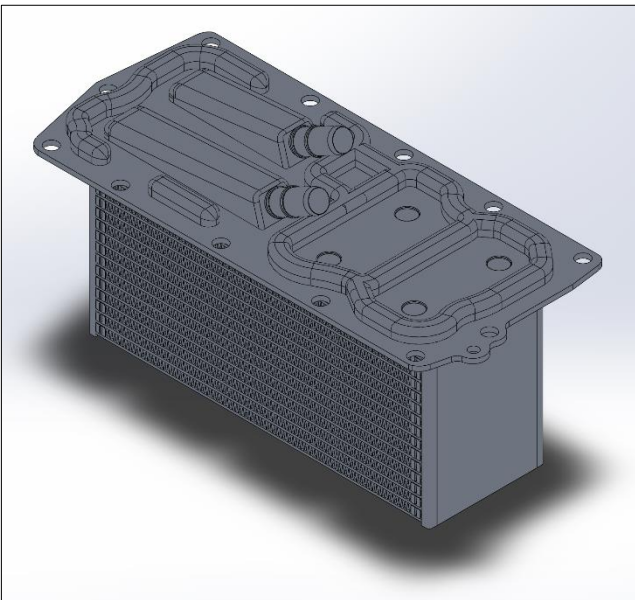
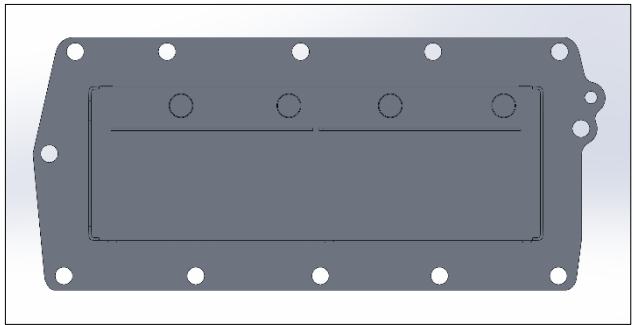
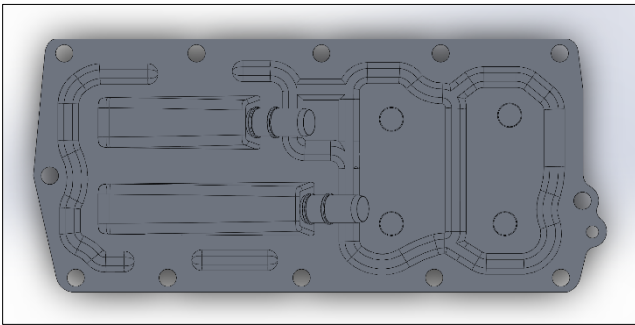
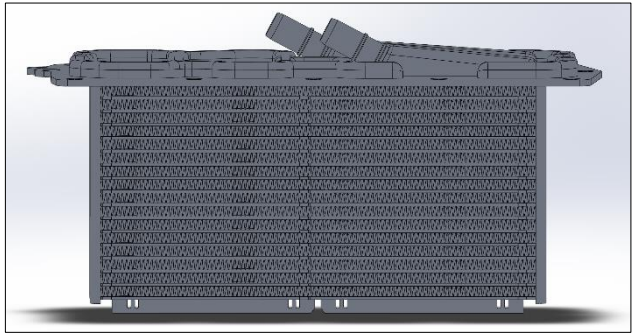
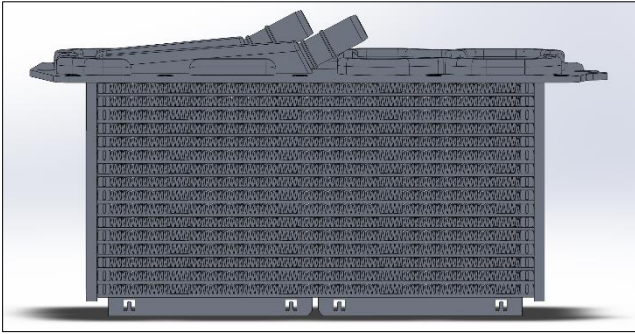


Figure 36 - CAD model of the charge air cooler

3.7.4 Charge air cooler research

When using a turbocharger (or supercharger) there is a large temperature rise which comes from compressing the intake air. Additionally, with a turbo the cold intake charge and the hot exhaust gases are in close proximity, leading to a further increase in temperature.

‘The intercooler is a radiator – or, more correctly, a heat exchanger – positioned between the turbo and the intake manifold. Its sole purpose is to get the heat out of the intake charge that the turbo put into the charge while compressing it.’ (Bell, 1997)

According to Bell (1997) the intake charge becomes denser when the temperature is reduced.

‘The increase in density is proportional to the change in temperature (measured on the absolute scale).’ (Bell, 1997)

Substituting density into the ideal gas equation gives:

$$\rho_{dry\ air} = \frac{p}{R \cdot T}$$

Where:

$\rho_{dry\ air}$ = Density of dry air [kg/m³]

p = Air pressure [Pa]

R = Specific gas constant for dry air [J/kg.K]

T = Temperature [K]

By increasing the density of the intake charge, a more efficient combustion process can take place, therefore increasing the power output from the engine. According to Mahle using a CAC reduces nitrogen oxide formation due to lower combustion temperature. Detonation is also reduced, reducing the change of engine failure when the engine is mapped for larger power.

According to Vestas Aircoil, the charge air cooler reduces the charged air intake from approximately 180° - 220° down to approximately 40°.

‘The charge air cooler is designed to cool the charged air from something in the region of 180°C to 220°C, down to around 40°C. It achieves this by using cooling water.’

(Vestas Aircoil, n.d.)

Using the Manifold Absolute Pressure, MAP_{req} calculated on page 64 (30.5psi) we see how a CAC can increase the density of air by approximately 45%.

$$\rho_{dry\ air} = \frac{210290}{287.05 \cdot 453.15} = 1.62 \text{ kg/m}^3 \text{ (without CAC)}$$

$$\rho_{dry\ air} = \frac{210290}{287.05 \cdot 313.15} = 2.34 \text{ kg/m}^3 \text{ (with CAC)}$$

Research was conducted into CAC types, as the intake manifold with integrated indirect CAC may be limiting the power available from the engine if the engine were to be remapped.

Indirect charge air cooler

Figure 37 shows the OEM indirect (water to air) CAC integrated into the intake manifold fitted to the EcoBoost engine.

The charged air is cooled by a coolant (water/antifreeze mix), and since water and aluminium has a far greater heat transfer coefficient than air and aluminium efficiency is improved. The coolant is cooled by external air in a separate low temperature heat exchanger.



Figure 37 - OEM indirect charge air cooler

Mahle explain how indirect charge air cooling offers improved packaging since the low temperature cooler required can be made more compact therefore increasing the free space at the front end of the vehicle. Additionally, thinner coolant lines can be used as opposed to the large-volume charge air hoses used for the direct CAC.

‘It is easy to get large internal flow area inside the water IC, since the most usable cores for this purpose are often air units with the flow reversed.’ (Bell, 1997, p. 67) (See Figure 38)

Bell (1997) explains how a smaller indirect CAC can be used as opposed to a larger direct CAC.

According to Mahle, due to the reduced volume of the indirect CAC, it can have up to 20% less pressure loss than a comparable direct CAC.

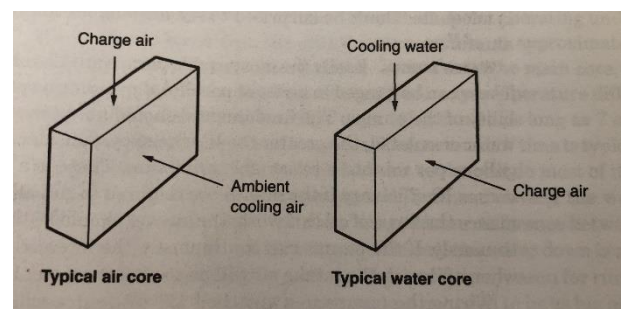


Figure 38 - Indirect and direct charge air cooler flow directions

‘This results in greater density recovery and higher volumetric efficiency in the cylinder.’
(Mahle, 2011)

Engine Basics explain how direct CAC the cooling air have flowing across the core, and the intake charge air through the cores. Indirect CAC have the cooling water flowing through the core, and the intake charge air flows across the core. (Engine Basics, n.d.)

With the CAC being integrated into the intake manifold the charge air routing is reduced, resulting in better response from the turbo as supported by Marsteller (2015);

‘Because the intercooler is so close to both the turbocharger and intake systems, piping will be extremely short, which allows for a shorter airflow route and less boost lag.’

(Marsteller, 2015)

According to Swift Performance Technology indirect CAC biggest advantage over direct CAC is its ability to lower the charge air intake temperature below the ambient air temperature. In order to do this ice water can be used as a cooling medium, like in PWR Performance Product’s air to ice intercooler. The only limitation is they can only be used in short burst before refiling with ice water.

Despite having many advantages, indirect CAC are a more complex setup and are often more expensive than a direct CAC. Trapped air can occur in the top regions of the core, which can reduce the performance of the CAC.

Direct charge air cooler

Direct charge air coolers rely on the flow of ambient air to cool the charge air intake. At high speeds direct CAC can offer excellent efficiency. According to Engine Basics the advantage of this type of CAC is that it requires no power to work. Neither does it use any liquid, so no chance of leaks. Providing it’s seeing good airflow it won’t suffer from heat soak.



Figure 39 - Airtec bar and plate direct charge air cooler for 1.6 EcoBoost

Placement of direct CAC therefore is very important. The efficiency of a direct CAC depends entirely of its mounting position and the amount of airflow it receives. There are various different configurations, for example; front mount, top mount, side mount etc.

3.7.5 Design

Bar and plate design

Generally, bar and plate style CAC are heavier, but this can have a positive effect on performance. The more mass a CAC has the higher its ability to absorb heat. Bar and plate design are more durable than tube and fin design.

‘Sealing is also superior for bar and plate designs because the brazing sheets and bars run the entire length of the core, which provides 100 percent sealing between the charged and ambient side.’ (Kirk, 2016)

Per given area a bar and plate design can have more rows in comparison to a tube and fin design. Therefore, a bar and plate design can have a higher airflow rate resulting in greater performance.

‘...a bar-and-plate core is more efficient in transferring heat in general. The design of the bar-and-plate allows for a significantly thinner material containing the airflow, which aids in producing greater heat transfer.’ (Mishimoto Automotive, 2015)

Tube and fin design

Tube and fin CAC are generally lighter than bar and plate CAC, they are also cheaper to construct. Tube and fin designs are more susceptible to heat-soak, where the CAC doesn’t have enough surface area to cool the hot charge air. They are generally used for OEM CAC.

‘Tube and fin intercoolers have oval tubes that are extruded from thin wall material, and at higher boost levels they are susceptible to damage from the tubes ballooning.’ (Kirk, 2016)

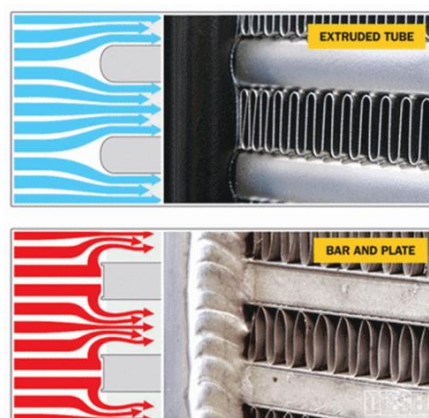


Figure 40 - Simplified flow path of a bar and plate vs a tube and fin charge air cooler. Bar and Plate designs use smooth transitions to reduce this wall effect

Other charge air cooler designs

Research was conducted into other types of CAC designs. Laminova heat exchangers use an air to liquid (indirect) style heat exchanger and are ‘...*especially designed for intercoolers in turbo and supercharged petrol and diesel engines.*’ (Laminova Ltd, n.d.)

Laminova further explain how ‘*We have a special core for air flow in charged engines. This core has longer and higher fins and fewer break-up zones (more surface) than the traditional core for liquid applications. The break up zones are not needed (like on oil cooler cores) as the air in normal conditions don’t act laminar the way oil does.*’ (Laminova Ltd, n.d.)

Research was conducted into Active Interchillers, which uses the engines air conditioning unit to refrigerate the coolant used in the direct CAC. This style system is used in the Dodge Demon and is labelled as the ‘SRT Power Chiller’. This allows the CAC to further reduce the charge air intake temperature.

Research was conducted into Reaction Engines Sabre Rocket, since it uses a very advanced pre-cooler. The pre-cooler is made from thousands of thin tubes to provide low weight and high surface area. Reaction Engines claim they are ‘...*the only people in the world with the heat exchanger manufacturing experience to bond thousands of joints in a single operation and achieve zero leakage. The joints in our pre-cooler modules are hermetically sealed, meaning that the gas which escapes can be measured by the molecule.*’

(Reaction Engines, n.d.)

Despite this technology being relatively new Reaction Engines develop, design and manufacture microtube heat exchangers and advanced thermal management, meaning it could be implemented into an automobile.

Direct charge air cooler design

According to Thulukkana (2013) when designing a heat exchanger, the construction type and flow arrangement are first selected. ‘*Selection of the construction type depends upon the following parameters:*

- 1. Fluids (gas, liquids, or condensing/evaporating) used on each side of a two-fluid exchanger*
- 2. Operating pressures and temperatures*
- 3. Fouling*
- 4. Whether leakage or contamination of one fluid to the other is allowed or not*

5. *Cost and available heat exchanger manufacturing technology*' (Thulukkana, 2013)

Below shows the charge air cooler efficiency equation. This equation can be used to determine the best performing CAC for the 1.5l EcoBoost.

$$E_i = \frac{T_{co} - T_{io}}{T_{co} - T_a}$$

Where:

E_i = Intercooler efficiency

T_{co} = Compressor outlet temperature [K]

T_{io} = Intercooler outlet temperature [K]

'A large part of the pressure loss through the IC system is determined by the internal flow area of the heat exchanger cores.' (Bell, 1997)

Where

Internal flow area = channel length x channel width x number of channels

Bell explains how the flow area can be approximated by using *Figure 41*.

Bell (1997) explains how the turbulators ensure that no laminar flow exists inside the core, increasing heat exchange whilst increasing flow loss. If space is available for a large volume core, then a core with dense tabulators should be selected, trading high turbulator drag for large internal flow areas. When space is limited, a core with low-density turbulators should be selected.

Swift Performance Technology explains how calculating the internal flow area is always a balancing act between too little and too much drag. The harder it is for the charged air to flow through the tubes then the more heat transfer from the air to the cooling fluid. If the charge air spends too long in the heat exchanger it will have a large pressure drop, meaning the turbo compressor will have to do extra work in order to achieve the same boost pressure.

Internal flow area affects pressure drop, since the larger the area, the higher the available surface for airflow to pass over. Flowing air will create a boundary layer on the surfaces inside the CAC, when the boundary layer grows the pressure drop will increase. Engine Basics explain how increasing the size of the CAC can also decrease pressure drop;

'The velocity of the air passing through the inside of the core is a function of the volume of air flowing and the passage cross-sectional area, and fluid friction is a function of velocity. So, if

the cross-sectional area increases and airflow velocity drops; then so does the friction generated, and the resulting pressure drop.' (Custom Performance Engineering, n.d.)

When the pressure ratio is too high, compressor surge will occur where the maximum flow is achieved.

3.7.6 Turbocharger Mathematics

The work per cycle, W_c [J] can be calculated using:

$$W_c = \frac{P \cdot n_R}{N}$$

Where:

P [W] = Engine power

N [revs/s] = Engine speed

n_R [rot/s] = Number of crankshaft revolutions for each power stroke

(X-Engineer, n.d.)

Therefore, at 6000RPM

$$W_c = \frac{135,000(2)}{100} = 2,700kJ$$

To calculate fuel mass flow rate, we rearrange the equation for fuel conversion efficiency, η_f :

$$\eta_f = \frac{P}{\dot{m}_f \cdot Q_{HV}}$$

Where:

P [W] = Engine power

$\dot{m}_f$ [kg/s] = Fuel mass flow rate

Q_{HV} [J/kg] = Lower heating value (Q_{HV} of petrol = 43.44MJ/kg)

(X-Engineer, n.d.)

Assuming conversion of chemical energy to kinetic energy is 33% efficient:

$$\dot{m}_f = \frac{135000}{0.33(43,440,000)} = 0.00942kg/s$$

Now we can calculate Brake Specific Fuel Consumption, BSFC [kg/J]:

$$BSFC = \frac{\dot{m}_f}{P} = \frac{0.00942}{135,000} = 7 \times 10^{-8} \text{ kg/J}$$

(X-Engineer,

n.d.)

Converting Kg/J to g/KW·h:

$$7 \times 10^{-8} (3.6 \times 10^9) = 251 \text{ g/KW·h}$$

(1.6 EcoBoost is rated at 245g/kW·h – so number seems correct)

Now convert from metric units to imperial using:

$$BSFC [\text{lb}/(\text{hp}\cdot\text{h})] = BSFC [\text{g}/(\text{KW}\cdot\text{h})] \times 0.001644$$

Therefore, EcoBoost's BSFC = 0.413lb/hp.h

$$W_A = hp \cdot AFR \cdot \frac{BSFC}{60}$$

Where:

W_A [lb/min] = Airflow actual

hp = Horsepower target (at flywheel)

AFR = Air / Fuel ratio

$BSFC$ [lb/hp.h] = Brake specific fuel consumption

(Garret Motion, n.d.)

$$W_A = 230.14 \left(\frac{0.413}{60} \right) = 22.2 \text{ lb/min}$$

$$W_A = 0.168 \text{ kg/s}$$

Now we can approximate Manifold Absolute Pressure, MAP_{req} (Assuming VE=1)

$$MAP_{req} = \frac{W_A \cdot R \cdot (460 + T_M)}{VE \cdot \frac{N}{2} \cdot V_d}$$

(Garret Motion, n.d.)

Where:

MAP_{req} [psi] = Manifold absolute pressure required

W_A [lb/min] = Airflow actual

R [J/kg.K] = Gas constant

T_M [F] = Intake manifold temperature

VE = Volumetric efficiency

N [RPM] = Engine speed

V_d [cubic inches] = Engine displacement

$$MAP_{req} = \frac{22.2 \cdot 639.6(460 + 130)}{1.6000 / 2 \cdot (1.5 \cdot 61)} = \frac{8,377,480.8}{274,500} = 30.5 \text{ psi}$$

Therefore,

$$\text{Gauge Pressure} = 30.5 - 14.7 = 15.8 \text{ psi}$$

To calculate compressor inlet pressure, P_{1C}

$$P_{1C} = P_{amb} - \Delta P_{loss}$$

(Garret Motion, n.d.)

$$P_{1C} = 14.7 - 1 = 13.7$$

Assuming 1 psi is lost due to air filter piping

To calculate compressor discharge pressure, P_{2C}

$$P_{2C} = MAP_{req} + \Delta P_{loss}$$

Where : MAP_{req} = manifold absolute pressure

ΔP_{loss} = pressure loss between the compressor and the manifold (assume 2 psi)

(Garret Motion, n.d.)

$$P_{2c} = 30.5 + 2 = 32.5 \text{ psi}$$

To calculate pressure ratio the equation below is used

$$\text{Pressure ratio} = \frac{\text{Compressor discharge pressure}}{\text{Compressor inlet pressure}}$$

(Garret Motion, n.d.)

Therefore,

$$\text{Pressure ratio} = \frac{32.5}{14.7} = 2.211$$

$$\text{Airflow rate (NA engine)} = \frac{cid \times rpm \times 0.5 \times E_v}{1728}$$

(Bell, 1997)

$$\text{Airflow rate (NA engine)} = \frac{(1.5 . 61) \times 6000 \times 0.5 \times 1}{1728} = 158.85 \text{ cfm}$$

$$\text{Airflow rate (Forced induction engine)} = \text{Pressure ratio} \times \text{Airflow rate (NA engine)}$$

(Bell, 1997)

$$\text{Airflow rate (Forced induction engine)} = 2.211 \times 158.85 = 351.22 \text{ cfm}$$

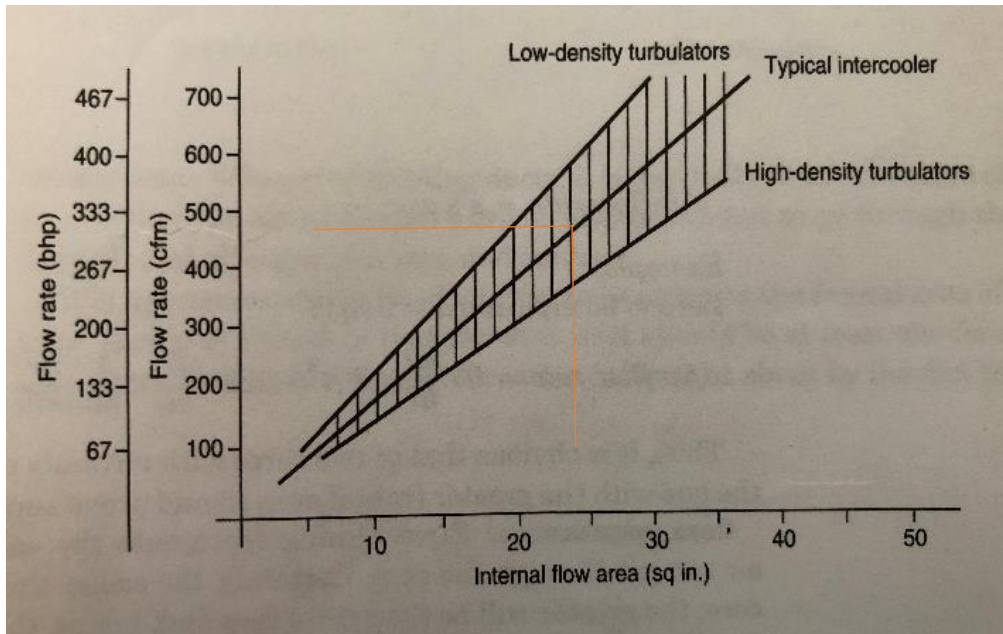


Figure 41 - Internal flow area approximation from flow rate (Bell, 1997)

According to Bell (1997) if we were to upgrade the CAC to a direct type, it would need an internal flow area of approximately 18in².

Another way of approximating internal flow area according to The Math (2009) is by using the suggested value of 6 square inches per 100hp;

$$\text{Internal flow area} = \frac{6(\text{Desired BHP})}{100}$$

(The Math, 2009)

$$\text{Internal flow area} = \frac{6(250)}{100} = 15\text{in}^2$$

Therefore, if we used a direct charge air cooler with internal area of approximately 18in² it would be perfectly capable of handling power figures up to 300hp.

The compressor map for the Borg Warner B0BG couldn't be found online. Had this been found, the approximate airflow value could have been compared to the figures on the graph.

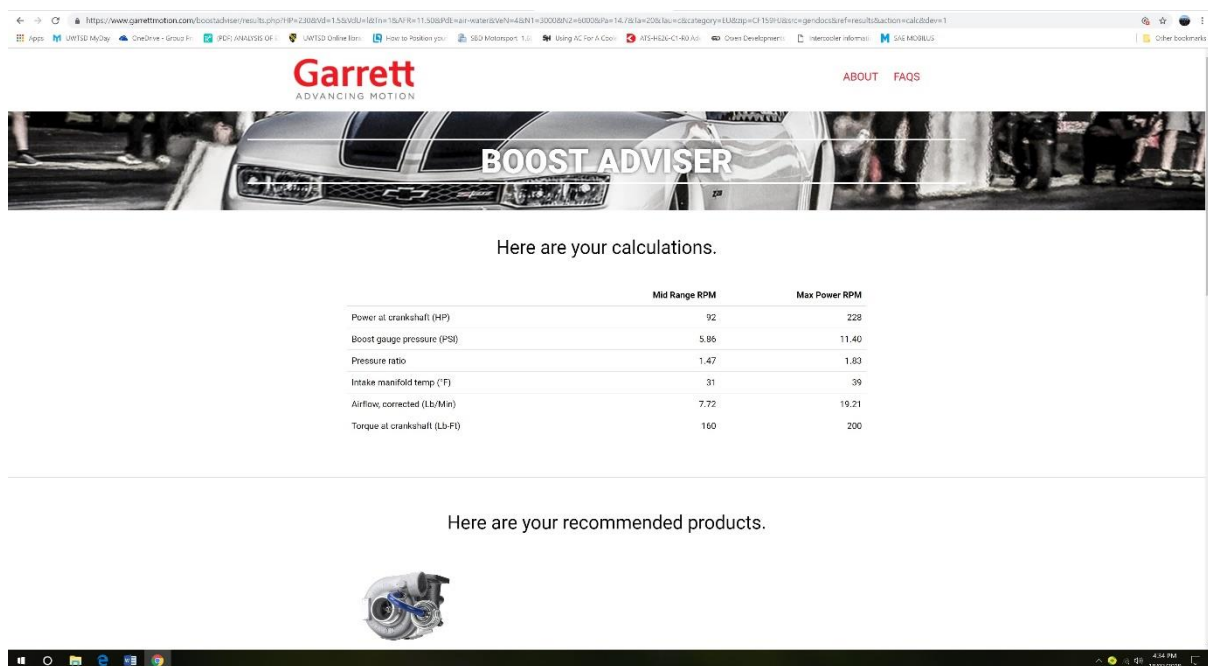


Figure 42 - Garrett's Boost Adviser

Instead Garrett's Boost Adviser was used, and the GT2052 turbo was recommended for the 1.5l EcoBoost (see Figure 42). The compressor map (see Figure 43) shows how at a pressure ratio of 2.21, the Garrett turbo pushes 17.5lb/min, therefore the approximated airflow value appears correct.

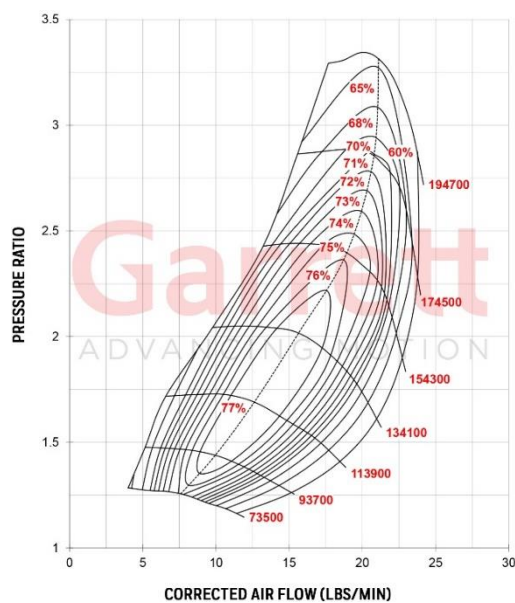


Figure 43 - Garrett GT2052 Compressor map

Carl P. Engelbrecht

3.8 Cylinder Head Development - Inlet

3.8.1 Introduction

The first point of air utilization, the intake system contributes a great deal to the breathing characteristics and therefore the performance of an engine. This area of investigation will look at performance improvements of the Ford 1.5 Ecoboost for use in the Zentrack motor vehicle through testing and analysis of the induction elements.

The components of the engine have presented various areas of improvement in reapplication as previous compact design considerations implemented are unnecessary within the new chassis allowing for redesign. The driving environment of the engine is considered 'economy', thus producing low end power at lower rpm ideal for urban usage and fuel economy. On the other hand, the required operating environment will be high performance, high revving and higher boost pressures making power higher in the rev range. However, keeping in mind, the development design specifications set in place by the managing body, John Avery. The first being, the performance improvements need to be cost effective as a large number will be produced aimed to enter the market as an affordable track day focused road car. Hence a strict price to performance goal of 40£/Kw is set. Secondly, minimal work and modification is required, reducing labour costs and inducing a quick turnaround, unless the performance achievement outweighs this factor.

The flow characteristics of the induction system is tested through the use of a steady state flow bench test and aimed to be further developed. However, to understand the restrictions the architecture of the system is to be first understood.

3.8.2 Existing Ford 1.5L Ecoboost Intake System

The whole induction system of the Ford Ecoboost engine is an advanced turbocharged system. A brief overview of the system is showcased in figure 1 (Motorreviewer, 2014). A key element within the system is the integrated charge cooler, discussed in cooling segment. Showcased in figure 7, the internal structure is drawn in figure 2, a plenum design (Bell, 2017) Air initially enters the first chamber of the intake manifold before the charge cooler through the throttle body, induced pressurised by the turbocharger. The air runs perpendicular to the charge cooler flow direction until the air is forced through the cooling fins, through a severe direction change of 90 degrees, the charge cooler in this configuration is speculated to cause a pressure difference. A large pressure difference will hinder flow rates, cooling

capacity and induce a loss in intake pressure. Once through the charge cooler air collects in the second chamber where it is scavenged by the engine, through the inlet ports, during operation. It is also observed that the inlet ports for cylinder one is recessed within a chamber extended from the main manifold body, the air would have to negotiate a radius and direction change before it would travel through the inlet orifice pair, another area of possible restriction. This design feature is intended to allow space for one of the engine mounting brackets, an absolute feature within the Zentrack chassis.

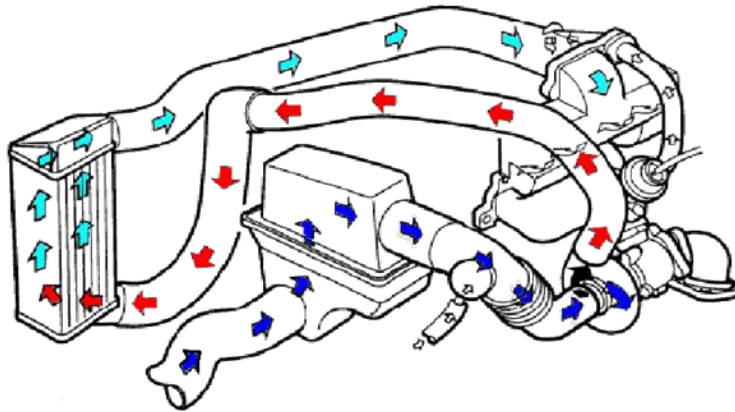


Figure 44 - Complete intake system for a turbocharged engine. The same configuration exists on the Ford 1.5 Ecoboost engine; however, the charge cooler is integrated into the intake manifold. <https://slideplayer.com/slide/10435478/>

The engine features four valves per cylinder, the most likely reasoning for this being an increased valve area providing a greater amount of gas flow. As well as reduced valve forces compared to a two-valve configuration, through lighter springs increasing longevity and reducing frictional losses. (Stone, 1999) The inlet ports are cast, identified through the rough surface finish seen in figure 1. Sharp areas are exhibited especially in the inlet ports which will induce flow separation as well as low pressure areas as the flow has to turn back on itself to fill the space, causing swirls. A design consideration from Ford to induce turbulence into the combustion chamber as it is known from recent development direct injection requires great amount of turbulence for operation due to the lack of fuel and air mixing time compared to port injecting. (Lee, et al., 2000) Development therefore on the front of port manipulation and surfacing is limited. The flow loss experienced from wall friction is at maximum 4%, therefore the benefits of machining are also minimal.

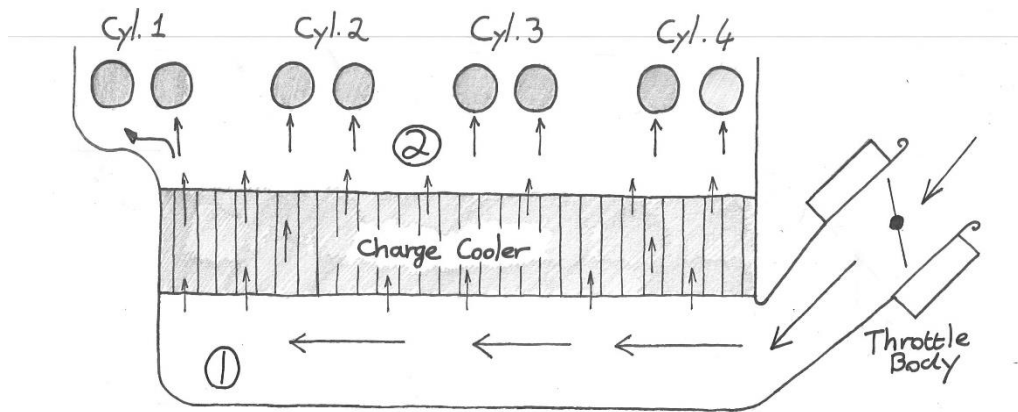


Figure 45 - Diagram of intake manifold including the integrated charge cooler.

A single Borg Warner turbocharger with unknown specification heads the induction system. The capabilities of the turbocharger are enough for greater amounts of boost pressure as discussed in section 4.1.3. However, no tests have been performed on the capabilities of the turbocharger.

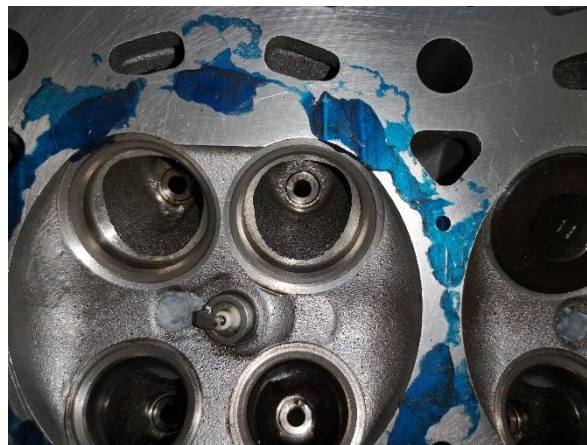


Figure 46 - Inlet ports of Ecoboost engine.

Reviewing the design of the whole induction system, the only element of plausible development is the intake manifold as this shows the greatest area of restrictions. Increase in boost pressure is also considered however without any test data and testing facilities it is a development area to be further explored once data is available.

3.8.3 Flow Bench Testing

In order to test the airflow restriction of the existing intake manifold design a flow bench test method is utilized. A test where air is blown or pulled through a cylinder head at a constant pressure, the air flow rate is then measured. Maintaining the exact same test conditions,

atmosphere conditions and test apparatus, for each different test allows for the results to be compared directly. The goal is to compare the flow coefficient, found from the maximum flow rate, between each setup and analyse the graphical trend and variations. (Jones, 2016) Both the intake and exhaust elements were tested. The exhaust testing was done for further development on the exhaust system by Aidan Chapman in the following research area.

3.8.4 Preparations

For the following test a spare Ford Ecoboost 1.5-cylinder head was found and used. Unfortunately, the cylinder head was supplied without valves, hence under the assumption that the 1.6 variants uses the same valves, a set was acquired and lapped to ensure a complete seal. Valve seal was tested through filling the inlet chamber with engine oil and examining for leaks.



Figure 47 - Image showing the lapping process of cylinder.



Figure 48 - Lapped valve compared to the original acquired valve.

In order to attach the cylinder head to the flow bench and adapter is required, mimicking the cylinder. In order to achieve this a cylinder was designed using SolidWorks CAD software to the specifications stated on the engine specifications sheet, within a tolerance of 1.5mm of the original bore (featured in section 4.1.2). In order to lower testing costs the cylinder was

designed to fit within an existing adaptor plate. The cylinder was then 3D printed out of PLA as seen in figure 4. Unbeknown an identical adaptor plate was produced previously with an elongated cylinder, therefore it was used instead. Hylomar Universal Blue non-setting joining compound was used to create an air tight seal around both the flow bench and adaptor plate as well as the cylinder head and adaptor plate.

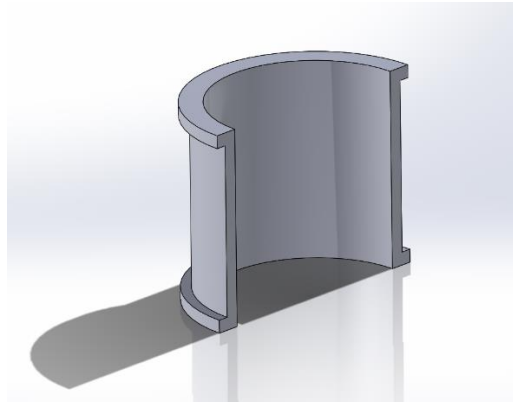


Figure 49 - Flowbench adaptor cylinder within SolidWorks.

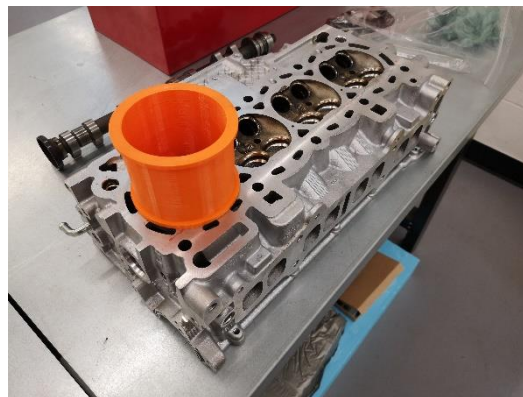


Figure 50 - 3D printed cylinder in position to mimic real engine cylinder.

Accurate valve actuating was achieved through an existing apparatus, summed up as a flat plate with two threaded holes next to each other drilled at 20 degrees, matching the valve stem travel angle. Two standard bolts where used to actuate the valve as they tightened clockwise. DTI gauges where attached as seen in figure 6 to measure the length of valve actuation, great effort was ensured that the plungers of the DTI gauges were exactly perpendicular to the bolt head, otherwise a travel in accuracy would occur.

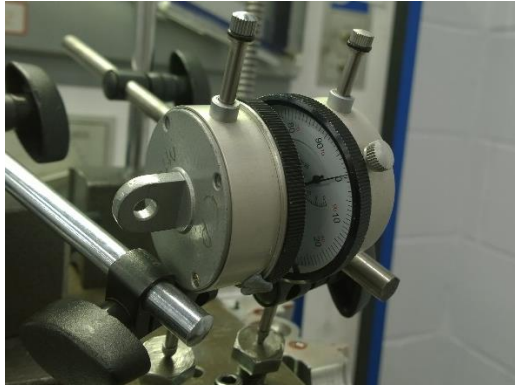


Figure 51 - DTI gauges mounted on the flow bench measuring amount of valve actuation through actuator plate.

In terms of cylinder head inlet entrance guides three different test conditions have been run using three different setups. As the test is focusing on the flow rate change that the intake manifold charge cooler induces, therefore the first inlet condition is the complete intake manifold as it would be featured in the vehicle, shown in figure 7. The throttle body blade was kept at 100% open. The second test condition was the intake manifold with the charge cooler removed, this is in effort to replicate an open to atmosphere system through bypassing the charge cooler. Thirdly the whole manifold was removed and 'Blu Tack', a reusable adhesive, was used to form a radiused inlet entrance, allowing for smoother flow. (Bohacz, 2002) The third open air test was used to support the reading of the second, in case the manifold intake ports affected the results mitigating the original intended use, for smooth ports.

The exhaust elements were also flowed, for the first condition the ports were left open with a radiused inlet similar to the one featured on the exhaust side. However, it became apparent analysing the design of the exhaust manifold that due to its combined manifold cylinders one and four would create air flow traveling semi parallel to the length of the cylinder head instead of perpendicularly outward such as with the inlet ports. Hence, without the turbocharger manifold the air would evacuate linearly, however with the manifold it would cause a change in flow direction, which would be susceptible to high pressure and therefore a loss in flow rate. Therefore, for the second test condition the turbocharger manifold was attached as observed in figure 10.



Figure 52 - Complete intake manifold system with an inbuilt charge cooler mounted on the flow bench.



Figure 53 - Intake manifold without charge cooler.

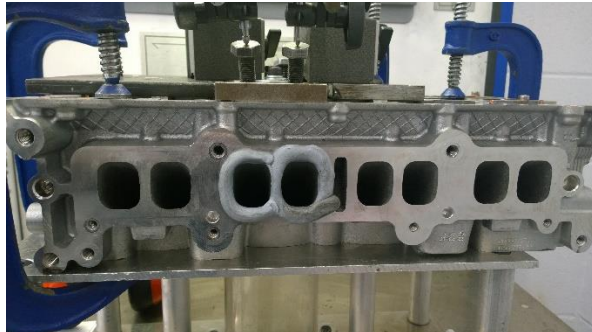


Figure 54 - Open air manifold using 'Blu Tack' as a radius's inlet entrance.

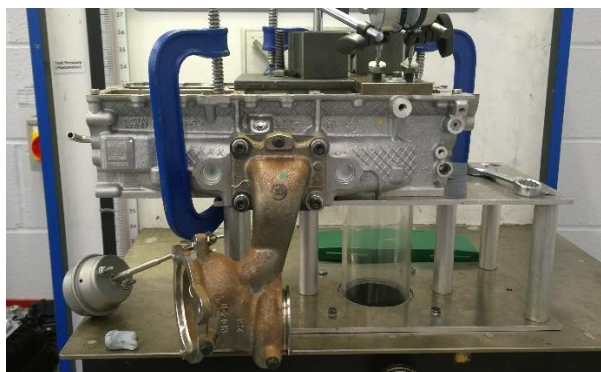


Figure 55 - Exhaust manifold attached for flow bench testing.

Before the testing commenced maximum actual valve lift, found through the measurement of the camshaft lobe, and theoretical maximum valve lift which is found through use of the equation in figure 11, requiring the measurement of the port diameter 'Di', Valve head Diameter 'D' and Seat Angle 'θ'. Theoretically this dictates the maximum valve lift of influence, a greater lift would not induce an increase in flow as the curtain area would exceed the port diameter, and hence 'Di' become the limiting factor. (Lumley, 1999) Lift for the exhaust valve is unknown, as than exhaust camshaft was not present in the acquired cylinder head, measurement request of other engines was denied, and no specification data of the standard engine was found. Data from the 1.6 variant shows a difference of 1.4mm between intake and exhaust valve lift at 6.20 mm. (Puma Speed, 2018)

$$L_{max} = \left(\frac{D}{4}\right) * \frac{\left(\frac{Di}{D}\right)^2}{\cos\theta \left(\frac{1 + \left(\frac{Di}{D}\right)}{2}\right)}$$

Figure 56 - Equation for maximum lift.

	Intake	Exhaust
<i>Port Diameter (mm), Di</i>	21.88	19.84
<i>Valve Head Diameter (mm), D</i>	30.13	24.90
<i>Seat Angle (θ)</i>	45	30

Figure 57 - Valve measurements.

Max. Actual Intake Camshaft lift = 7.392 mm

Calculated Intake Lmax = 6.509 mm

Calculated Exhaust Lmax = 5.079 mm

Figure 58 – Measured and calculated maximum lifts.

Compared to the actual intake valve lift of 7.392 mm the calculated Lmax is 6.509 mm which indicates that the valve is overshooting, a technique used to increase fill in the combustion chamber through opening the valve to greater degrees earlier after TDC.

3.8.5 Test Procedure

The test procedure is done through attaching the cylinder head, configured for the desired test condition, to the adaptor, then to the flow bench using multiple clamps, the sealant is used heavily ensuring a good seal. Valve actuation mechanism are attached, and the DTI gauges are set in place. The bolts are lightly tightened against the valve cups until there is no movement present, no more. DTI gauges are then zeroed, ensuring the plunger runs parallel with the bolt longitudinally. Consistent test pressure is at 25' of water is adhered to for each test. Flow bench operation is completed using the SF-300/600 Flowbench Operator's Manual (segment attached in appendix). The valves are opened at increments of 0.5mm until specified maximum, allowing for a level of accuracy while minimising testing time. One flow bench test takes approximately an hour. For each test condition the test was completed a total of 3 times, ensuring consistency.

3.8.6 Flow Bench Result analysis

The flow bench test data after each test was recorded, then entered in excel data tables (found in appendix), the mean average of all three test run results were taken. Math channels were used to find the flow coefficient 'Cf', through the use of the equation below. Maximum perfect flow was found through multiplying the valve area with the maximum flow rate of the flow bench. Measured flow is the flow rate of the flow bench multiplied by the valve area. The machine specific flow calibration, featured in the appendix, was used to find the final flow rate for each range. At each test a level of leakage was experienced, this was deducted from the following results in order to provide a fair comparison. The maximum leakage factor experience was 8% which equated to 0.62 l/s.

$$Valve\ Area = \frac{\pi * D^2}{4}$$

Figure 59 - Equation for Valve Area.

$$Cf = \frac{Maximum\ Perfect\ flow\ (l/s/mm^2)}{Measured\ Flow\ (l/s/mm^2)}$$

Figure 60 - Flow coefficient formula.

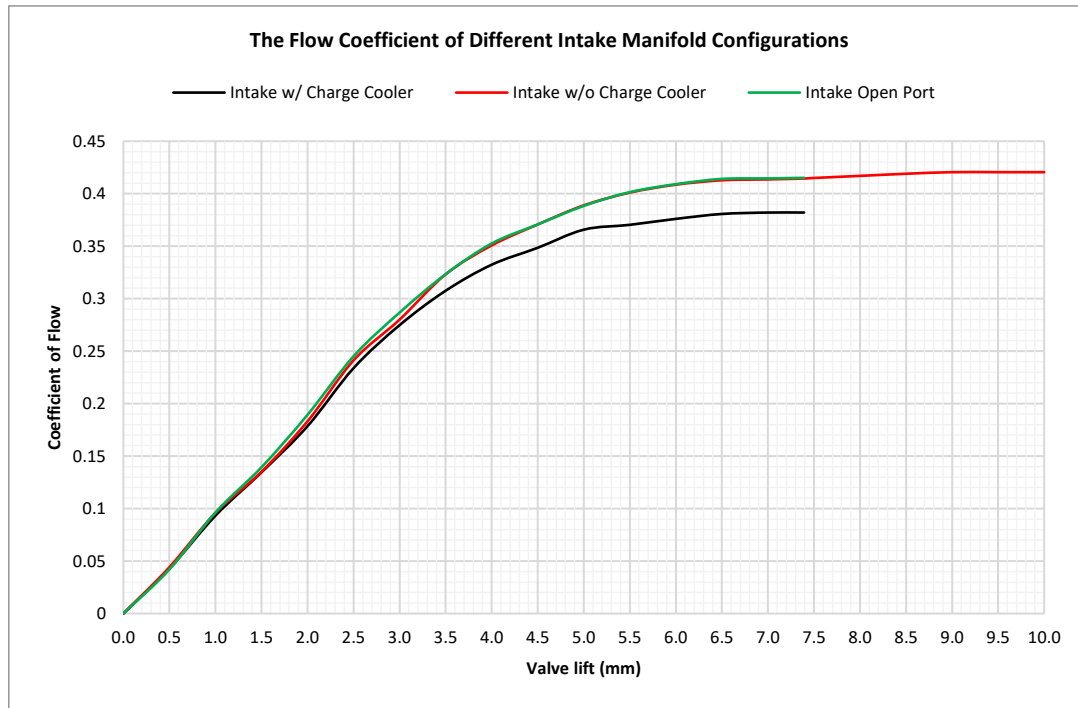


Figure 61 - Graph comparing the flow coefficient to valve lift for different intake manifold configurations.

Examining the results for the intake manifold there is a clear increase in flow past 2mm of valve lift when the charge cooler is removed. The results trend of the open intake manifold closely matches that of the completely open system with radiused inlets, validating the test procedure. However below 3.5 mm of lift there is a fluctuation of results which can be equated to averaging errors, visual recording errors or an unknown flow restriction. At 6.5mm of valve lift the flow rate increase flattens in all three test configurations indicating that the curtain area has exceeded the port diameter, hence 'D' has become the flow restriction. The lift value matches the calculated 'Lmax' of the intake valve, validating the calculation. Hence the measured camshaft valve lift is in fact overshooting as previously discussed. Reviewing the results from extreme valve lift yielded an insignificant flow increase of 0.39 l/s compared to the actual valve lift maximum.

The flow through the exhaust was also graphed. A great flow decrease with the addition of the turbocharger manifold was observed after 2.0 mm of valve lift. A variation in different cylinder ports are also observed, due to design of the integrated exhaust manifold and its geometry causing flow direction changes. This is further expanded upon the exhaust development section.

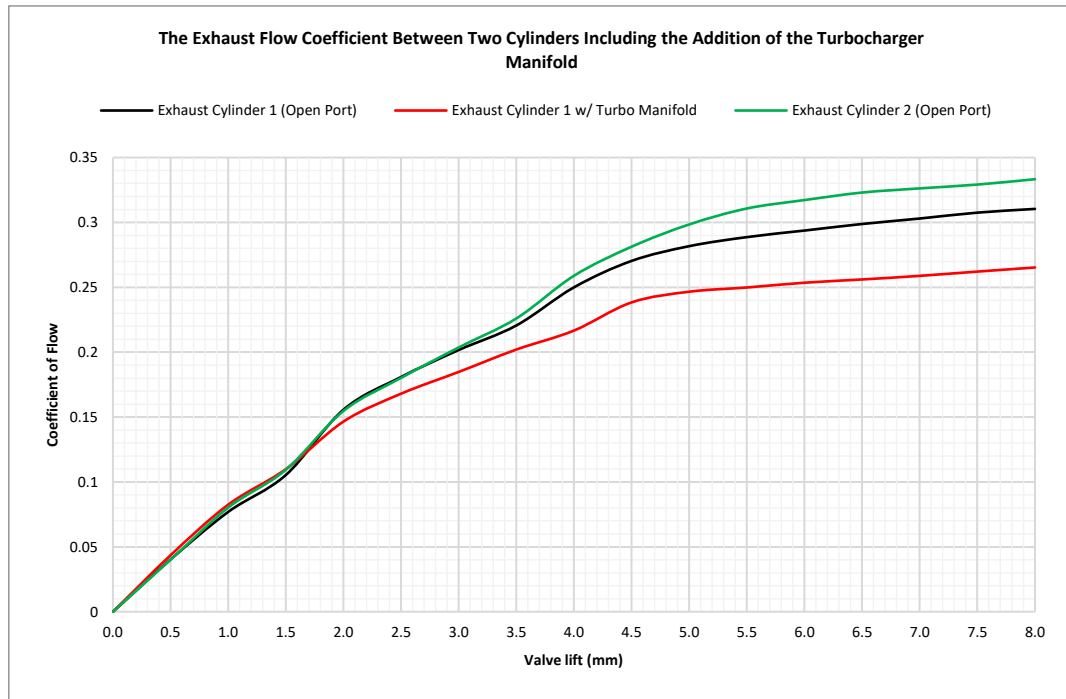


Figure 62 - Graph comparing the flow coefficient of different exhaust cylinders, with the addition of the exhaust manifold, against valve lift.

3.8.7 Conclusion and Performance Considerations

In conclusion it must be first stated that there are many factors that had there been more testing procedures, apparatuses and a functioning engine, would have aided the conclusion and supported the validity of the findings. That being said the understanding, testing and subsequent analysis allowed for aid to the understanding of the restrictions that reside within the induction system and therefore the areas of focus when improving for a performance application. The findings to head further development can be summed up in the following key points,

- There is relatively no benefit to altering the shape of the cylinder port as this will most likely hinder the induced air turbulence within the cylinder negatively affecting combustion. The benefits gained from re-surfacing of the intake ports will yield a relatively poor level of improvement compared to cost factors, a maximum flow increase of 4% may be achieved naturally however this is most likely mitigated by the turbulence inducing step.
- The main area of flow restriction observed from flow bench test data is the integrated charge cooler. Due to the 90 degree change in air motion, allowing for a possible pressure difference to occur, hence hindering the input pressure of the turbocharger. Ideally this would be further tested on a running engine with pressure sensor probes on either side of the charge cooler. An ideal improvement would be a performance plenum tapering to the

last cylinder, providing even flow compensation, and a larger more efficient external water to air charge cooler. This would also aid turbocharger efficiency, lowering the possible pressure differential, likely increasing turbocharger longevity due to lighter operational requirements.

- Thirdly, further investigation of turbocharger pressure and operation characteristics would prove beneficial. However, without engine testing and data gathering under operation this area was currently untestable. No data sheet on the turbocharger was acquired.

The set target of a performance increase of 40 £/Kw is still a viable target however without further testing with an operational engine it cannot be validated. Clear areas of improvements have been identified which can be further investigated for viability.

Aidan Chapman

3.9 Cylinder Head Development – Exhaust

3.9.1 Work content

Port moulding

After completion of the flow bench tests, the head was removed from the bench and cleaned of any contaminants from inside and around the ports. Once cleaned, the valves were re-inserted and sealed closed to prevent leakage. Afterwards, a thin layer of grease was applied throughout the port and along any faces that would be in contact with the silicon, as this would act as a release agent, making removal of the mould easier, and prevent any sticking or tearing.

Once the preparation had been completed, the silicon was mixed at a 10:1 scale as per manufacturer recommendation, using weight as the associated ratio measurement. The silicon chosen was Polycraft GP-3481F RTV Silicone Mould Making Rubber, due to its high strength, high elasticity, high tear resistance, and relatively low viscosity, thus allowing for easy mixing and pouring, as the low viscosity would fill in any gaps easier, and the higher drying strength allowing for more force to be applied upon releasing. After mixing and pouring, each mould was left for at least the recommended 24 hours curing time needed.

Once cured, the necessary valves were removed, and the silicon was extracted from the ports. This was done using a length of welding wire, inserted into the port, to help generate an air

gap along the length of the port. This gap was critical in aiding the release of the mould from the surface of the port. Once the moulds had been lifted from the surface, a careful push/pull technique was used to slide the mould from the port.

For the exhaust, only half of the head was to be modelled, and then, using SolidWorks, the other half would be mirrored to generate the full piece. The exhaust port proved troublesome to release, due to the inbuilt manifold, and therefore had to be cut into 2 sections. The two sections comprised of the ports for cylinders 1 and the ports for cylinder 2. Once extracted, the two sections were then reattached using superglue.

Insights and instructions gained from (Anon., 2006)

Computer Aided Design

Once the port moulds had been successfully extracted and reattached, a CAD model was to be made using SolidWorks. To do this effectively, the models were 'spliced', and each section of the model was traced to provide a series of sketches to help aid the part creation. Alongside this, a tracing of the plan views of each port was also taken, including markers where each splice had been taken, to help reference where each section fitted together along the plane.

The sketches were then scanned back into the computer, and uploaded into SolidWorks, where the Box, Spline, circle and Fillet tools were used to help re-create each sketch needed.

Once all 2D sketches had been completed on their respective planes, which had been generated from the plan view reference sketches, the loft and extrude tools were used to connect each sketch and start to 'build' the port. Once all lofts were completed, the port was checked to see if anything didn't match up properly, and the models were adjusted accordingly. Once checked, the exhaust model was mirrored about the edge, to provide a full manifold model for simulation. Upon completion of the models, each one was then shelled out to 1mm, to provide the inside of the cylinder head.

Upon entering the SolidWorks flow simulation CFD software, lids were then generated at each entrance and exit to seal off the calculated volume.

Computational Fluid Dynamics

The CFD was generated using SolidWorks flow sim and was mainly used as a guide to see how the pressure was distributed internally around each port. With no real comparable data available due to not being able to run the engine, the inlet and outlet velocities were set close to that of the maximum flow bench readings (around 30 m/s), and the pressure openings were set to around 10bar of pressure. The interior surface roughness was set to 1.4 micrometres to mimic that of cast aluminium, and the mesh was generated.

The mesh was set lower than the project would have benefitted from, as interior CFD usually requires a higher mesh generation than that of exterior, and was set to an initial mesh of 3, with a minimum gap size of 0.01m. This was mainly down to the computational capabilities of the computer being used at the time due to use of the computational computer facilities not being an option, and higher mesh generations causing major crashes and failures.

However, as a general overview of the pressure distribution was the only goal of the project, again due to not gaining any comparable and useable real-life data from the running engine, the data can still be considered valuable findings.

3.9.2 Research

Computation Fluid Dynamics

What is CFD?

CFD is means of solving fluid flow using computer-based simulations and is “the analysis of systems involving fluid flow, heat transfer and associated phenomena.” (Versteeg & Malalasekera, 2007)

CFD is used when there are no analytical solutions available, therefore computational methods are required. “In such cases without a known solution, CFD is used to approximate a solution” (Udoewa & Kumar, 2012). And thus, computational methods are used to generate cheaper, faster or more convenient solutions to complex fluid flow problems. CFD usually requires a 3-step method; Pre-processing, Numerical solving, and post processing. These are usually defined by creating a CAD model or ‘volume/area’, running the CFD solver to find the specified problems, and then generating a solution.

Mesh Generation

Mesh Generation is the practice of dividing the larger model into sections, or “elements”. By generating smaller sub areas within the larger structure, the computer can analyse each section individually, therefore generating a much more accurate representation of what is happening. Meshes can be created generally, i.e. for the whole model, but can also be generated locally, being able to localize a mesh to a more intricate area, with extra refinement, again allowing for further enhancements in accuracy.

Cylinder moulding

After Searching through online forums and help sites, it became apparent that taking a mould of the cylinder head ports was a necessary in achieving the correct internal geometry required for further testing and development. Alongside this, obtaining a port mould is a great visual

reference, and gives the ability to gain a rough understanding of how the airspeed and pressure will vary throughout the system. As pressure and airspeed react as opposing values, the ideal ‘perfect’ outcome would be to keep both values as constants throughout the system, to help to contain as much energy within the system as possible. However due to boundary friction and the shape of the ports, this is impossible, so the main goal is to ensure the pressure and airspeeds vary as little as possible, and to ensure that these changes happen gradually, instead of instantaneously. (Anon., 2014)

Cylinder porting and polishing

Porting is the process of removing excess mass from the inside of the cylinder head ports, by use of grinding, to help increase airflow, and level areas of high and low pressure. This helps to boost overall performance by means of allowing “air and fuel to get in faster, while exhaust can be pushed out more quickly, too”. (Haynes, 2018)

Polishing is a step further, where grinding equipment is then used to try to generate a smoother, more ‘mirror like’ surface finish, so the gasses can flow “more freely and as such, they can flow faster”. (Haynes, 2018)

3.9.3 Experiments

CFD of pressure distribution

From the Software, two separate plots for internal pressure were able to be generated, a surface plot and a flow trajectory. These two plots can visually show on the models, how the pressure ranges along the individual faces of the ports. From each plot, it can be seen where the areas of high and low pressures are, and therefore, these can be used as a rough guide for cylinder porting, to help show the areas that would require more attention than others, to help lower pressure build-up, and increase airflow.

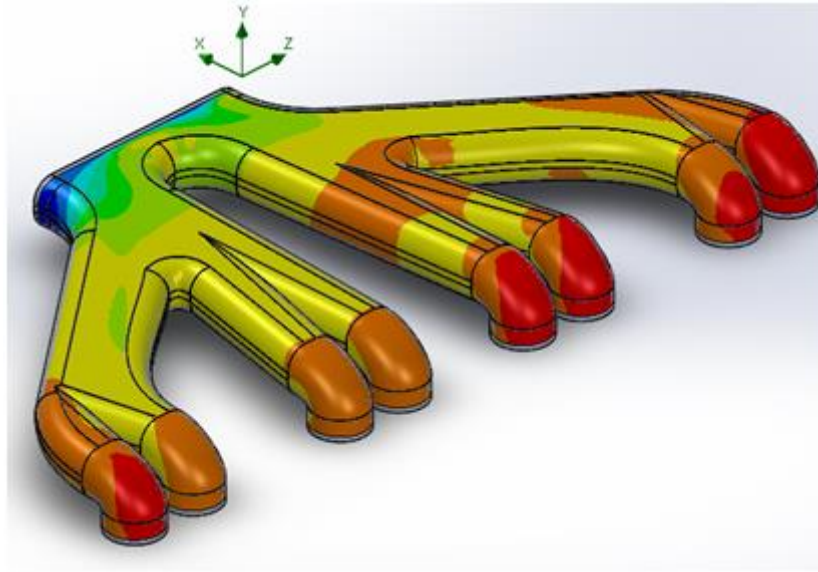


Figure 63: Wall Pressure Distribution of Exhaust

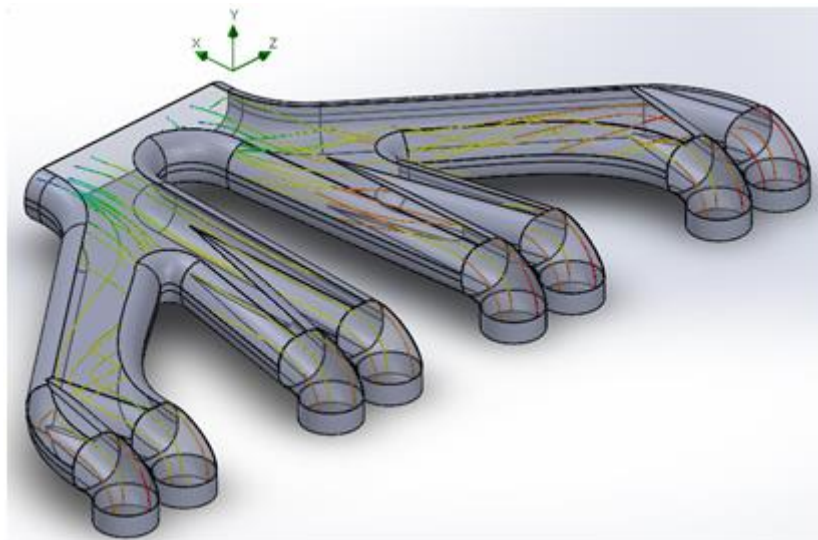


Figure 64: Flow Trajectory of Exhaust gasses

From the exhaust pressure distribution, it can clearly be seen that the entry into the system has the highest distribution of pressure. When running the simulation at the required inlet flow velocity, measured on the flow bench, the pressure distributions shown plots between 8.5 and 9.5bar. To help aid airflow, the entry points on the curved section of pipe exiting the cylinder would have to be widened, and the exit angle reduced to a smoother curve. The right-hand side of the model, on cylinder 3, shows a slight anomaly at the junction between 3 and 4, with higher pressures being generated at this point. This is more than likely down to a vortex that was generated within the model during computation.

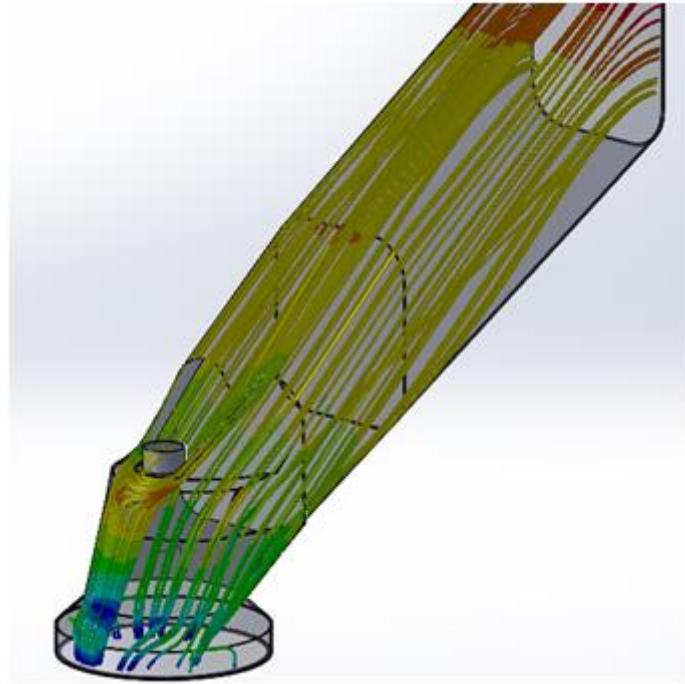


Figure 65: Flow Trajectories of Inlet Air

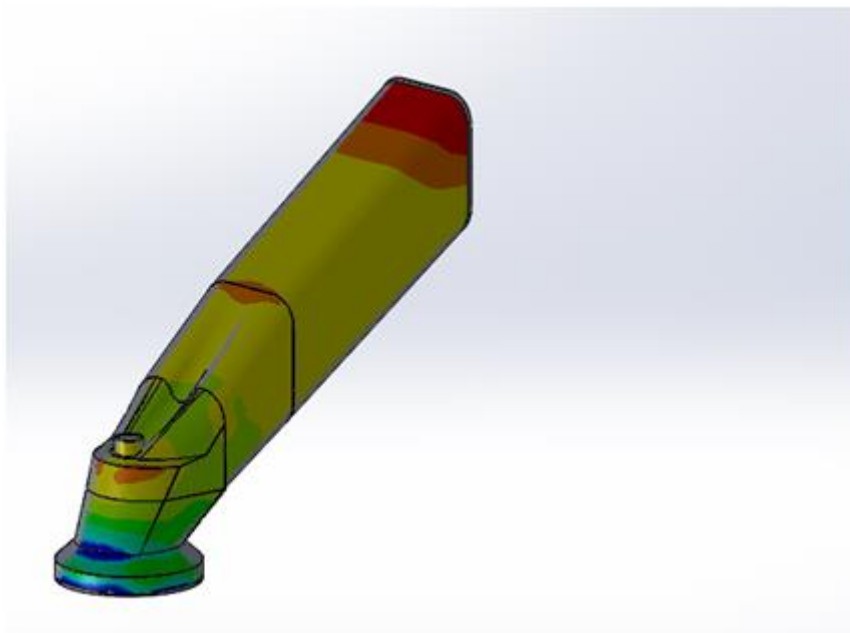


Figure 66: Wall Pressure Distribution of Inlets

The inlets show that most of the pressure builds up as the inlet narrows and the airflow has to turn to enter the cylinder, these too can be widened out and smoothed to ensure greater airflow is able to enter the cylinder. The pressure distribution shown, again using entry velocities taken from the flow bench results show a pressure distribution between 8.7 and 9.2bar.

The flow distribution can be verified by examples shown at (Vizard, 2012)

3.9.4 Designs

CAD Models

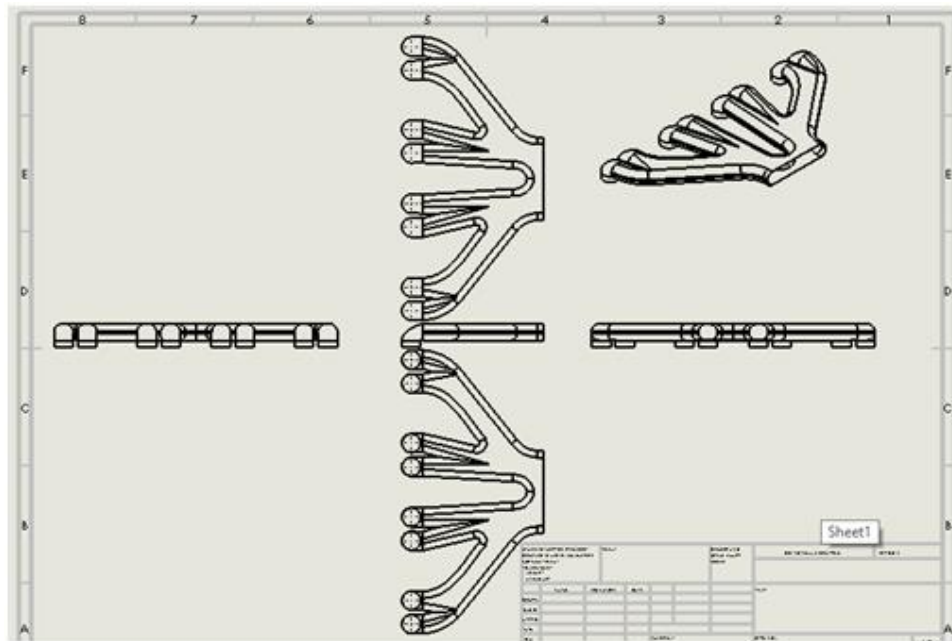


Figure 67: Exhaust CAD Drawing

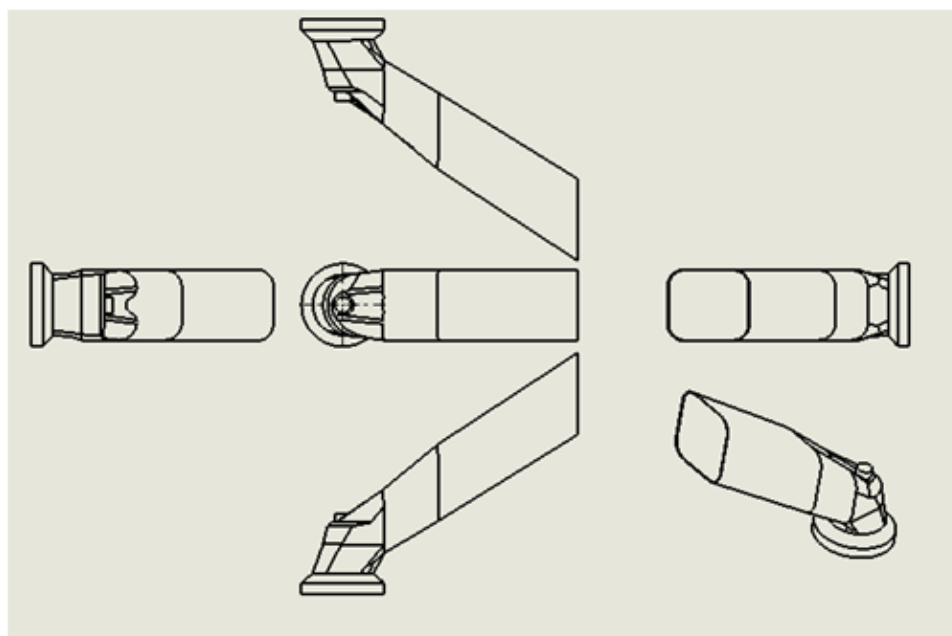


Figure 68: Inlet CAD Model

3.9.5 Development

Porting – A Theoretical development of the Cylinder head

With further time allowance and increased resource capabilities, the next step into this experiment would be to look at porting the real-life head, using the pressure distribution models generated. From these models, the pressure can be seen to accumulate just after entry into the system, and therefore, these areas would be considered the most during the porting phase. Although these areas would be worked the most, the whole system would need to be ported and polished, to help keep a consistent surface finish, and to help create a more even distribution of pressure, to reduce changes in air speed and energy loss.

To help combat this, after each phase of porting, a new mould and CAD model would be generated, and a new iteration of CFD would be run through each system. This would help to show how each change affects the airflow and pressure within the system and would show any areas that are underperforming. With this in mind, the head can then be modified further to accommodate any new findings, until the system is successfully performing at the specified standard.

Table 3: Percentage of flow losses in intake port.

No.	Source of flow loss	% of loss
1.	Wall Friction	4
2.	Contraction at push-rod	2
3.	Bend at valve guide	11
4.	Expansion behind valve guide	4
5.	Expansion 25 degrees	12
6.	Expansion 30 degrees	19
7.	Bend to exit valve	17
8.	Expansion exiting valve	31
	Total	100

Figure 69: Table showing intake flow restriction percentages (Kadir, 2008)

From Kadir's findings, it can be seen which areas of the port design contribute most to the flow loss into the cylinder. Using this table, alongside the pressure distributions gained, a clear guide can be established on what areas require the most attention porting the intakes. These include any sharp bends, or any areas where the airflow is able to expand.

This information can also be accredited to the exhaust ports, where sharp bends in sudden increments in volume can alter the desired airflow significantly. Therefore, it further back the

point that concentrating on bends and ‘voids’ would be the most efficient outcome for increased performance gains overall.

3.9.6 Analysis

Analysis of work carried out

The CFD on the cylinder head was successful, as pressure distributions can clearly been seen throughout the models, which would easily help when it came to porting. Also, there are no major problems noticeable on the simulation or on the results gained, and the models seem to be accurate and close to that of their real-life counterparts.

However, without any comparable data, or any data from the running engine, no conclusions can be drawn as to how successful the results are, or if they genuinely mimic real-life scenarios. With real data, and better computing capabilities, the models could be run more effectively within the simulation, and any results could be ‘honed’ in. Also, without real life data, it also adds more potential error into the models themselves, as without comparable results, there could be inaccuracies within the models created, that otherwise are unbeknownst.

Further work

With an increase in time and resources, further work can be carried out on the 1.5 Eco boost cylinder head. With a running engine, a thorough analysis could be run using the CFD software, with more useful results being obtained. With these results, goals could be established, in term of increasing flow through porting and other means. With this extra information, an in-depth study and further development could be conducted on the head, including actual porting, and in turn, a comparison review of before and after could be carried out.

Alongside this, further tuning and work could be conducted on the exhaust systems, starting from the manifold. Due to increased airflow through the head, the exhaust would have to be redesigned and manufactured to accommodate for this increase, or else higher pressures would be noticed, with a potential to create catastrophic damage to the system.

4. Group Meetings

It was important for the group to hold regular meetings in order to plan the project, set weekly tasks and to highlight any problems that have arisen. It was the responsibility of all the team members to attend the meetings and to present any work that had been completed. The group decided that meeting once a week in addition to meeting with the group supervisor was required to keep the project on track. Using the online library booking system, a room was booked for two hours weekly. Each meeting was recorded using the meetings template. The venue, date of the meeting and all group members present were recorded. Also, each objective that was discussed during the meetings were recorded, as was which group members the objective involved and whether the objective has been completed or still ongoing. All the meetings recorded were kept in a folder and then uploaded to this report (see Appendix).



PRIFYSGOL CYMRU
Y Drindod Dewi Sant
UNIVERSITY OF WALES
Trinity Saint David
SWANSEA - ABERTAWE

Swansea's Metropolitan University
Prifysgol Metropolitan Abertawe

Engineering Management: Meeting Record Template

Meeting: Venue: Date:

Attendees:

Name	Signature

No	A/D/I	Objective/Description	Who	When	Status
1					
2					

Figure 70 Meetings Template (Sandford, 2014)

5. Achievements and Progress Relative to the Initial Plan and Timing Chart

5.1 Cooling development

Achievements

- Measured the intercooler, created a CAD model on SolidWorks.
- Researched and planned ways to develop the intercooler and cooling efficiency of the engine.
- Found solutions and analysis to back up work conducted at a later stage.
- Calculated an approximate value for the internal flow area of a upgraded direct charge air cooler.

Delays Which Affected Original Timescale

- Unable to test the cooling efficiency through the heat exchanger in the dyno workshop.
- Unable to collect data from the dyno, as the engine was not running. This made us unable to calculate any thermodynamic equations, resulting in no calculations.
- No CAD model of the chassis, unable to optimise the best placement for the radiator.
- Limited coolant hoses for sale online, therefore we couldn't find exactly which coolant hoses are needed to run the engine.

5.2 Cylinder head development

Achievements

- Flow Bench testing carried out efficiently and generated good, clear results.
- CAD models generated from accurate port moulds.
- Self-taught SolidWorks CFD carried out effectively using the data available.

Delays Which Affected Original Timescale

- Acquisition of the correct cylinder head, due to being in use by another student, delayed time significantly.
- Sourcing of Valve and spring sets.
- Waiting for delivery of silicon for moulding.
- Silicon moulds could not be created until after flow bench tests had be completed, meaning the two projects were dependant on one another.

5.3 Dynamometer development

Achievements

- CAD model was created and FEA performed within SolidWorks of the adapter shaft.
- Enhanced and calibrated 3D printer through trial and error to find the best settings for accurate prints.
- Self-taught SolidWorks Electrical to create a basic schematic of engine power circuit.
- Knowledge and experience on different manufacturing processes.

Delays Which Affected Original Timescale

- Waiting for parts (rubber bushings) to arrive to securely mount the engine onto the dyno trolley.
- Sourcing a method to raise the engine (adjustable height mounts) took longer than predicted.
- The adjustable height mounts needed to be re drilled as the thicker rubber bushings altered the width of the mounting bracket legs, delaying the trolley fitment.
- Extensive modifications were required in order to mount the trolley onto the dynamometer frame leading to delay.
- Limited access was granted to the dyno cell without supervision.
- The Honda engine was in use for classes, lessons and demonstrations, which meant we couldn't work during those times, so the engine needed to be disconnected and then reconnected beforehand, both of which took time to do.

5.4 Engine calibration & Parts Acquisition

Achievements

- Completed the parts list with everything known in quick succession and returned it to those in control of finance for the project. This was successfully completed before the expected deadline by five days.

Delays Which Affected Original Timescale

- Finance within the project was limited and took far longer to substantiate than what was first anticipated by the group.
- Lack of communication through all parties involved in the project made it difficult to correctly upon the matters that arose because of the finance situation.

- We were unable to obtain any parts required to even begin testing, let alone development work.

6. Review of Project Successes and Failures

6.1 Cooling development

Project Successes

- Optimum internal flow area calculated for a direct charge air cooler for the 1.5.
- Data analysis research conducted, I.e., Graphs resulting in efficiency. (Provided the parts required and engine running. The cooling development would be achieved showing the same results researched.
- Researched different types of charge air cooler designs that could be used on the 1.5. Ideally a few different designs would be tested on the engine and a new manifold would be manufactured for use with a direct charge air cooler.
- Scale CAD model created but internals not accurate. Had the intake manifold been designed a flow simulation could have been conducted.

Project Failures

- No data was collected to analyse (manifold abs pressure, coolant temperatures, boost pressure, ECU).
- Unable to calculate thermodynamic equations because the required data wasn't obtained.
- The ECU wasn't delivered, so it meant we couldn't adjust the clutch-controlled water pump.
- Couldn't test the heat exchanger in the laboratory.
- Bodywork wasn't delivered from John Avery; therefore, we couldn't study the most efficient positioning for radiator in terms of its aerodynamic effect.

6.2 Cylinder head development

Project Successes

- Flow Bench Test complete.
- Cylinder ports moulded correctly, allowing for greater accuracy within CAD.
- CFD analysis Complete.

Project Failures

- No data from the live engine to enable cross-reference of results.
- Timing setbacks prevented development of head/exhaust from taking place.
- Potential unknown inaccuracies due to lack of real data.

- Difficulty generating conclusions without data comparison.

6.3 Dynamometer Fitment

Project Successes

- Engine to trolley fitment completed.
- Adapter shaft was designed and manufactured exactly to specifications; no alterations or modifications needed.
- The engine was successfully mounted onto the dyno for future students and projects to utilise.

Project Failures

- Unfortunately, a real-life test was never performed on any of the components situated on the trolley or the adapter shaft to test their ability to withstand torsional forces being applied through them while the engine is in operation on the dynamometer.

6.4 Engine calibration & parts acquisition

Project Successes

- Developed an appreciation for the health and safety aspect of working with an engine dynamometer and how to minimise the risks involved.
- Gained a simplified understanding of how to utilise the engine dynamometer to aid in the development of an ICE.
- Achieved a basic knowledge of engine control module calibration in respect to fuelling control and boost control implications.
- Expanded our knowledge on new software programmes which could pose useful for further work applications.
- Enhanced our ability to work with an external partner on an engineering project and obtained a new understanding of the potential downfalls that can occur within it.

Project Failures

- Weren't able to meet any target time projections other than to have created the required parts list. This is due to an unfortunate lack of communication and funding for this project as a whole.
- Well exceeded the time parameters set, which in reality would cause a catastrophic loss for a company which could ruin a smaller unknown manufacturer.
- Didn't get the chance to run the engine on the dynamometer meaning we couldn't put into practice any of the research of purposed plans. As a result, it made it very difficult to do any analytical work for most of the aspects we have mentioned.
- We were unable to make any advancements or gains in engine power or efficiency.
- Didn't meet the target parameters asked of us by the partner, meaning we failed in regarding the project plan.

7. Conclusion

To conclude this project, we've had a varied series of successful points that were certainly joined with an unfortunate amount of unsuccessful points, that happened to be out of our control. But we've most certainly benefited from this project, regarding how to organise ourselves within an engineering project in order to try and achieve the highest work rate while trying to maintain a good morale and interest within the chosen area. Projects we will encounter in industry will likely comprise of the same challenges negotiated in this project, therefore we feel it has prepared us for work in industry. Through each individual chapter of this report, it's clear that we took this project on with a collective mindset and worked together to create a proposal on how to enhance Ford Motor Company's 1500cc EcoBoost 4-cylinder internal combustion engine. Starting with engine control module calibration through the use of dynamometer and different drive cycle testing methods based on the application or driving style. Then there's a very clear explanation of the stages involved and what may be required for mounting an engine to said dynamometer and also how to go about engineering and manufacturing a new 'one off' part that is specific to an engine. Moving onto how the challenge of thermal management would've been handled when gaining a substantial amount more power and torque than was originally intended, whilst being constrained by placement and packaging within a small sports/track vehicle. Finally, outlining areas of concern within the cylinder head after having tested the flow of air that comes from how it's been manufactured, suggesting ways to further gain free flow through the exhaust and better air control of distribution via the induction geometry.

As for the chassis development work, that was first purposed as a task within the specified brief, was taken out of consideration shortly into the project. Because as a team, we agreed with both the client and our projects leadership that the engine development was a substantial enough project on its own. Along with the chassis having been returned to the client, so there wasn't any possible work that could've been conducted past what we had previously, partially investigated.

To sum this project up, it could've certainly been taken much further than what we've managed to achieve, but it was most definitely educational and has given those involved an experience of how real-world, small engineering projects can develop and deviate from the original plan, therefore demanding a new strategy and plan to adjust to said alterations.

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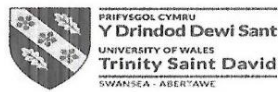
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9. Appendix



Swansea's Metropolitan University
Prifysgol Metropolitan Abertawe

Engineering Management: Meeting Record Template

Meeting: 12:00pm - 13:00pm Project meeting 2 Venue: 103 I.Q. Date: 9/10/18

Attendees:

A/D/I (Action/Decision/Information)

Name	Signature
Stuart Harries	
Aidan Chapman	
Joshua Bass	
Iwan Hopkins	
Carl-Peter Englebrecht	
Danny Richardson	
Mike Anderson	
Callum-James Hill	

No	A/D/I	Objective/Description	Who	When	Status
1	A	Viewed engine in workshop to check parts and condition	everyone	9/10/18	complete

Figure 71 Meeting 1 front page

09/10/18

No	A/D/I	Objective/Description	Who	When	Status
2	A	Met with Pedro to discuss engine condition and which engine was ours (2 in workshop)	everyone	9/10/18	complete
3	I	Went to computer room to research ECU and parts	everyone	9/10/18	complete
4		- Meeting adjourned - Next meeting scheduled 11/10/18			
5					
6					
7					

Figure 72-Meeting 1 back page

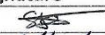
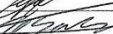
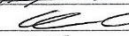
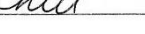
09/10/18

Engineering Management: Meeting Record Template

Meeting: 13:00pm - 15:00pm Project meeting 2 Venue: 103...115 Date: 11/10/18

Attendees:

A/D/I (Action/Decision/Information)

Name	Signature
Stuart Harries	
Mike Anderson	
Danny Richardson	
Carl Peter Englebrecht	
Joshua Boes	
Iron Hopkins	
Aidan Chapman	
Callum-James Hill	

No	A/D/I	Objective/Description	Who	When	Status
1	D	Discussed possible Job roles - Engine running - Dyno test - Fitment - ECU + wiring / control systems, cooling, Exhaust Intake	everyone	11/10/18	Complete

Figure 73-Meeting 2 front page

11/10/18

No	A/D/I	Objective/Description	Who	When	Status
2	D	Decided to split into pairs to do some background research	everyone	11/10/18	Complete
3	I	Control Systems - Danny + Aidan Engine cooling - Josh + Iron Inlet manifold - Carl P + Mike Exhaust manifold - Stuart + Callum	everyone	11/10/18	Complete
4	D	Deadline for this research has been set for 15/10/18 - Discuss in group project lecture with Pedro.	everyone		Ongoing
5	A	Created a OneDrive account to share files, research and recordings.	Mike	11/10/18	Complete
6	A	- loaned relevant books from the library	everyone	11/10/18	Complete
7		- Meeting adjourned - next meeting scheduled 18/10/18			

Figure 74-Meeting 2 back page

11/10/18

Engineering Management: Meeting Record Template

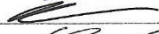
Meeting: 17:00 - 18:00 Project meeting 3

Venue: lib 103

Date: 18/10/18

Attendees:

A/D/I (Action/Decision/Information)

Name	Signature
Stuart Harris	
Mike Anderson	
Joshua Bass	
Carl - Peter Englebrecht	
Callum - James Hill	
Danny Richardson	
Aidan Chapman	
Ian Hopkins	

No	A/D/I	Objective/Description	Who	When	Status
1	A	Discussed research from previous meeting and the differences between the 2 engines	Everyone	18/10/18	Complete

Figure 75-Meeting 3 front page

18/10/18

No	A/D/I	Objective/Description	Who	When	Status
2	I	Mark still hasn't put engines on stand - Emailing Mark	Mark Fall	18/10/18	Ongoing
3	D	Discussed mounting alternator as tensioner For Aux belt and air con delete (fake pulley)	Everyone	18/10/18	Complete
4	D	Discussed prices for parts - Emailing John + Pedro parts list	Danny Stuart	18/10/18	Complete
5		- Meeting adjourned - next meeting scheduled 25/10/18			
6					
7					

Figure 76-Meeting 3 back page

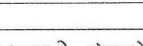
18/10/18

Engineering Management: Meeting Record Template

Meeting: 13:00pm - 15:00pm Project meeting 4 Venue: 103...1.6... Date: 25/10/18

Attendees:

A/D/I (Action/Decision/Information)

Name	Signature
Stuart Harnes	
Aidan Chapman	
Mike Anderson	
Fran Hopkins	
Carl-Peter Englebrecht	
Callum-James Hill	
Danny Richardson	
Joshua Bass	

No	A/D/I	Objective/Description	Who	When	Status
1	A	Still waiting on Pedro to respond about Parts	Pedro	25/10/18	Ongoing

Figure 77- Meeting 4 front page 25/10/18

No	A/D/I	Objective/Description	Who	When	Status
2	I	Emailed Powertorque about parts list but no response	Danny	25/10/18	Ongoing
3	A	-Team Building (Project management coursework talk)	everyone	25/10/18	Complete
4	A	Started to develop Gantt chart for the 1st Project viva and project management coursework	Everyone	25/10/18	Ongoing
5		- Meeting adjourned - Next meeting Scheduled 5/11/18 Start of project week			
6					
7					

Figure 78- Meeting 4 back page 25/10/18



Engineering Management: Meeting Record Template

Meeting: Project Week meeting 1

Venue: 103 IQ Date: 5/11/18

Attendees:

A/D/I (Action/Decision/Information)

Name	Signature
Stuart Hornes	
Mike Anderson	
Joshua Bress	
Danny Richardson	
Carl-Peter Englebrecht	
Ifan Hopkins	
Aidan Chapman	
Callum-James Hill	

No	A/D/I	Objective/Description	Who	When	Status
1	D	Discuss Job Roles Danny → Team manager Stuart → Technician/Meeting co-ordinator	Everyone	5/11/18	Complete

Figure 79- Project Week Meeting 1 front page 05/11/18

No	A/D/I	Objective/Description	Who	When	Status
2	D	Joshua → Thermal management / Engine scrutineer Mike → Head Designer Carl-Peter → Workshop Co-ordinator	"	"	"
3	D	Aidan → CFD analyst technician Callum → Dynamometer/manufacture Specialist Ifan → Head Researcher	"	"	"
4		Talk to Gareth + Dave for machining	Mike Callum-James	5/11/18	Complete
5		Talk to Paul, Pedro and Mark Fall about Stripping engine	Stuart Danny Mike Callum-James	5/11/18	Complete
6		Talk to John about using the flowbench	Aidan Carl-Peter	5/11/18	Complete
7		- Meeting adjourned - Next meeting scheduled 6/11/18 with Pedro			

Figure 80- Project Week Meeting 1 back page 05/11/18

Engineering Management: Meeting Record Template

Meeting: Project Week meeting 2 with Pedro

Venue: 1st floor IQ Date: 6/11/18

Attendees:

A/D/I (Action/Decision/Information)

Name	Signature
Stuart Harries	
Carl-Peter Englebrecht	
Aidan Chapman	
Mike Anderson	
Danny Richardson	
Callum-James Hill	
Leon Hopkins	
Joshua Bass	

No	A/D/I	Objective/Description	Who	When	Status
1	D	-Possibly sending 1.5 loom to SCS for modification	Danny Stuart	6/11/18	Ongoing

Figure 82- Project Week Meeting 2 front page

06/11/18

No	A/D/I	Objective/Description	Who	When	Status
2	A	-Sourcing 1.6 loom for ECU	Danny Stuart	6/11/18	Ongoing
3	A	-See if 1.6 mounts match 1.5 otherwise Scraping the 1.6	Mike Callum-James	6/11/18	Complete
4	A	-Possibly take 1.6 injectors	Stuart Danny	6/11/18	Ongoing
5	D	-main aim is passing Euro 6 -main project aim is how to overcome factory limits	Everyone	6/11/18	Ongoing
6		- meeting adjourned - next meeting scheduled 7/11/18 with Pedro			
7					

Figure 81- Project Week Meeting 2 back page

06/11/18

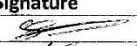
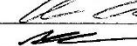
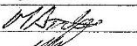
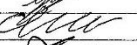
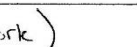
Engineering Management: Meeting Record Template

Meeting: Project Week Meeting 3 With Pedro

Venue: 1st Floor IQ Date: 7/11/18

Attendees:

A/D/I (Action/Decision/Information)

Name	Signature
Stuart Harries	
Iwan Hopkins	
Aidan Chapman	
Mike Anderson	
Carl-Peter Engelsbrecht	
Callum-James Hill	
Danny Richardson	

Not Present: Joshua Bass (at work)

No	A/D/I	Objective/Description	Who	When	Status
1	D	Discussed with Pedro about possible areas we could research further	Everyone	7/11/18	Completed

Figure 83- Project Week Meeting 3 front page

07/11/18

No	A/D/I	Objective/Description	Who	When	Status
2	I	- Calibration - fueling, mass of fuel + mass of air	Danny Stuart	7/11/18	Ongoing
3	A	Cylinder head - Induction - exhaust	Aidan Carl-Peter	7/11/18	Ongoing
4	A	Ricardo Lowe CFD Analysis	"	7/11/18	Ongoing
5	D	While waiting for 15 gearbox, mock up fake mounts in order to make progress in chassis fitment	Mike Callum	7/11/18	Ongoing
6		Meeting adjourned - next meeting scheduled 16/11/18			
7					

Figure 84 Project Week Meeting 3 back page

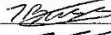
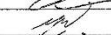
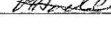
07/11/18

Engineering Management: Meeting Record Template

Meeting: 11:00 am - 12:00 pm Project Meeting 5 with Pedro Venue: 1st Floor IQ Date: 16/11/18

Attendees:

A/D/I (Action/Decision/Information)

Name	Signature
Stuart Harries	
Joshua Bass	
Aidan Chapman	
Iwan Hopkins	
Carl-Peter Englebrecht	
Callum-James Hill	
Danny Richardson	
Mike Anderson	

No	A/D/I	Objective/Description	Who	When	Status
1	I	Pedro has invoice for ECU - Waiting for John	Pedro Johnny	16/11/18	Ongoing

Figure 9-15 Meeting 5 front page 16/11/18

No	A/D/I	Objective/Description	Who	When	Status
2	D	Engine with 1.6 gearbox discussed	everyone	16/11/18	Completed
3	A	Engine on dyno trolley discussed	Mike Callum	16/11/18	Ongoing
4	A	Discussed what we plan/are doing Whilst waiting for parts	everyone	16/11/18	Ongoing
5	A	Talk to mark falls about a possible 3 rd engine	Stuart Danny	16/11/18	Completed
6	A	Find student who has the 1.5 head	Aidan Carl-Peter	16/11/18	Ongoing
7		Meeting adjourned Next meeting scheduled 19/11/18			

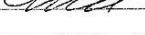
Figure 9-16 Meeting 5 back page 16/11/18

Engineering Management: Meeting Record Template

Meeting: 9am - 11:00am Project Meeting 6 with John Avery Venue: Cafe Date: 19/11/18

Attendees:

A/D/I (Action/Decision/Information)

Name	Signature
Mike Anderson	
Stuart Harries	
Danny Richardson	
Callum - James Hill	

No	A/D/I	Objective/Description	Who	When	Status
1	D	Can use 1.5 gearbox and will offer the 1.6 gearbox as an option.	John Avery	19/11/18	Completed

Figure 86- Meeting 6 front page 19/11/18

No	A/D/I	Objective/Description	Who	When	Status
2	D	Discussed driveshafts - mentioned that a thread will need to be put on the hub end	everyone	19/11/18	Completed
3	I	for ECU - with the lock and key	everyone	19/11/18	Completed
4	D	John authorised porting of the head - inlet and exhaust	everyone	19/11/18	Completed
5		Meeting adjourned next meeting scheduled ~~20/11/18~~ 19/11/18			
6					
7					

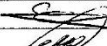
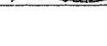
Figure 85- Meeting 6 back page 19/11/18

Engineering Management: Meeting Record Template

Meeting: 2pm Project Meeting with Pedro Venue: 101 IA Date: 19/11/18

Attendees:

A/D/I (Action/Decision/Information)

Name	Signature
Stuart Harnes	
Carl-Peter Englebrecht	
Aidan Chapman	
Iwan Hopkins	
Callum-James Hill	
Joshua Bass	
Danny Richardson	
Mike Anderson	

No	A/D/I	Objective/Description	Who	When	Status
1	I	Caught Pedro up with earlier meeting with John Avery	everyone	19/11/18	Completed

Figure 88- Meeting 7 front page 19/11/18

No	A/D/I	Objective/Description	Who	When	Status
2		Pedro will speak with Ford Bridgend about possible visit to site	Pedro	19/11/18	Ongoing
3		Dent Pedro group gantt chart	Iwan	19/11/18	Completed
4		Meeting adjourned next meeting scheduled 26/11/18			
5					
6					
7					

Figure 87 Meeting 7 back page 19/11/18

Engineering Management: Meeting Record Template

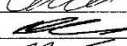
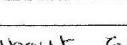
Meeting: 2pm Project meeting 9 with Pedro

Venue: 101 IQ

Date: 26/11/18

Attendees:

A/D/I (Action/Decision/Information)

Name	Signature
Stuart Harries	
Mike Anderson	
Denny Richardson	
Iwan Hopkins	
Callum-James Hill	
Adrian Chapman	
Joshua Bass	
Carl - Peter Englebracht	

No	A/D/I	Objective/Description	Who	When	Status
1	I	Pedro spoke with graham about ECU and parts	Pedro Graham	26/11/18	Completed

Figure 89- Meeting 8 front page

26/11/18

No	A/D/I	Objective/Description	Who	When	Status
2	D	Discussed the possibility of organising trips to Ford Bridgend	Pedro	26/11/18	Ongoing
3	A	Need to research plumbing for coolant and air	Iwan Josh	26/11/18	Ongoing
4	A	Dyno trolley fitment discussed	Mike Callum	26/11/18	Ongoing
5	A	Engine control board - investigate parts needed	Callum	26/11/18	Ongoing
6		Meeting adjourned next meeting scheduled 30/11/18			
7					

Figure 90- Meeting 8 back page

26/11/18

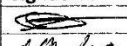
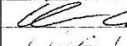
Engineering Management: Meeting Record Template

Meeting: Ham Project Meeting 9 with Pedro

Ver. Je: Outside 107 Date: 30/11/18
 I &

Attendees:

A D/I (Action/Decision/Information)

Name	Signature
Stuart Harris	
Mike Anderson	
Denny Richardson	
Iren Hopkins	
Callum - James Hill	
Carl-Peter Englebrecht	

No	A/D/I	Objective/Description	Who	When	Status
1	I	- John wants to fit 1.6 engine in chassis - Pedro doesn't see any advantage	John	30/11/18	Ongoing

Figure 92- Meeting 9 front page 30/11/18

No	A/D/I	Objective/Description	Who	When	Status
2		John will supply 1.6 Parts	John	30/11/18	Ongoing
3		John hasn't given Pedro ECU Invoice	John Pedro	30/11/18	Ongoing
4		Pedro says to email Mark Fall about Shipping head off 1.5 engine	Carl-Peter	30/11/18	Completed
5		IF John brings spare 1.6 engine in, Put 1.5 head on and mount in chassis	Mike Callum Stuart Denny	30/11/18	Ongoing
6		Meeting adjourned Next meeting scheduled 3/12/18			
7					

Figure 91- Meeting 9 back page 30/11/18

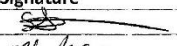
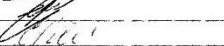
Engineering Management: Meeting Record Template

Meeting: 2nd Project Meeting 10 with Pedro

Ver. Je: 102 IG Date: 3/12/18

Attendees:

A, D/I (Action/Decision/Information)

Name	Signature
Stuart Haines	
Mike Jackson	
Iwan Hopkins	
Carl Peter Englebrecht	
Callum-James Hill	
Joshua Bass	
Aidan Chapman	

Not Present: Danny Richardson

No	A/D/I	Objective/Description	Who	When	Status
1		Pedro spoke to John Avery about rebuilding 2L engine	Pedro John	3/12/18	Ongoing

Figure 93- Meeting 10 front page 3/12/18

No	A/D/I	Objective/Description	Who	When	Status
2		Email Parts list to John Avery for the 1.5 engine	Penny	3/12/18	Completed
3		Scs has 1.5 on dyno for Development - possibly go as John Avery guest to see setup	everyone	3/12/18	Ongoing
4		Discussed Presentation with Pedro	everyone	3/12/18	Completed
5		Meeting adjourned next meeting scheduled 5/12/18			
6					
7					

Figure 94- Meeting 10 back page 3/12/18

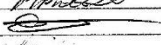
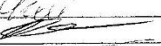
Engineering Management: Meeting Record Template

Meeting: 9am - 3pm Project Meeting 11

Venue: Room 3 Library Date: 5/12/18

Attendees:

A/D/I (Action/Decision/Information)

Name	Signature
Danny Richardson	
Mike Anderson	
Stuart Harries	
Callum-James Hill	
Brian Chapman	
Joshua Bries	
Carl-Peter Englebrecht	
Iwan Hopkins	

No	A/D/I	Objective/Description	Who	When	Status
1		Completed 1 st Project presentations	everyone		

Figure 95- Meeting 11 front page 5/12/18

No	A/D/I	Objective/Description	Who	When	Status
2		Completed several rehearsal before presenting the Powerpoint presentation at 3:30 p.m	everyone		
3		Meeting adjourned Next meeting scheduled for after exam period			
4					
5					
6					
7					

Figure 96- Meeting 11 back page 5/12/18

Engineering Management: Meeting Record Template

Meeting: 2nd Project Meeting with Pedro

Venue: 101 IQ Date: 28/1/19

Attendees:

A/D/I (Action/Decision/Information)

Name	Signature
Carl - Peter Englebrecht	
Pidan Chapman	
Mike Anderson	
Joshua Bass	
Callum - James Hill	
Danny Richardson	
Irony Hopkins	
Stuart Harries	

No	A/D/I	Objective/Description	Who	When	Status
1		Shown Pedro the completed Adaptor Shaft	Mike	28/1/19	Completed

Figure 97- Meeting 12 front page 28/01/19

No	A/D/I	Objective/Description	Who	When	Status
2		Caught Pedro up with trolley mounting	Mike Callum	28/1/19	Completed
3		Discussed missing parts Discussed Inlet & exhaust progress	Stuart Danny Pidán Carl-Peter	28/1/19	Ongoing
4		Discussed Zeno 150 mph crash	everyone	28/1/19	Completed
5		Meeting adjourned next meeting scheduled 4/2/19			
6					
7					

Figure 98- Meeting 12 back page 28/01/19