

Investigation into Internal Combustion Engine
Efficiency, Friction Reduction and Waste Heat
Recovery Systems

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Declaration of Authenticity

This thesis is the result of the author's original research. It has been composed by the author and has not been previously submitted for examination which has lead to the award of a degree.

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Date:

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Abstract

This paper will demonstrate how friction is caused in an internal combustion engine and a way of reducing the friction by offsetting the crankshaft from the cylinder axis. It will then look at the efficiency of the engine and how the energy is being wasted. A comprehensive study of a turbocharger was carried out to prove the validity of using a simple model. Then research into waste reduction and recovery systems will be carried out with the calculation of potential recovery efficiencies.

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Introduction

Internal combustion engines are facing stricter emissions tests to reduce harmful pollutants the engine gives out. The most efficient engines are only able to convert around 30 to 40% of the fuel power into useful work and the rest is wasted. A significant part of the heat wasted into the coolant system comes from friction within the engine. A modern trend to improve emissions drive cycle efficiency is to downsize the engine to reduce pumping losses. These downsized engines are then heavily boosted to bring the performance back up. The resulting high cylinder pressures increase piston assembly friction.

Internal combustion engines use roughly 2/3 of the oil produced in the world and around 30% is wasted through down the exhaust stream. There is a huge amount of energy to be recovered however current technologies are very inefficient or expensive and complex systems. With more electrification of cars the engine is not needed to be so powerful but rather more efficient. This has led to the implementation of over expansion cycles that previously were not beneficial on internal combustion engine only vehicles.

This report will demonstrate how a friction model can be created to predict the overall friction of an engine and see which parts are the dominant factors. The friction calculated will then be compared to real data to see the validity of the model. A way of reducing the friction of the piston skirt by offsetting the crankshaft seeing what effects this has on other aspects of the engine's characteristics. By comparing the effect of offset at full and part load it can be seen if the theoretical optimum offset is the same for both or if the offset should be optimised depending on how the car will be driven. Using Test data from a Ford 1L Ecoboost engine thermal efficiency will be calculated along with what happens to the fuel energy that doesn't get used for mechanical work. A model of the turbocharger will be generated to see the power it can extract from the exhaust gas. The model will include kinetic energy which many text books is said to be negligible therefore can be left out. The author wanted to see how the kinetic energy effected the transfer efficiency between the turbine and compressor to see if the kinetic energy really was negligible when comparing the two. Finally waste heat recovery systems will be researched to see how much of the wasted heat can realistically be recovered.

1 Friction Model

Frictional losses are a significant fraction of the power produced by an internal combustion engine, in the region of 10-15% in normal road driving conditions. Friction effects the maximum brake torque and the brake specific fuel consumption directly and can be the difference between a good engine design and an average engine design (Heywood, 1988). The friction within an engine can be categorised into three main parts. Mechanical friction, the friction of the moving parts within the engine such as the valvetrain, piston assembly, and crankshaft. Accessory work, the work required to power the accessories such as the oil pump, water pump and alternator. Pumping work, the work done during the intake and exhaust strokes (Ferguson, 2001).

A friction model will be created consisting of piston assembly friction, camshaft friction, crankshaft bearing friction, water and oil pump friction and pumping losses. Using cylinder pressure and engine geometry data from a ford fox 1.0L Ecoboost engine. This model will then be compared to measured data to see how accurate the friction model is.

The friction will be calculated in terms of Friction Mean Effective Pressure (FMEP), this gives a measurement of friction per unit volume therefore it can be used to compare to any capacity engine.

$$FMEP \times Engine\ Displacement = Work\ Done\ in\ 2\ Revolutions$$

$$FMEP = \frac{Work\ Done\ in\ 2\ Revolutions}{Engine\ Displacement}$$

Or

$$FMEP = \frac{Work\ Done\ in\ 2\ Revolutions\ per\ Cylinder}{Cylinder\ Displacement}$$

Equation 1.1 Friction Mean Effective Pressure

The answer will then be divided by 10^5 to convert from Pascal's to bar.

Specifications of the engine can be found in the appendix 8.5.

1.1 Piston Skirt Friction

A large part of the friction within an engine comes from the piston assembly and in particular the piston skirt interaction with the cylinder wall. The piston skirt guides the piston motion within the cylinder (Manning, 2012).

Figure 1.1 shows how much of the total FMEP the piston assembly is accountable for. At full load it is responsible for nearly half of the mechanical friction.

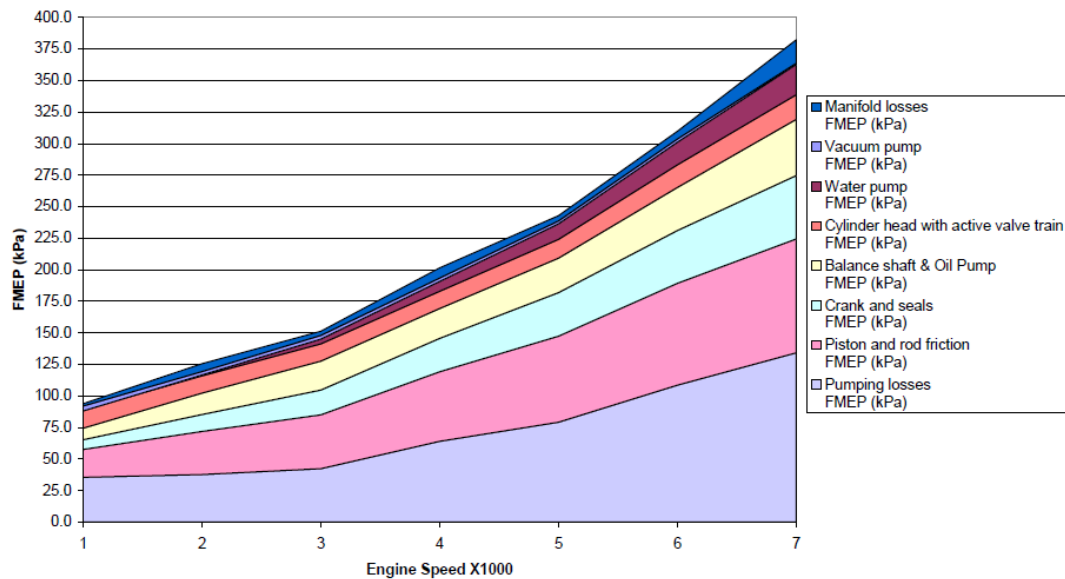


Figure 1.1 Total Engine Friction (Mackay, 2009)

As the gas force from combustion pushes down on the piston, the connecting rod provides a reacting force and pushes back. Due to the connecting rod being at an angle this forces the piston into the cylinder wall and the cylinder wall pushes back creating a side force. This force is the cause of friction. The friction force acts at right angles to the side force.

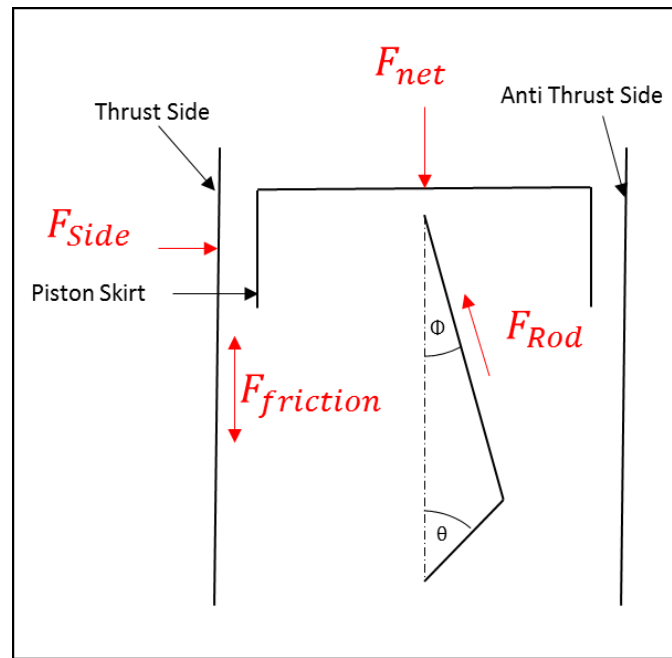


Figure 1.2 Forces

To calculate the side force the connecting rod needs to be calculated.

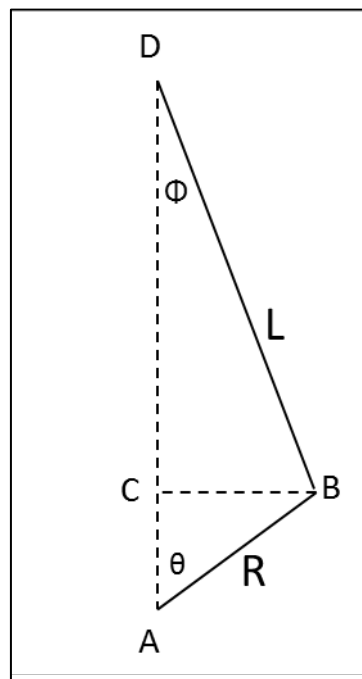


Figure 1.3 Connecting Rod Angle Diagram

From Figure 1.3 it can be seen that triangles ABC and BCD share a common side BC .

For triangle ABC

$$BC = R.\sin\theta$$

For Triangle BCD

$$BC = L.\sin\phi$$

Therefore

$$L.\sin\phi = R.\sin\theta$$

This can be rearranged for ϕ

$$\phi = \sin^{-1} \left[\frac{R}{L} \cdot \sin\theta \right]$$

Equation 1.2 Connecting Rod Angle

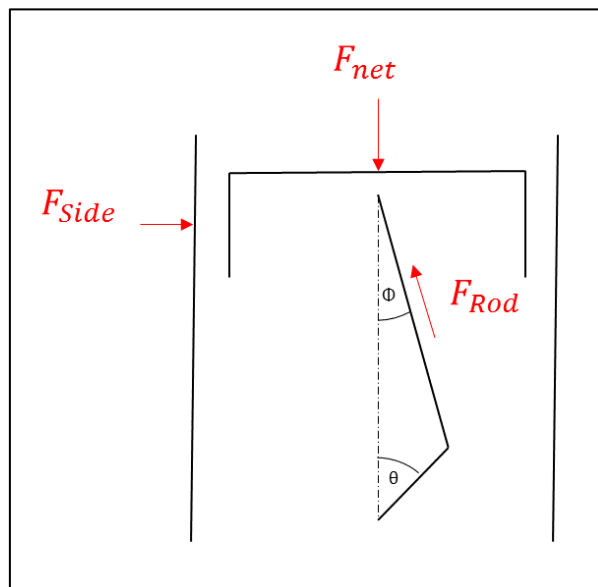


Figure 1.4 Resolving Forces

From Figure 1.4 resolving the force vertically gives

$$F_{net} = F_{rod}.\cos\phi.$$

Resolving horizontally gives

$$F_{side} = F_{rod}.\sin\phi.$$

Dividing the second equation by the first gives

$$\frac{F_{side}}{F_{net}} = \frac{F_{rod}.\sin\phi}{F_{rod}.\cos\phi}$$

The F_{rod} 's cancel leaving

$$\frac{F_{side}}{F_{net}} = \frac{\sin\phi}{\cos\phi}$$

As

$$\frac{\sin}{\cos} = \tan$$

Then

$$\frac{F_{side}}{F_{net}} = \tan\phi$$

Rearranging gives

$$F_{side} = F_{net} \cdot \tan\phi$$

Equation 1.3 Side Force

It can also be seen in Figure 1.4 that

$$F_{rod} = \frac{F_{net}}{\cos\phi}$$

Equation 1.4 Connecting Rod Force

The net force acting on the piston is

$$\text{Net Force} = \text{Gas Force} - \text{Inertia Force}$$

Equation 1.5 Net Force

Where

$$\text{Gas Force} = \text{Cylinder Pressure} \times \text{Cylinder Bore Area}$$

Equation 1.6 Gas Force

And

$$\text{Inertia Force} = \text{Reciprocating Mass} \times \text{Acceleration}$$

Equation 1.7 Piston Assembly Inertia Force

The reciprocating mass is the piston, gudgeon pin and a third of the connecting rod mass (Bosch, 2004).

The acceleration was calculated by first of all calculating the distance the piston travels per crankshaft degree then taking the change in distance per degree to give the velocity, then taking the change in velocity per degree to give acceleration. This gives acceleration per degree per degree and to convert to seconds it is multiplied by $6. \text{RPM}^2$.

The distance the piston has travelled down can be calculated by finding the vertical components of both the connecting rod and crank throw when they are at an angle.

When the piston is at TDC the connecting rod and crank pin are in a vertical line therefore the vertical component = $R + L$ where R is the crank throw and L is the connecting rod length Figure 1.5.

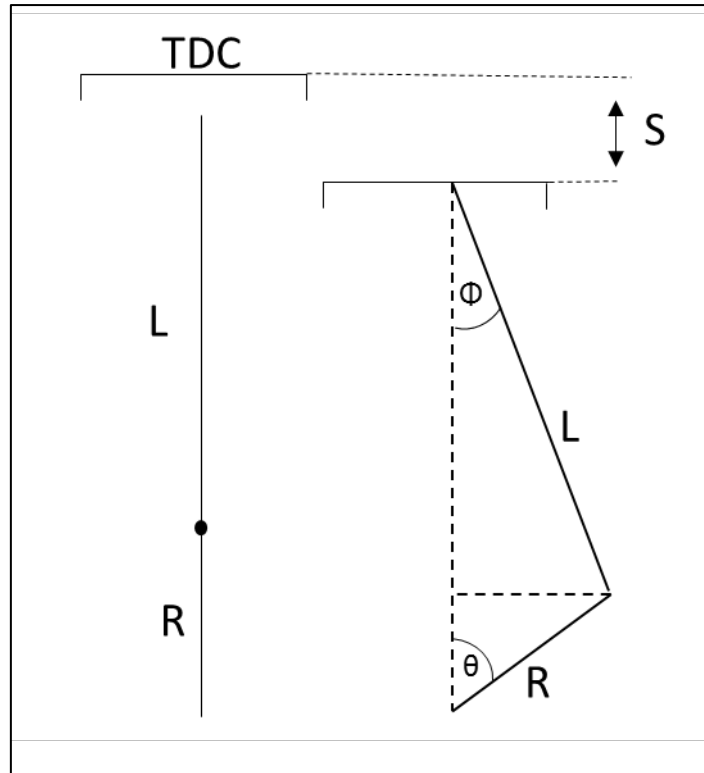


Figure 1.5 Distance Piston Has Moved

From Figure 1.5 it can be seen that when the crankshaft has rotated the vertical component of the crank throw is $R \cdot \cos\theta$ and the vertical component of the connecting rod is $L \cdot \cos\phi$

Therefore the new height

$$R \cdot \cos\theta + L \cdot \cos\phi.$$

Therefore

$$S = R + L - (R \cdot \cos\theta + L \cdot \cos\phi).$$

Equation 1.8 Distance Piston Has Travelled

Once the side force is calculated it is multiplied by the friction coefficient to give the friction force that the engine must overcome. The friction coefficient use is 0.1 however there will be further discussion of this later in the report.

$$\text{Friction Force} = \mu \times \text{Side Force}$$

Equation 1.9 Piston Skirt Friction Force

The work done by the engine against the friction is

$$\text{Work Done} = \text{Friction Force} \times \text{Distance Piston has Travelled.}$$

Equation 1.10 Work Done by Piston against Friction

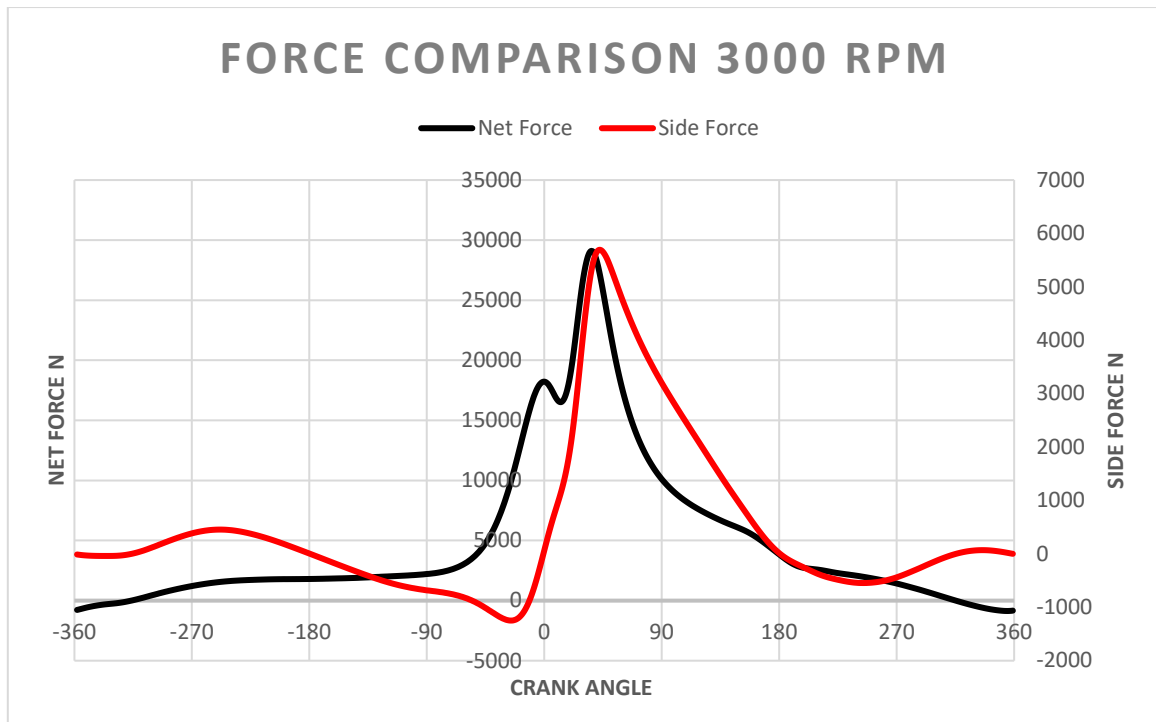


Figure 1.6 Force Comparison

It can be seen in Figure 1.6 that maximum friction force comes slightly after maximum net down force. This is because the connecting rod angle has increased and this has a greater effect on the side force than the reduction in net down force acting on the piston.

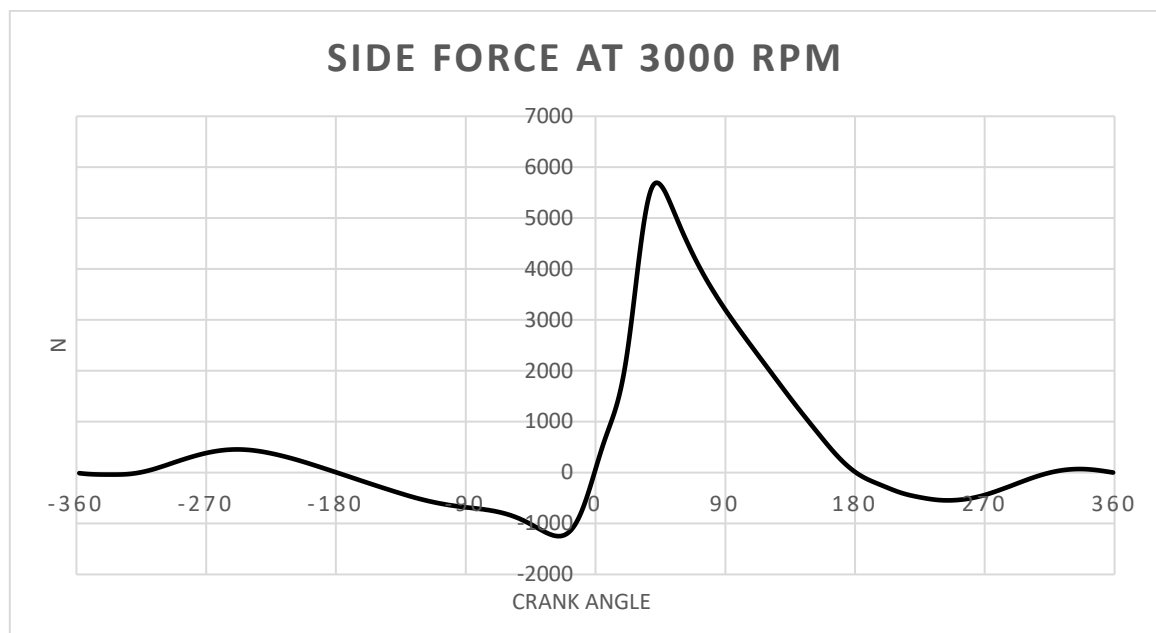


Figure 1.7 Side Force

Peak piston skirt friction occurs on the power stroke. This is due to the net force being much greater than that of the other strokes.

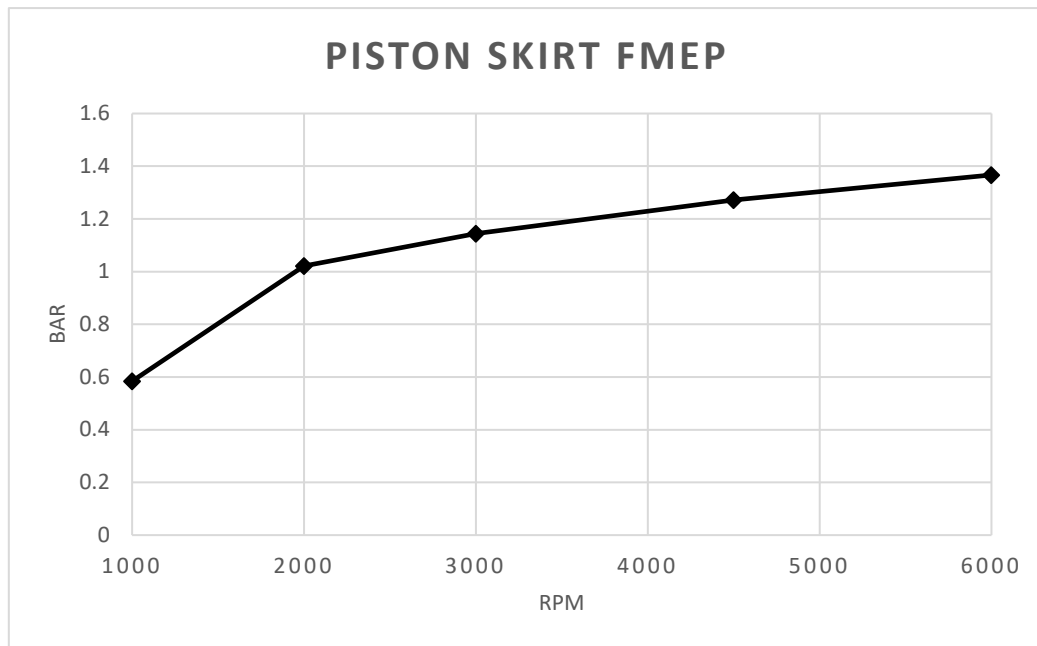


Figure 1.8 Piston Skirt Mean Effective Pressure

Through the RPM range the skirt friction increases, initially this is because the gas pressure has increased due to better combustion. As the engine reaches peak RPM although the gas pressure has reduced slightly due to choking the inertia forces have increased more than the gas force has reduced giving an increase in friction.

1.2 Compression Ring

The top piston ring also known as the compression ring seals the combustion chamber to maximise pressure build up to maximise the gas force. However, this causes friction as the gas pressure gets in behind the ring to create a radial force pushing the ring into the cylinder liner to create the seal Figure 1.9. The radial force is gas pressure multiplied by the inner ring area. Throughout the RPM the ring friction follows the trend of the peak pressure. The piston ring also has another function and that to remove heat from the piston to the cylinder wall. Reducing the piston ring thickness would reduce the friction by reducing the area and reducing the radial force, however the ring needs to be thick enough to enable adequate heat transfer away from the piston.

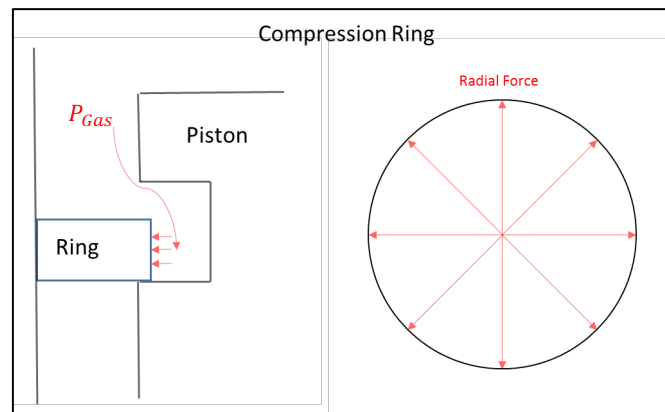


Figure 1.9 Compression Ring Forces

$$\text{Radial Force} = \text{Cylinder Pressure} \times \text{Compression Ring Inner Area}$$

Equation 1.11 Compression Ring Radial Force

$$\text{Radial Friction Force} = \mu \times \text{Radial Force}$$

Equation 1.12 Compression Ring Radial Friction Force

$$\text{Work Done Against Friction} = \text{Radial Friction Force} \times \text{Distance Piston Has Moved}$$

Equation 1.13 Compression Ring Work Done Against Friction

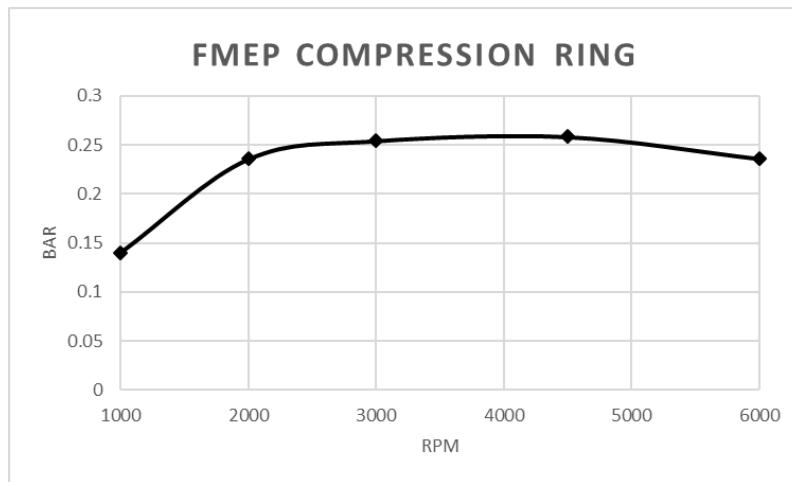


Figure 1.10 Compression Ring Friction Mean Effective Pressure

At low RPM the volumetric efficiency is low therefore combustion is not very good then the peak pressure is low therefore the ring friction is low. As the RPM increases so does the volumetric efficiency and the peak gas pressure therefore the ring friction has increased. At high RPM the engine begins to choke and volumetric efficiency is reduced, peak gas force has reduced and so has the friction.

1.3 Camshaft Friction

As the camshaft contacts the valve bucket to push the valve down it is resisted by two forces, the spring force and the inertia force of the valve assembly. The spring force is dependent on the spring rate, the initial compression when the spring is installed and the valve lift. The inertia force is made up from the acceleration of the moving valve assembly. As the camshaft slides across the top of the bucket there is then a resisting force acting perpendicular to the contact force. By multiplying this force by the friction coefficient to give a friction force, friction torque can then be calculated.

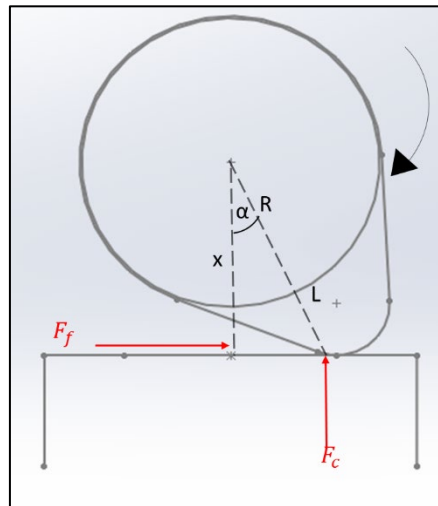


Figure 1.11 Camshaft Friction

$$\text{Contact Force} = \text{Spring Force} + \text{Inertia Force}$$

Equation 1.14 Camshaft Contact Force

$$\text{Spring Force} = \text{Spring Rate} \times (\text{Initial Compression} + \text{Lift})$$

Equation 1.15 Valve Spring Force

$$\text{Inertia Force} = \text{Mass} \times \text{Acceleration}$$

Equation 1.16 Valve assembly Inertia Force

$$\text{Friction Force} = \mu \times \text{Contact Force}$$

Equation 1.17 Camshaft Friction Force

$$\text{Resisting Torque} = \text{Friction Force} \times (\text{Base Circle} + \text{Valve Lift})$$

Equation 1.18 Camshaft Resisting Torque

The valve lift is used as this is the vertical distance from the centre of the camshaft to the bucket therefore perpendicular to the friction force. Calculating the valve lift from camshaft lift can be found in appendix.

$$\text{Work Done per Degree} = \text{Resisting Torque} \times \left(\frac{2\pi}{360} \right)$$

Equation 1.19 Camshaft Work Done per Degree

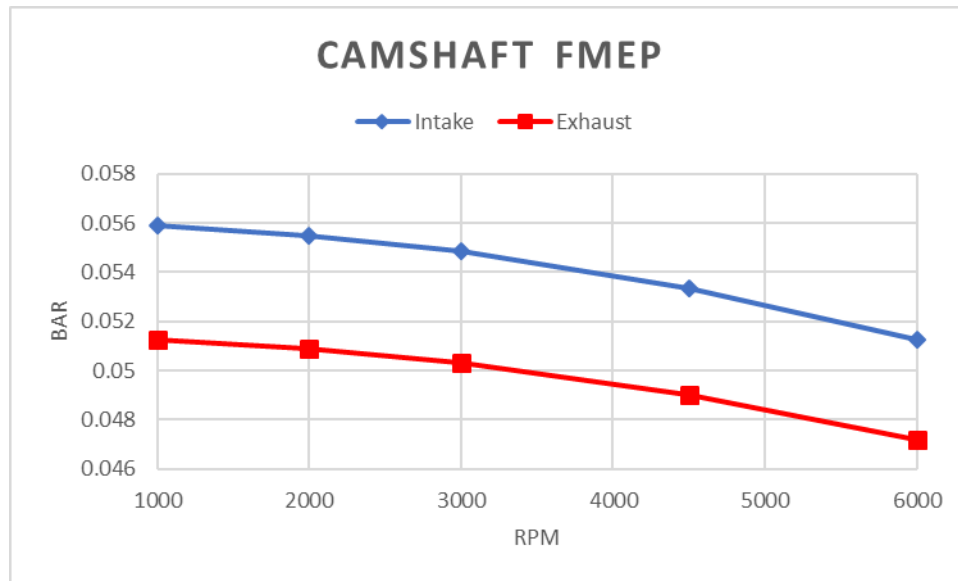


Figure 1.12 Friction Mean Effective Pressure

The intake camshaft friction is higher than the exhaust camshaft friction because the intake valve spring rate is slightly higher than the exhaust valve spring, 14 N/mm and 13 N/mm respectively. The intake camshaft lift is also higher than the exhaust camshaft. Both of these factors means that the spring force will be increased, which increase the contact force and therefore the friction force. Because the intake camshaft has a higher lift this means that the torque arm is longer and therefore the resisting torque has increased.

The camshaft FMEP reduces as the engine speed increases. As the RPM increased the Inertia force increases but the spring force remains the same. The average inertia remains constant over the RPM range because the inertia increases the same amount in both positive and negative directions, however it is the negative inertia force that reduces the friction. As can be seen in Figure 1.13 around the acceleration is negative when the valve lift is above roughly half of maximum lift. At higher lift the torque arm is bigger therefore the resisting torque is increased. Because the inertia force will be negative in these regions it means that the contact force will be reduced. This decrease in contact force outweighs the increase in contact force that is caused by the greater positive inertia force at the beginning and end of lift stages due to the torque arm being shorter.

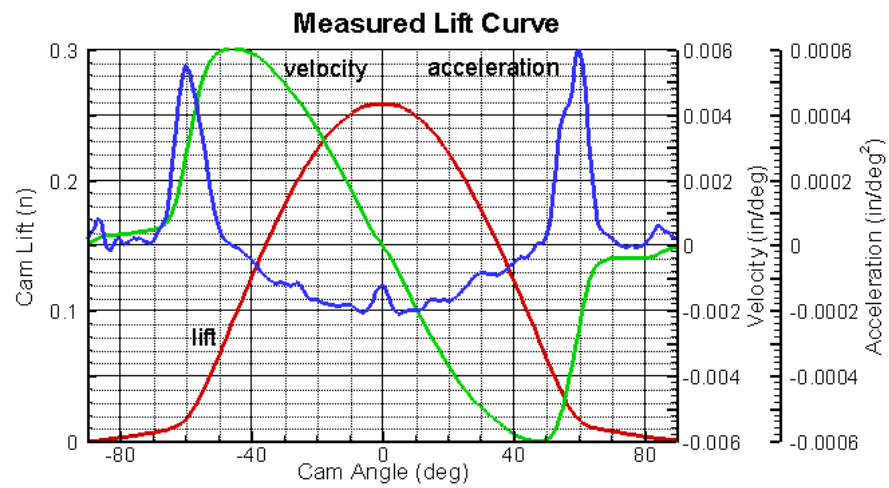


Figure 1.13 Valve Dynamics

1.4 Crankshaft Bearings

The crankshaft bearings are hydrodynamically lubricated meaning an oil film is built up between the bearing and the journal. Oil sticks to the crankshaft and rotates with the crankshaft at the speed of the engine. Oil also sticks to the bearing journal which is stationary. This creates a velocity gradient within the oil and the velocity reduces as it gets further away from the rotating crankshaft. It can be imagined that the oil is made up of lots of layers that are rotating at different speeds to each other. This causes shearing within the oil which is how the friction is created. The resistance to shearing within the oil is its dynamic viscosity. If the velocity gradient is increased by either a higher velocity or a thinner oil film the friction will increase because the velocity difference between the layers of oil will have increased therefore the shearing force would increase. Therefore, the crankshaft bearing friction increases with RPM.

$$\text{Resisting Force} = \text{Shear Stress} \times \text{Surface Area}$$

Equation 1.20 Crankshaft Bearing Resisting Force

$$\text{Shear Stress} = \mu \times \frac{v}{d}$$

Equation 1.21 Shear Stress in Oil Film

Where μ is the dynamic oil viscosity, v is the tangential velocity and d is the oil film thickness.

$$v = r \cdot \omega$$

Equation 1.22 Tangential Velocity

Where r is the bearing radius and ω is the angular velocity.

$$\omega = \text{RPM} \times \frac{2 \cdot \pi}{360}$$

Equation 1.23 Crankshaft Bearing Angular Velocity

To determine the oil film thickness the bearing clearance was used (Clevite, 2005) suggests that a safe journal bearing clearance for passenger vehicles is 0.001 inch for every 2 inch diameter bearing therefore

$$\frac{0.001}{2} = 0.0005$$

By multiplying the bearing diameter by this number gives the clearance.

$$\text{Surface Area} = \text{Bearing Circumference} \times \text{Bearing Length}$$

The bearing circumference and length were unable to be measured. However (Ferguson, 2001) states that typical crankshaft main bearing for automotive spark ignition engines are typically sized to be roughly 65% of the cylinder bore and bearing lengths are 35-40% of the cylinder bore.

Therefor

$$\text{Work Done in 2 revolutions} = \text{Resisting Force} \times 4\pi \times \frac{D}{2}$$

Equation 1.24 Crankshaft Bearing Work Done

Where D is the bearing diameter.

The dynamic oil viscosity varies with oil grade and temperature. The grade and temperature used was a SAE 10w-40 at 100°C.

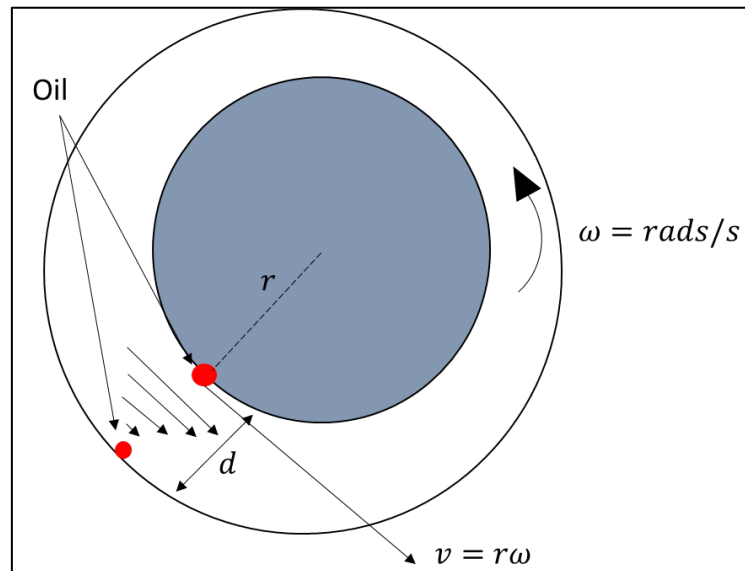


Figure 1.14 Crankshaft Bearing Friction

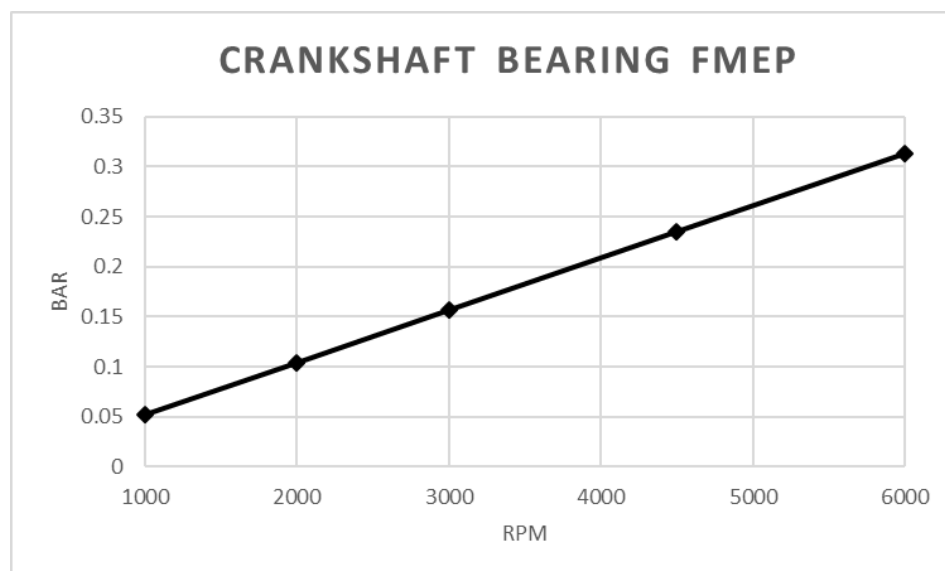


Figure 1.15 Crankshaft Bearing Friction Mean Effective Pressure

From the Equation 1.21 the only change is the velocity is due to increasing RPM and this is the cause for the rise in friction as it causes a higher velocity difference between the layers of oil.

1.5 Oil and Water Pump

In order to cool and lubricate the engine oil and water must be pumped around the engine. The oil and water pumps are driven from the engine therefore they are taking away useable power away from the engine. The only data available for the pumps were the pressures therefore flow rates where estimated.

The work done per second by the pump is

$$\text{Work Done} = \text{Pressure} \times \text{Change in Volume}$$

Equation 1.25 Work Done by Oil and Water Pumps

The change in volume is the flow rate of the pump in m^3 per second and the pressure is the desired pressure of the fluid.

As the work done is calculated per second, the number of revolutions per second needs to be calculated.

$$\text{Revolutions per Second} = \frac{RPM}{60}$$

$$\text{Work Done in 2 Revolutions} = 2 \times \left(\frac{\text{Work Done per Second}}{\text{Revolutions per Second}} \right)$$

Assuming the pump has a mechanical efficiency of 75% the mechanical work done becomes

$$\text{Mechanical Work Done in 2 Revolutions} = \frac{\text{Work Done in 2 Revolutions}}{0.75}$$

$$FMEP = \frac{\text{Mechanical Work Done in 2 Revolutions}}{\text{Engine Displacement}}$$

Equation 1.26 Oil and Water Pump FMEP

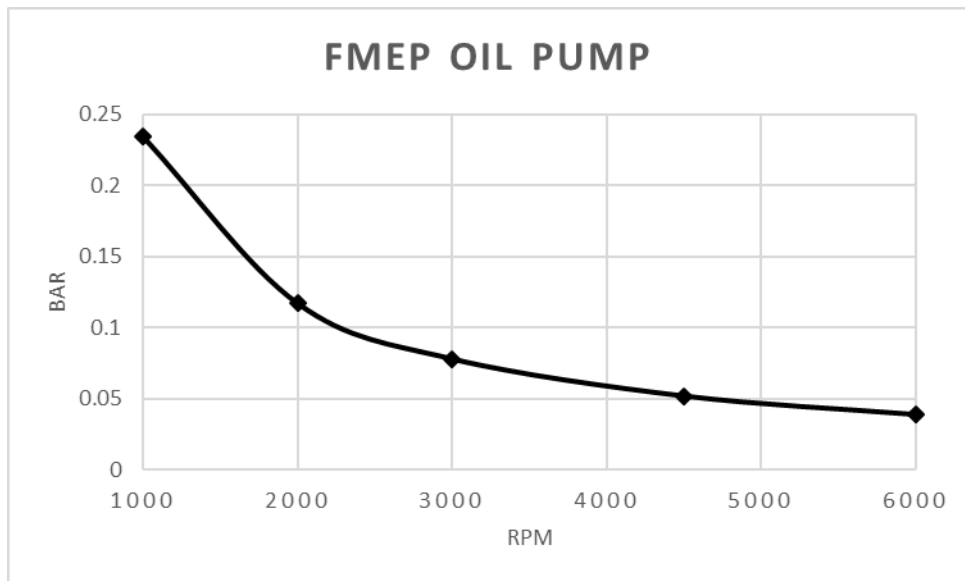


Figure 1.16 Oil Pump Friction Mean Effective Pressure

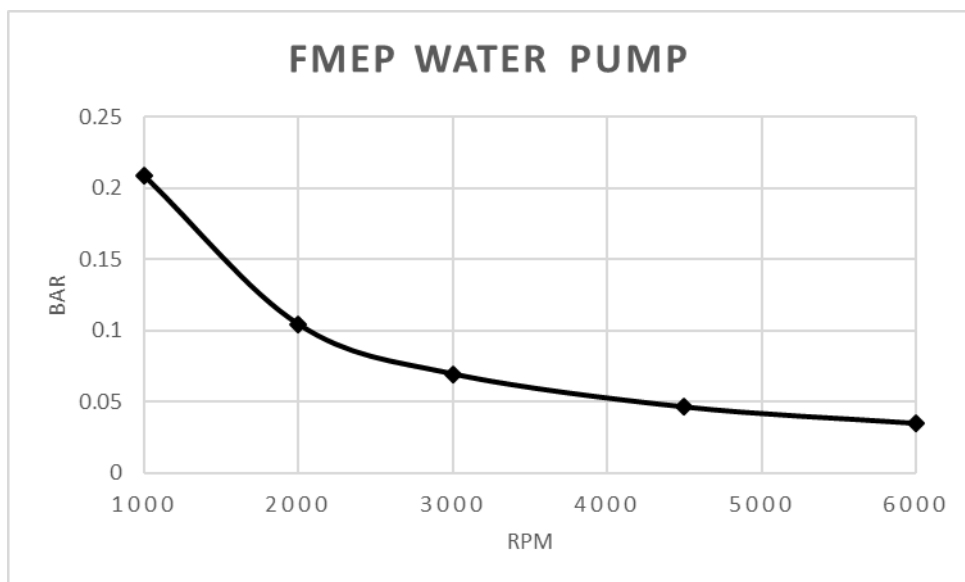


Figure 1.17 Water Pump Friction Mean Effective Pressure

1.6 Pumping Mean Effective Pressure

The pumping mean effective pressure is the work done by the engine to bring in fresh charge into the cylinder and also to expel the exhaust gas. During the intake stroke the piston pulls air into the cylinder but there are restrictions in the intake path such as the throttle, air filter and intake manifold. The largest restriction can be the throttle. When the engine is running at part load the throttle is partially closed meaning the area the air can pass through is greatly reduced. This makes it difficult for the piston to pull air into the cylinder and increase the work done during the intake stroke. The exhaust system also has restrictions such as the exhaust manifold, turbocharger (if the engine is boosted), catalytic converter, silencer and tail pipe (Ferguson, 2001). As the RPM increases the exhaust gas need to be pushed out of the cylinder at a faster rate and this causes a build-up of pressure that the piston has to work against because the gas cannot pass through the valve quick enough.

$$\text{Pumping Work} = \text{Work done Exhaust} - \text{Work Done Intake}$$

Equation 1.27 Pumping Work

The work done on the exhaust stroke will be negative because the change in volume is negative as it is getting smaller. The absolute value needs to be used in Equation 1.27. The intake work is deducted from the exhaust work because the more of a restriction in the intake system, making it more difficult to bring in air, the pressure will drop which will result in less work done on the intake stroke therefor the less subtracted from the exhaust work done means the higher the pumping work.

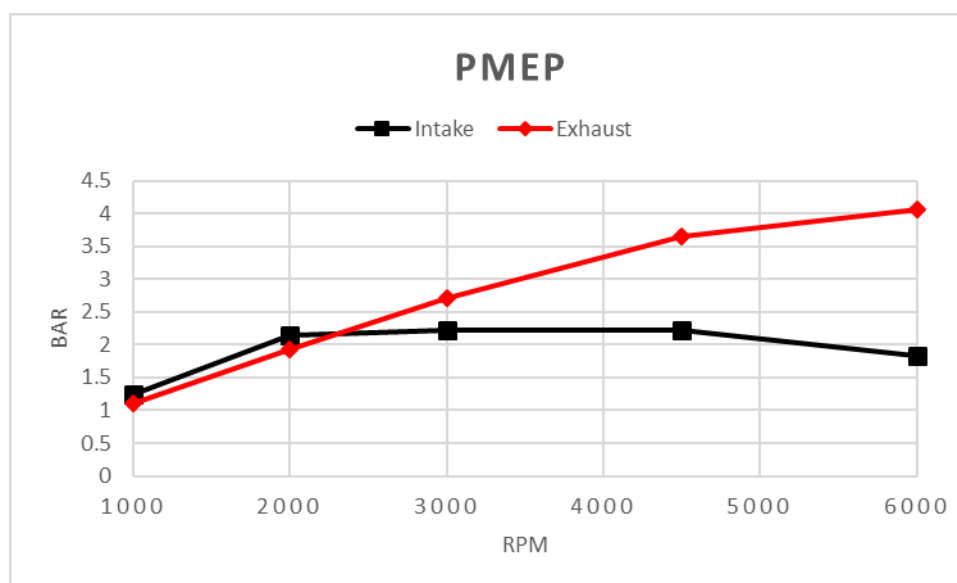


Figure 1.18 Pumping Mean Effective Pressure

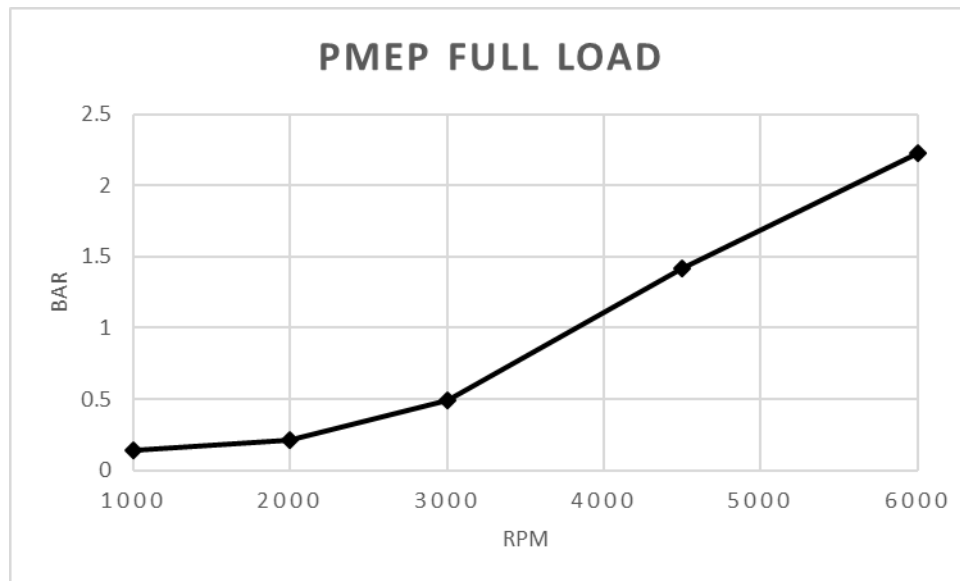


Figure 1.19 Total Pumping Mean Effective Pressure

The PMEP at low RPM is low because the cylinder pressure will not be as high and the piston pushes the exhaust gases out slower therefore a restriction isn't caused. As the RPM increase the cylinder pressure increases so the piston does more work to push the exhaust gas out and because it is pushing it out faster the restriction of the port size means the pressure will increase again resulting in more work done by the piston.

1.7 Total Friction

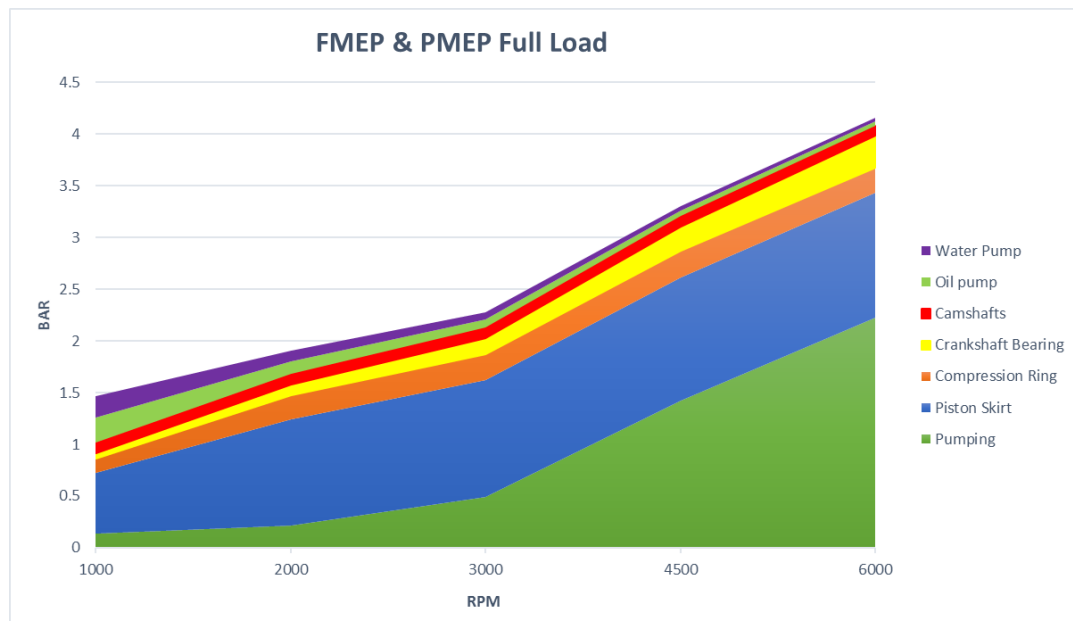


Figure 1.20 Total Friction at Full Load

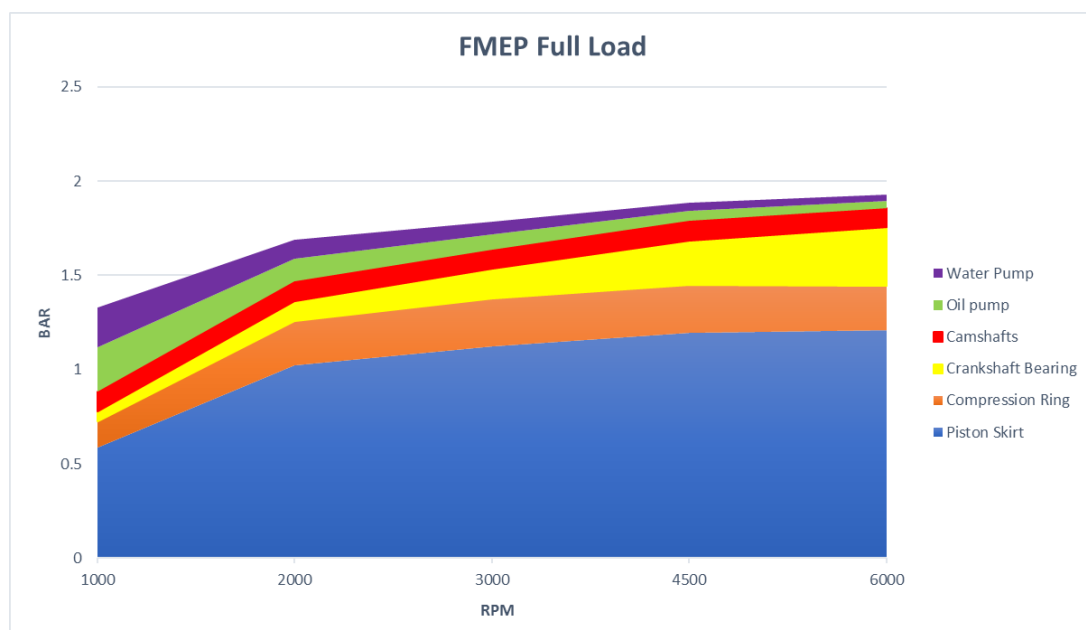


Figure 1.21 Friction Mean Effective Pressure at Full Load

Figure 1.20 shows that at high RPM the PMEP becomes a dominant part of the total friction. As seen in Figure 1.18 the Exhaust PMEP rises with RPM. A reason for this high Exhaust PMEP could be that the engine is turbocharged therefore there is a restriction in the exhaust system. Figure 1.21 shows that the piston skirt is the major cause of mechanical friction within the engine

1.8 Friction model validation

Additional test data was obtained which included the total measured FMEP of the engine and cylinder pressures. The cylinder pressure data was put into the model of the total calculated FMEP could then be compared to the measured data.

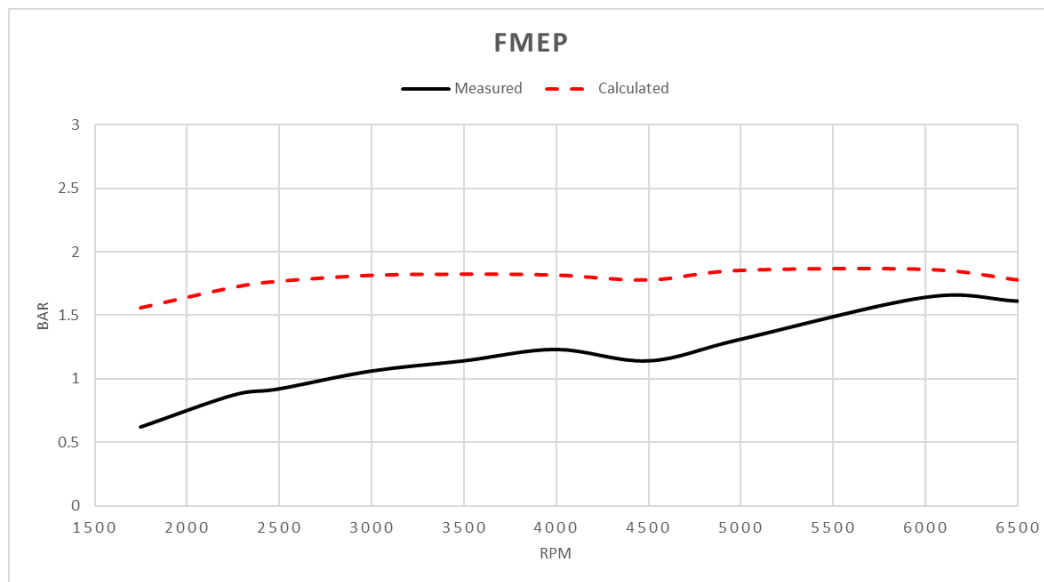


Figure 1.22 Friction Mean Effective Pressure Comparison

Table 1 Calculated FMEP Error

RPM	1750	2250	2500	3000	3500	4000	4500	5000	6000	6500
Error	60%	49%	48%	42%	37%	32%	36%	29%	12%	9%

It can be seen from Figure 1.22 that at low load the model over predicts by a large amount the FMEP compared to the real measured data. This is most likely due to the friction coefficient used for the calculations. A friction coefficient of 0.1 across all RPM ranges for piston skirt, piston rings and the camshafts. In reality the coefficient will be higher or lower at certain RPM's and also throughout the cycle. (Calabretta, 2010) states how the friction coefficient is complex variable as it is dependent on contact force, sliding speed, oil film thickness, temperature, viscosity and surface texture.

A common way the friction coefficient is shown is through a Stribeck curve Figure 1.23 that shows how the lubrication regime changes depending on load, speed and viscosity also known as the hersey number. However obtaining what the value of hersey number corresponds the different lubrication regimes that will then be able to give a suitable friction coefficient has proven difficult.

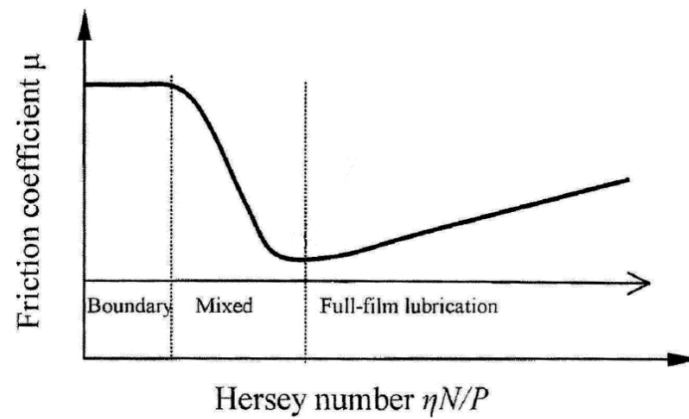


Figure 1.23 Stribeck Curve

FMEP Percentage of Total								
	Piston Skirt	Compression Ring	Crankshaft Bearings	Intake Camshaft	Exhaust Camshaft	Oil Pump	water Pump	Piston Assembly
1250	50%	14%	5%	4%	4%	10%	13%	64%
1500	55%	15%	6%	4%	4%	7%	10%	70%
1750	56%	15%	6%	4%	3%	8%	8%	71%
2000	58%	15%	6%	3%	3%	7%	6%	74%
2250	58%	15%	7%	3%	3%	7%	5%	74%
2500	59%	15%	8%	3%	3%	7%	5%	75%
3000	59%	15%	9%	3%	3%	7%	4%	75%
3500	59%	15%	10%	3%	3%	6%	3%	75%
4000	59%	15%	12%	3%	3%	5%	3%	74%
4500	59%	14%	14%	3%	3%	5%	3%	73%
5000	60%	14%	15%	3%	3%	4%	2%	74%
5500	60%	13%	16%	3%	3%	4%	2%	73%
6000	59%	13%	17%	3%	3%	3%	2%	72%
6500	58%	12%	20%	3%	3%	3%	2%	70%

Figure 1.24 Calculated Friction Percentages of Total

Mahle Measured Data	
RPM	Piston and Connecting Rod
1000	42%
2000	43%
3000	42%
4000	44%
5000	43%
6000	42%
6800	40%

Figure 1.25 Measured Piston and Connecting Rod Percentages of Total

From Figure 1.24 it can be seen that the piston assembly friction is a very large proportion of the total friction as high as 75%. Comparing this to a test carried out by MAHLE Powertrain Figure 1.25 this is much higher. This gives a good indication that it is the Piston Skirt part of the friction model that is over predicting.

The author was advised not to spend too long trying to work on finding a way of calculating the correct coefficient of friction and instead move on with the project.

2 Offset Crankshaft

The effect of offsetting the crankshaft to reduce friction will be analysed in this chapter. Part Load and full load data will be analysed to see how the offset effects each of these.

2.1 Connecting Rod Angle

Having seen that the two components that effect the side force Equation 1.3 and therefore the friction force of the piston skirt are the gas force and the connecting rod angle. A reduction in either of these will reduce the side force. However, a reduction in gas force is not desirable as this will reduce the engine torque and reduce performance. A reduction of connecting rod angle during high gas force events can greatly reduce friction.

A way to reduce the connecting rod angle during the power stroke is to offset the crankshaft towards the thrust side of the cylinder.

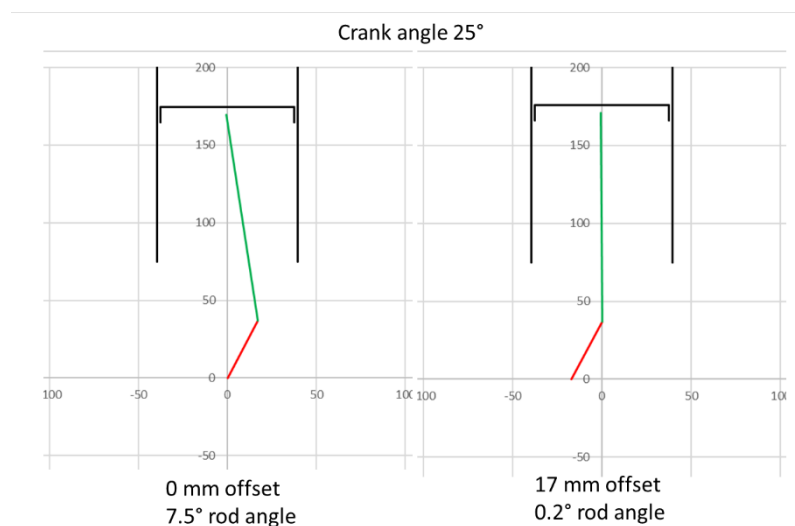


Figure 2.1 Crankshaft Offset Visualisation

As can be seen in Figure 2.1 by offsetting the crankshaft 17mm, the rod angle at around peak pressure can be reduced virtually to zero.

The net force at 25° crank angle is 22849 N at 3000 RPM full load.

With no offset the connecting rod angle is 7.5° therefore,

$$\text{Side Force} = 22849 \times \tan(7.5) = 3008.13\text{N}.$$

With an offset of 17mm the connecting rod angle is reduced to 0.2° the side force is then reduced to $79.76N$ a 97% decrease in side force. This shows the effectiveness of how offsetting the crankshaft from the cylinder axis can reduce friction.

With the offset crankshaft the new connecting rod angle needs to be calculated.

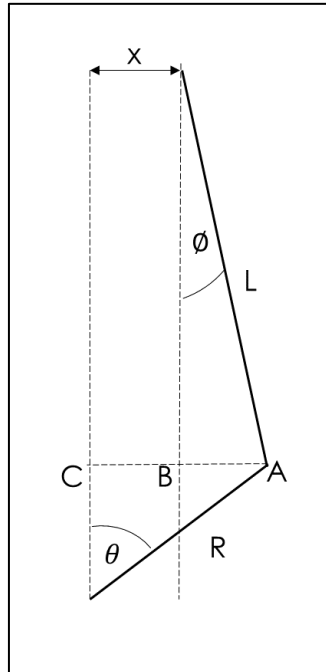


Figure 2.2 Connecting Rod Angle with Offset Crankshaft

From Figure 2.2 it can be seen that

$$R \cdot \sin \theta = AC$$

And

$$L \cdot \sin \phi = AB.$$

Side AC is longer than side AB by length $BC = X$, the distance of the offset.

Therefore

$$L \cdot \sin \phi = R \cdot \sin \theta - X.$$

This can then be rearranged to give ϕ , the new connecting rod angle.

$$\phi = \sin^{-1} \left[\frac{R}{L} \cdot \sin \theta - \frac{X}{L} \right]$$

Equation 2.1 Connecting Rod Angle with Crankshaft Offset

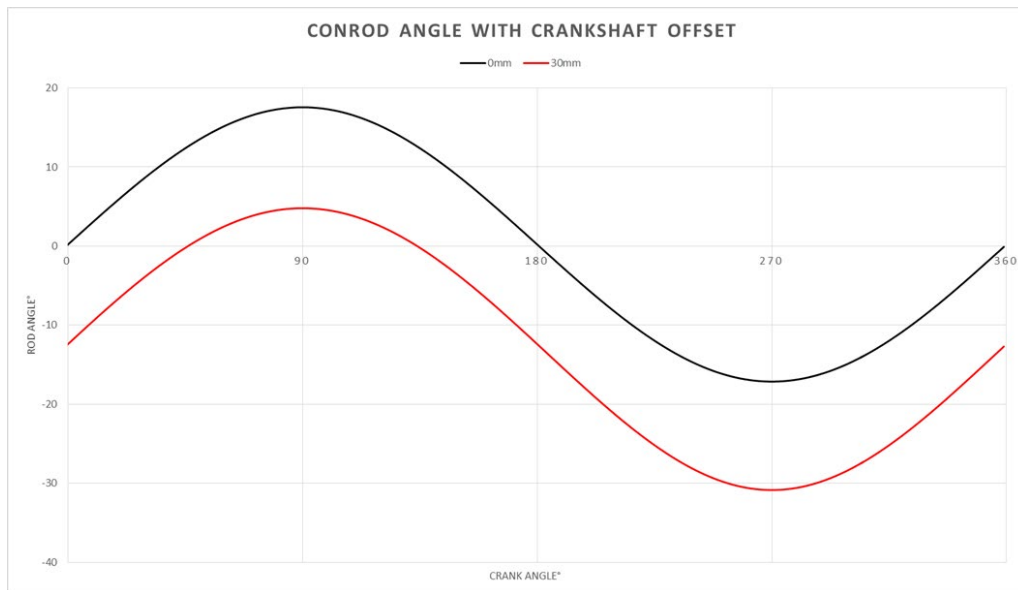


Figure 2.3 Connecting Rod Angle Comparison

The new connecting rod angle was plotted with an offset of 30mm compared with 0mm offset Figure 2.3. It can be seen that although the connecting rod angle is greatly reduced on the power stroke (0° - 180° Crank Angle), the connecting rod angle has increased substantially on the exhaust stroke (180° - 360° Crank Angle). When the gas force is much greater than the inertia force for example at full load and mid RPM then the increased connecting rod angle on the exhaust and compression strokes isn't too much of a problem as the reduction in the power stroke is so great, however when the inertia force has increased compared to the gas force for example at high RPM where the inertia force increases exponentially and also at part load where the gas force is reduced then this increase in connecting rod angle does start to make the friction rise again.

2.2 Effect on Side Force

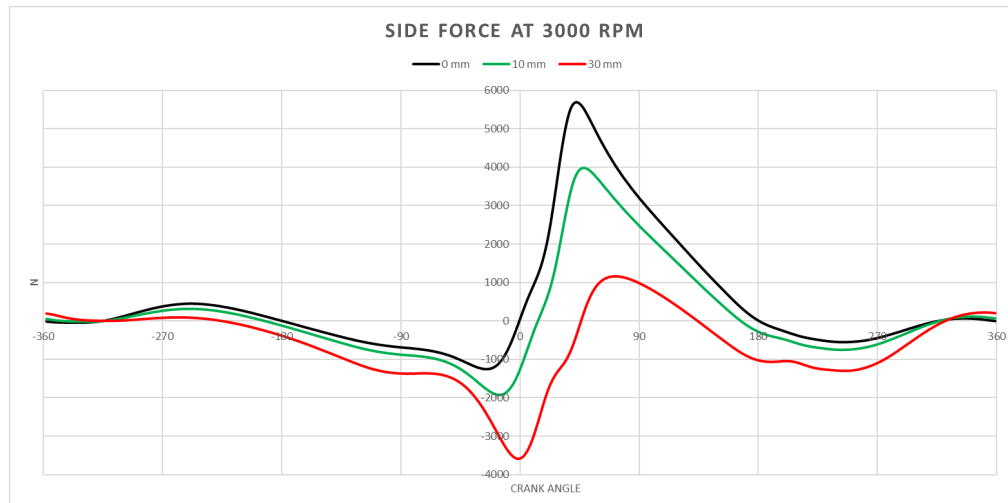


Figure 2.4 Side Force Comparison

It can be seen in Figure 2.4 that with an offset of 30 mm the side force during the power stroke has reduced greatly. The side force during compression and the exhaust stroke has now increased due to the larger connecting rod angle.

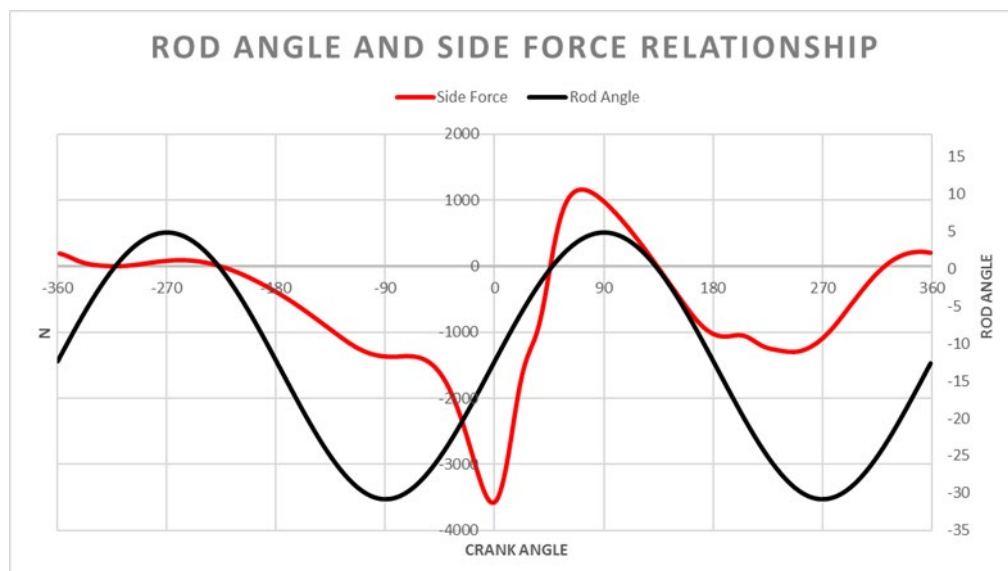


Figure 2.5 Connecting Rod Angle and Side Force Relationship

Figure 2.5 shows that when the connecting rod angle is negative i.e. pushing the piston into the antithrust side of the cylinder and that the increase in angle shows an increase in magnitude of the side force. As can be seen that the side force now has a higher magnitude negative force and also the force in negative more often throughout the cycle A modern piston has asymmetric skirts due to

weight saving and optimisation. The skirt on the thrust side is much larger than the skirt on the anti-thrust side because without offset the thrust side see much greater load and the anti-thrust side does not need as much material therefore weight can be saved. But with offset the load seen on the thrust side have been reduced but increased on the anti-thrust side which means pistons may have to be designed slightly differently than they are for engines with no offset.

2.3 Effect on Stroke

The stroke increases with crankshaft offset, this is important because this now changes the displacement of the engine and if it is a race engine the engine may become illegal. The reason for the increase in stroke is that the increase in crank angle at where bottom dead centre is means that the piston now sits lower in the cylinder. The advantage of this is now that the compression ratio has increased providing the same clearance volume is kept.

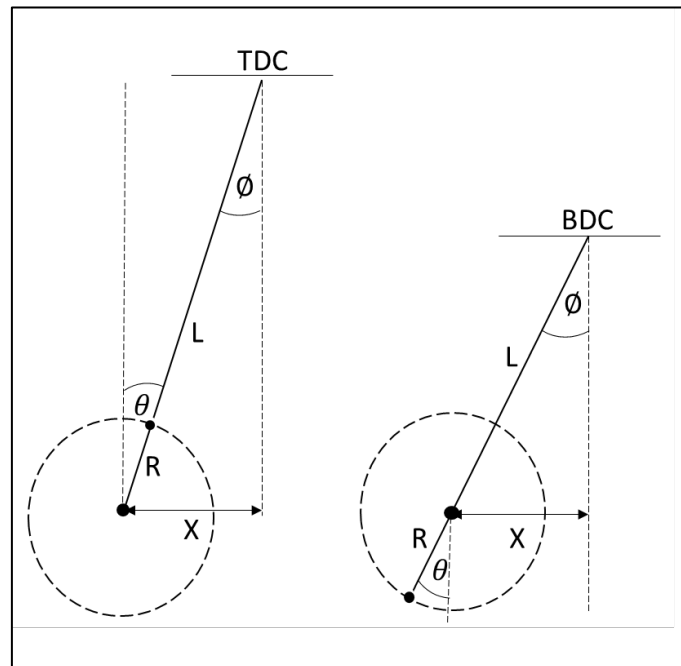


Figure 2.6 Offset Crankshaft Stroke

To calculate the stroke the distance the piston has travel from TDC to BDC must be calculated. Without off set this would simply be $R+L$ for TDC and $L-R$ for BDC, however due to the crankshaft and connecting rod being at an angle their vertical components need to be calculated. It is assumed that at TDC and BDC that crank pin and connecting rod are inline. The theory was tested on data from (Shin, et al., 2004) and shown to be correct.

From Figure 2.6 it can be seen that at TDC

$$\Phi_{TDC} = \frac{X}{L + R}$$

And at BDC

$$\phi_{BDC} = \frac{X}{L - R}$$

Therefore

$$\text{Vertical Component at TDC} = (L + R) \cdot \cos \phi_{TDC}$$

$$\text{Vertical Component at BDC} = (L - R) \cdot \cos \phi_{BDC}$$

As

$$\text{Stroke} = \text{Piston Height at TDC} - \text{Piston Height at BDC}$$

$$\text{Stroke} = (L + R) \cdot \cos \phi_{TDC} - (L - R) \cdot \cos \phi_{BDC}$$

Equation 2.2 Stroke with and Offset Crankshaft

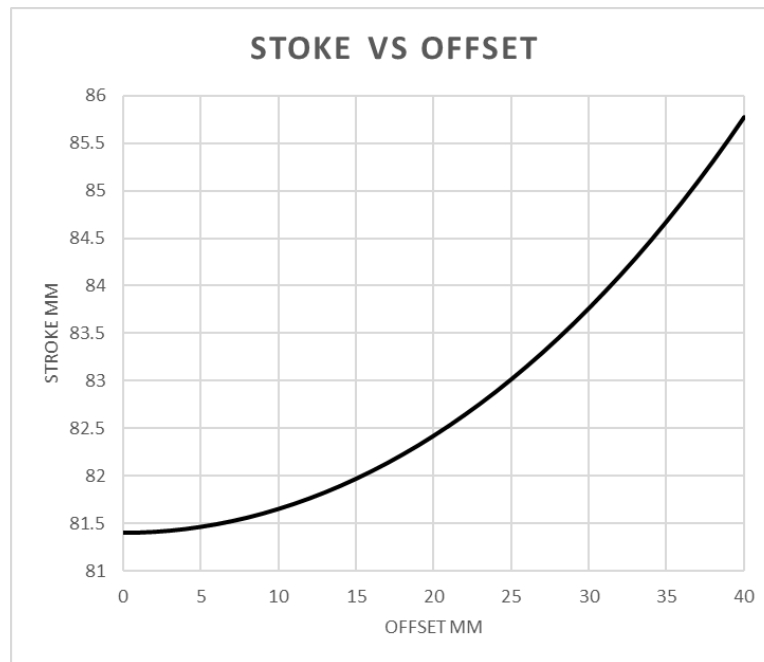


Figure 2.7 Stroke Vs Offset

Table 2 Offset effect on TDC and BDC Rod Angle

Offset mm	TDC Rod angle °	BDC Rod angle °
5	1.64	3.07
10	3.28	6.15
15	4.93	9.25
20	6.57	12.38
25	8.23	15.54
30	9.89	18.76
35	11.56	22.03
40	13.24	25.39

Figure 2.7 shows that as the offset increases so does the stroke. This is because the connecting rod angle at BDC is greater than that at TDC by roughly double. This means that for how much the initial reduction in height is at TDC then the reduction in height at BDC is roughly double therefore increasing the stroke.

From Figure 2.6 is can be see that at TDC

$$\theta = \Phi_{TDC}.$$

And at BDC

$$\theta = \Phi_{BDC}.$$

Therefore

$$\text{Crankshaft Angle at TDC} = \Phi_{TDC}$$

And

$$\text{Crankshaft Angle at BDC} = \Phi_{BDC} + 180$$

Table 3 Effect of Offset Piston Motion

Offset mm	TDC Crank Angle °	BDC Crank Angle °	Crank ° of Expansion	Crank ° of Compression
5	1.64	183.07	181.43	178.57
10	3.28	186.15	182.87	177.13
15	4.93	189.25	184.33	175.67
20	6.57	192.38	185.80	174.20
25	8.23	195.54	187.32	172.68
30	9.89	198.76	188.87	171.13
35	11.56	202.03	190.48	169.52
40	13.24	205.39	192.15	167.85

Table 4 Effect on Compression Ratio

Offset mm	Compression Ratio
0	9.93
5	9.94
10	9.96
15	9.99
20	10.04
25	10.11
30	10.19
35	10.29
40	10.41

2.4 Effect on FMEP

Table 5 Offset Effect on Piston Skirt FMEP Full Load

FMEP Piston Skirt Full Load						
Offset mm	1000	2000	3000	4500	6000	Average
0	0.587	1.023	1.123	1.193	1.208	1.027
1	0.574	1.002	1.102	1.173	1.190	1.008
2	0.562	0.982	1.082	1.153	1.172	0.990
3	0.555	0.968	1.068	1.140	1.162	0.979
4	0.543	0.949	1.048	1.121	1.146	0.961
5	0.532	0.930	1.029	1.103	1.130	0.945
6	0.524	0.916	1.015	1.090	1.120	0.933
7	0.513	0.898	0.997	1.073	1.105	0.917
8	0.502	0.880	0.979	1.056	1.091	0.902
9	0.495	0.866	0.966	1.044	1.082	0.891
10	0.485	0.850	0.949	1.029	1.070	0.876
11	0.475	0.833	0.933	1.014	1.059	0.863
12	0.467	0.820	0.920	1.003	1.050	0.852
13	0.458	0.805	0.905	0.990	1.041	0.840
14	0.449	0.790	0.891	0.978	1.032	0.828
15	0.442	0.778	0.879	0.967	1.026	0.818
16	0.434	0.764	0.867	0.957	1.019	0.808
17	0.426	0.752	0.855	0.948	1.014	0.799
18	0.419	0.740	0.844	0.939	1.009	0.790
19	0.413	0.728	0.833	0.931	1.007	0.783
20	0.407	0.718	0.825	0.925	1.006	0.776
21	0.401	0.707	0.815	0.919	1.004	0.769
22	0.396	0.699	0.807	0.915	1.005	0.764
23	0.392	0.691	0.802	0.913	1.009	0.761
24	0.387	0.682	0.794	0.909	1.010	0.756
25	0.384	0.676	0.790	0.910	1.017	0.756
26	0.383	0.672	0.788	0.913	1.026	0.757
27	0.379	0.665	0.783	0.912	1.031	0.754
28	0.380	0.664	0.784	0.919	1.044	0.758
29	0.381	0.664	0.787	0.927	1.060	0.764
30	0.384	0.667	0.792	0.938	1.077	0.772

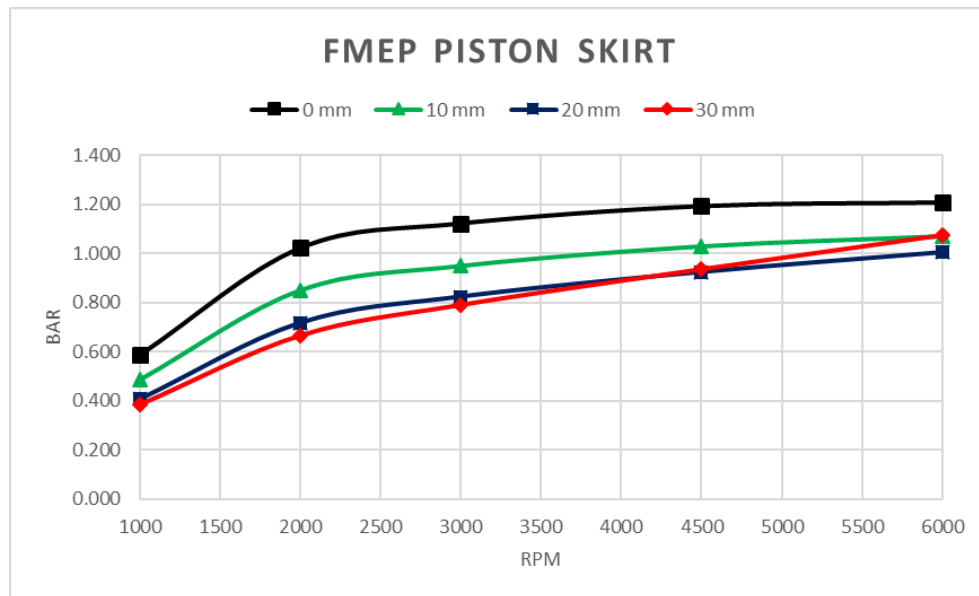


Figure 2.8 Comparison of Piston Skirt FMEP with Different Offsets

The increase in offset reduces the FMEP up until a certain point and then the FMEP starts to rise again. This is because although the friction has reduced on the power stroke it is making the compression stroke and exhaust stroke worse due to the increased rod angle and once the offset is enough to make the rod angle so great on these strokes the friction on these strokes start to overcome the gains made on the power stroke. This can be seen in Figure 2.8 where the FMEP reduces with each offset up until 30mm. After 3000 RPM the FMEP rises to roughly the same as 20mm offset and at 6000 RPM it's higher than the FMEP with 10mm offset. Table 6 shows that with 27mm offset this produces the lowest average FMEP across the RPM range with around a 27% reduction from 0mm offset.

Table 6 Offset Effect on Piston Skirt FMEP Part Load

FMEP Piston Skirt Part Load								
Offset (mm)	1	3	4	9	14	18	19	Average
0	0.120	0.141	0.205	0.267	0.228	0.412	0.408	0.255
1	0.119	0.140	0.202	0.263	0.224	0.405	0.402	0.251
2	0.118	0.138	0.199	0.259	0.221	0.398	0.397	0.247
3	0.118	0.138	0.197	0.257	0.220	0.394	0.394	0.245
4	0.117	0.136	0.195	0.253	0.217	0.387	0.388	0.242
5	0.116	0.135	0.192	0.249	0.214	0.381	0.384	0.239
6	0.115	0.134	0.190	0.247	0.213	0.377	0.381	0.237
7	0.114	0.133	0.188	0.244	0.210	0.371	0.376	0.234
8	0.114	0.132	0.186	0.241	0.208	0.365	0.372	0.231
9	0.113	0.131	0.184	0.238	0.207	0.361	0.370	0.229
10	0.112	0.130	0.182	0.236	0.205	0.356	0.367	0.227
11	0.112	0.130	0.181	0.233	0.203	0.351	0.363	0.225
12	0.112	0.129	0.179	0.231	0.202	0.348	0.362	0.223
13	0.111	0.128	0.178	0.229	0.200	0.344	0.359	0.221
14	0.111	0.128	0.176	0.227	0.199	0.340	0.357	0.220
15	0.111	0.127	0.175	0.225	0.198	0.337	0.356	0.219
16	0.111	0.127	0.174	0.224	0.198	0.335	0.355	0.218
17	0.111	0.127	0.173	0.223	0.197	0.332	0.355	0.217
18	0.111	0.127	0.172	0.221	0.197	0.330	0.354	0.216
19	0.111	0.127	0.172	0.221	0.197	0.329	0.355	0.216
20	0.112	0.127	0.172	0.221	0.197	0.328	0.356	0.216
21	0.112	0.127	0.171	0.220	0.197	0.327	0.357	0.216
22	0.113	0.128	0.172	0.220	0.198	0.327	0.359	0.217
23	0.114	0.129	0.172	0.221	0.199	0.328	0.362	0.218
24	0.115	0.130	0.173	0.223	0.201	0.330	0.365	0.220
25	0.115	0.131	0.173	0.223	0.202	0.330	0.368	0.220
26	0.117	0.132	0.175	0.225	0.204	0.333	0.373	0.223
27	0.118	0.134	0.176	0.227	0.207	0.336	0.379	0.225
28	0.119	0.135	0.177	0.228	0.209	0.338	0.383	0.227
29	0.121	0.137	0.179	0.231	0.213	0.343	0.390	0.231
30	0.123	0.139	0.182	0.235	0.216	0.349	0.398	0.235

Table 7 Test Information

LOGPOINT	Engine Speed	BMEP
	rpm	Bar
1	800.1	0.713
3	1500	0.997
4	1500	2.617
9	2000	4.004
14	2500	2.622
18	3000	7.503
19	4000	5.503

At part load it can be seen that the lowest average FMEP occurs at 21mm offset, 6mm less than at full load. And at this offset the FMEP is reduced by around 15%. This is due to the fact that net force on the power stroke is not so dominant as combustion pressures are lower therefore the gas force has reduced as seen in Figure 2.9. Whereas the inertia force stays the same at each RPM between part load and full load.

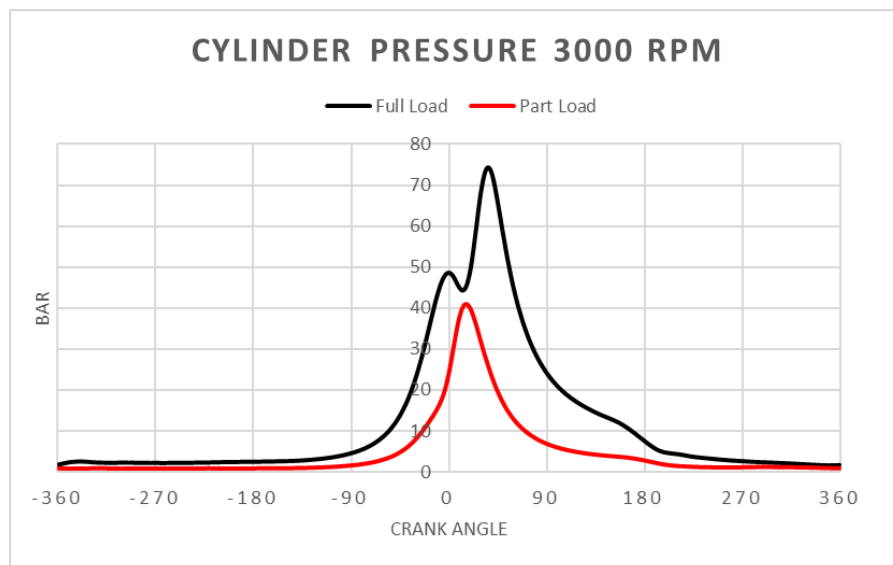


Figure 2.9 Cylinder Pressure Comparison

2.5 Effect on Torque

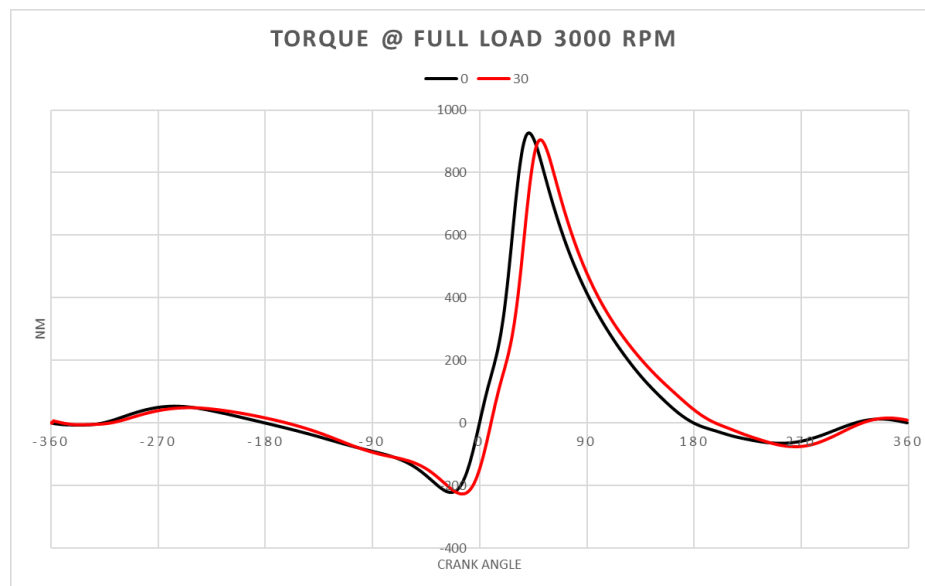


Figure 2.10 Torque Comparison

Looking at the torque throughout a cycle Figure 2.10 it can be seen that the peak torque has been reduced slightly. By looking at how the torque has been calculated it can be seen why this is.

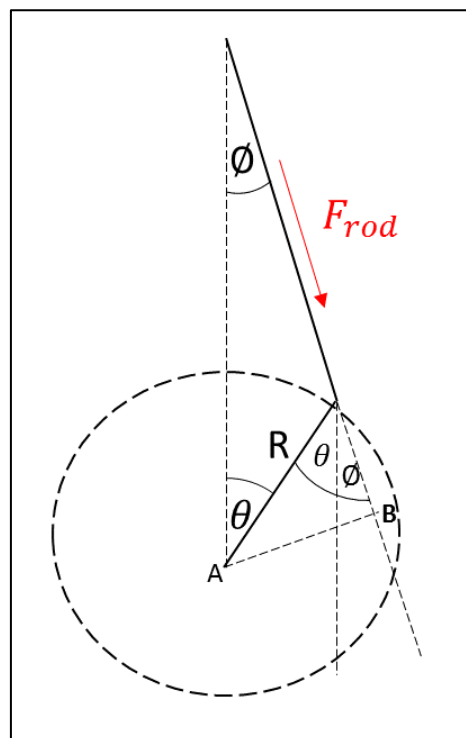


Figure 2.11 Torque Diagram

$$\text{Torque} = \text{Force} \times \text{Distance}$$

The force applied to rotating the crankshaft is the rod force and its perpendicular distance away from the rotational axis of the crankshaft is length AB from Figure 2.11.

$$AB = R \cdot \sin(\theta + \Phi)$$

Therefore

$$Torque = F_{rod} \times R \cdot \sin(\theta + \Phi)$$

Equation 2.3 Instantaneous Crankshaft Torque

Having seen previously that offsetting the crankshaft reduces the rod angle at peak pressure, this makes $R \cdot \sin(\theta + \Phi)$ a smaller number therefore the torque becomes smaller.

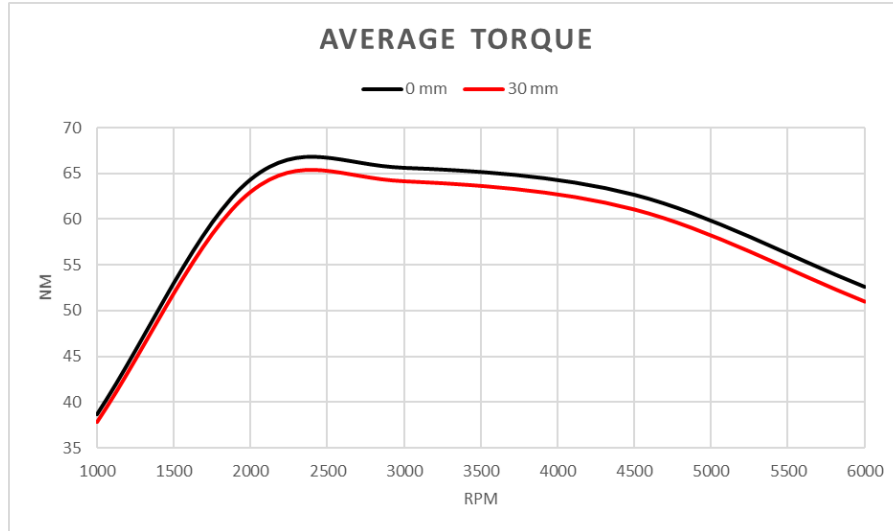


Figure 2.12 Average Torque Comparison

Comparing the average torque across the RPM range with 0mm and 30mm offset shows that there is only a very small reduction, averaging to around 2.5%.

2.6 Effect on Piston Motion

(Shin, et al., 2004) stated that there is potential for better combustion in an engine with crankshaft offset due to the piston dwelling longer at TDC and giving a more constant volume combustion. In order to see whether this is true the distance that the piston had moved down 15° either side of TDC was studied. Looking at a standard offset around 10mm it seems that there is very little if any difference in dwell time at TDC and when increasing the offset to an extreme of 95mm it shows that in fact the piston dwells less at TDC therefore it doesn't look like any advantage will be gained.

The method for calculating the distance the piston have moved is the same as previously except the starting position changes from

$$R + L$$

To

$$(R + L)\cos\phi_{TDC}.$$

Giving the equation

$$S = (R + L) \cdot \cos\phi_{TDC} - (R \cdot \cos\theta + L \cdot \cos\phi)$$

Equation 2.4 Distance Piston has Moved with Offset Crankshaft

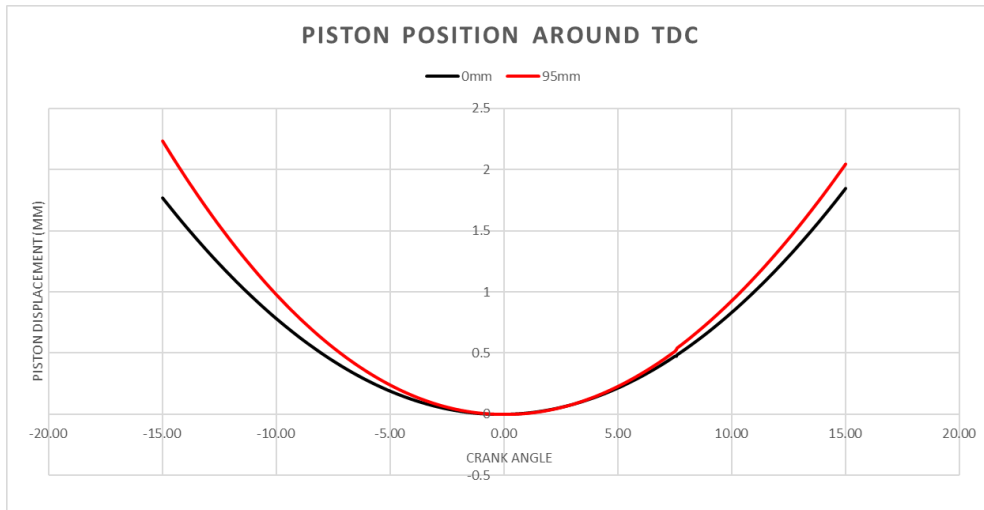


Figure 2.13 Piston Position Comparison

Figure 2.13 shows that with 95mm offset as the piston is approaching TDC it is further away than what it would be with 0mm offset. This means the volume would be greater. This can also be seen after TDC where it becomes further away from TDC quicker

2.7 Conclusion

By offsetting the crankshaft from the cylinder axis towards the thrust side of the engine it has been shown that the side force can be reduced greatly which in turn reduces the friction. However there is a limit as to how much offset can be used before the FMEP starts to rise again. For part load the best offset distance is less than full load because the reduction in work done on the power stroke is not as great as the increase in work done on the compression and exhaust strokes because of the lower combustion pressures. The study showed that although the reduced connecting rod angle does reduce the torque it is only by a small amount and it is not likely to negate the benefits of reduced friction. Claims that the piston may dwell around TDC for longer resulting in a more constant volume and better combustion have been disproved and in fact shows that at very high offset the piston dwells less at TDC. The stroke increases with offset which means that the cylinder displacement will increase. This needs to be considered as it may make the engine illegal if used in racing, or even could be used as an advantage at lower level racing where the modification may not be spotted by scrutineers. If the clearance volume is kept the same then the compression ratio will increase. This will improve the efficiency of the engine but it could also induce knock. The optimum offset is different for part load and full load therefore the offset distance should be tailored for the application of the vehicle

2.8 Further Work

The best offset for full load was shown to be 27mm, however this might not be physically possible due to engine geometry and may cause contact between the connecting rod and the cylinder. A 3D CAD model of the engine block, crankshaft and connecting rod assembly could be created and then modified to incorporate offset and then it can give an idea as to how much offset is possible with the current engine architecture.

Research suggests that the piston secondary motion (tilt) is effected by offsetting the crankshaft which has an effect on the friction (Nakayama, et al., 2000). A more complex model of the piston skirt friction could be created looking at how the secondary motion is affected.

3 Waste Energy Recovery

3.1 Thermodynamics

When looking at how much energy is in the exhaust gas the first law of thermodynamics can be applied in the form of a steady flow energy equation (Lumley, 1999), for 1 Kg of exhaust gas

$$\dot{Q} - \dot{W} = \dot{m} \left[\left(h + \frac{v^2}{2} + gz \right) \right]_{out} - \dot{m} \left[\left(h + \frac{v^2}{2} + gz \right) \right]_{in}$$

Equation 3.1 Steady Flow Energy Equation

Where $\dot{Q}$ is the heat transfer rate into the system, $\dot{W}$ is the work transfer rate out of the system, $\dot{m}$ is the mass flow rate, h is the enthalpy per unit mass, $\frac{v^2}{2}$ is the kinetic energy per unit mass and gz is the potential energy per unit mass. However many of these terms can be neglected to make a simplified model such as the heat transfer rate as there is no added heat to the system, the kinetic energy because for 1 Kg of fluid at 45 m/s only equates to 1 KJ/Kg and the change in potential energy is dependent on change in height which is virtually zero (Lumley, 1999). With these factors neglected Equation 3.1 becomes

$$-\dot{W} = \dot{m}(h_{out} - h_{in})$$

And then rearranged to

$$\dot{W} = \dot{m}(h_{in} - h_{out})$$

Equation 3.2 System Energy per Second

The enthalpy used for the calculations is the average enthalpy per Kg of the exhaust gas.

3.2 Engine Efficiency

Internal combustion engines use around 2/3 of the oil in the world and 30-40% of the energy is wasted as hot exhaust gas. This means there is a lot of available energy that could potentially be recovered. With modern vehicles becoming more electrified there is now a use for recovered energy to go back into charging the battery for example on a hybrid vehicle. Using test data from a ford fox 1.0L Ecoboost engine it can be calculated how much energy is being lost.

First of all the fuel power can be calculated to gain an idea of how efficient the engine is.

$$\text{Fuel Power} = \text{Mass Flow Rate of Fuel} \times \text{Stoichiometric Calorific Value}$$

Equation 3.3 Maximum Fuel Power

To give the absolute efficiency of the engine the stoichiometric calorific value is used to give the maximum fuel power. This is also assuming that combustion is 100% efficient.

$$\text{Engine Efficiency} = \frac{\text{Power Produced by Engine}}{\text{Fuel Power}}$$

Equation 3.4 Engine Efficiency

The power produced by the engine was obtained from the data.

	Thermal Efficiency															
	800	1000	1250	1500	1750	2000	2250	2500	3000	3500	4000	4500	5000	5500	6000	6500
FL		27.95%	28.24%	28.10%	29.00%	29.42%	29.37%	29.47%	28.29%	28.91%	29.46%	29.68%	27.34%	27.23%	26.83%	27.53%
200						29.44%	29.28%	29.58%	28.50%	29.89%						
190						29.73%	29.94%	29.97%	28.35%	30.21%						
180					29.18%	30.16%	30.33%	30.31%	29.83%	29.80%	29.47%					
170				28.22%	28.55%	30.48%	30.61%	30.61%	30.61%	29.39%	30.30%	30.11%				
160				28.51%	29.23%	30.88%	31.08%	30.96%	31.03%	31.37%	30.89%	30.61%	28.50%			
150				29.05%	29.68%	31.01%	31.22%	31.02%	31.67%	31.63%	31.35%	31.34%	30.12%	28.42%		
140			28.29%	29.21%	30.20%	31.11%	31.38%	31.23%	32.10%	31.96%	31.57%	31.89%	31.19%	29.99%	27.68%	
130			28.91%	30.14%	31.14%	31.47%	31.93%	32.06%	32.69%	32.26%	31.69%	32.02%	31.68%	30.57%	28.71%	
120			29.12%	30.74%	30.78%	31.50%	32.15%	32.45%	33.19%	32.33%	31.53%	31.97%	31.87%	30.88%	29.67%	28.23%
110		27.94%	29.67%	30.78%	30.95%	32.06%	32.41%	32.54%	33.04%	32.54%	31.66%	31.95%	31.83%	31.25%	29.85%	28.86%
100		27.90%	30.72%	30.67%	31.63%	32.67%	32.87%	32.52%	32.78%	32.65%	32.16%	31.94%	31.54%	30.94%	30.20%	28.53%
90		29.37%	31.57%	31.57%	32.26%	33.18%	33.23%	32.83%	32.94%	33.02%	32.32%	31.92%	31.38%	30.71%	30.05%	29.09%
80		30.67%	31.60%	32.13%	32.81%	33.32%	33.31%	32.89%	33.04%	33.13%	32.10%	31.76%	30.76%	30.08%	29.68%	28.72%
70		31.01%	31.76%	31.84%	32.09%	32.72%	32.58%	32.32%	32.78%	32.69%	31.78%	31.49%	30.36%	29.71%	29.06%	28.02%
60		30.87%	31.36%	31.55%	31.73%	32.25%	32.11%	31.51%	31.90%	31.95%	31.05%	30.84%	29.70%	28.98%	28.22%	27.29%
50		30.46%	30.45%	30.45%	31.09%	31.35%	31.26%	30.90%	30.97%	30.71%	30.02%	29.55%	28.83%	28.13%	27.09%	26.25%
40	26.75%	28.58%	28.79%	28.92%	29.16%	29.60%	29.48%	29.29%	29.08%	28.73%	28.29%	27.65%	26.91%	26.18%	25.42%	24.55%
30	25.27%	26.17%	26.36%	26.26%	26.58%	26.76%	26.68%	26.84%	26.59%	26.04%	25.69%	24.98%	24.20%	23.42%	22.98%	21.91%
20	22.32%	21.89%	22.28%	22.06%	22.77%	22.80%	22.96%	22.84%	22.77%	22.17%	21.56%	20.82%	20.11%	19.52%	18.96%	17.91%
10	16.14%	13.93%	15.48%	15.94%	15.91%	15.79%	15.82%	15.60%	16.34%	15.42%	14.18%	13.49%	13.22%	13.17%	12.89%	12.23%
5	9.87%	8.46%	8.94%	9.40%	11.24%	9.42%	9.06%	9.78%	8.52%	9.25%	8.33%	8.13%	7.59%	8.29%	7.48%	7.40%
2	3.70%	6.07%	2.57%	4.23%	4.48%	5.31%	4.82%	4.23%	4.37%	4.11%	3.56%	3.68%				

Figure 3.1 Thermal Efficiency

Figure 3.1 shows that at best the engine is only 33% efficient at turning the fuel power into useful work. This means that 66% of the fuel power is being lost. It is estimated that 1/3 of the fuel power goes into mechanical work, 1/3 is lost to the coolant system and 1/3 leaves the engine as hot exhaust gas.

From the mass flow rate of the air and fuel and the temperatures of the exhaust gas, the energy per second of the gas can be calculated.

As Equation 3.2 shows, the work rate or power out of the system is $\dot{W} = \dot{m}(h_{in} - h_{out})$. It can be imagined that the system being looked at is the complete exhaust system, therefore h_{in} is from when the exhaust gas leaves the cylinder and h_{out} is from the exhaust gas when it exits the exhaust system into the atmosphere, for further calculations h_{in} and h_{out} will be named h_1 and h_2 respectively

$$\text{Exhaust Gas Energy per Second} = \dot{m}(h_1 - h_2).$$

Where $\dot{m}$ is the mass flow rate of the air and fuel, h_1 is the average enthalpy per Kg of the exhaust gas after leaving the cylinder and h_2 is the average enthalpy per Kg of the exhaust gas when it leaves the exhaust system. Assuming the temperature of the exhaust goes back down to an ambient temperature of 20°C as it leaves the system then the enthalpy will be 0, therefore the equation can be simplified to

$$\text{Exhaust Gas Energy per Second} = \dot{m} \times h_1.$$

Equation 3.5 Exhaust Gas Power

	Exhaust Power Kw															
	800	1000	1250	1500	1750	2000	2250	2500	3000	3500	4000	4500	5000	5500	6000	6500
FL		8.34	15.21	24.28	34.65	47.15	55.42	65.62		86.04	103.92	105.91	120.74	125.72	125.52	110.18
200						47.13	55.61	65.09		95.72						
190						43.81	50.71	59.80	72.21	85.62						
180					33.21	40.00	46.83	54.74	67.18	78.29	97.83					
170				24.20	29.53	36.59	42.96	50.03	63.03	70.93	88.87	99.83				
160				22.59	26.91	33.39	38.82	44.97	56.99	67.64	81.19	90.37	110.51			
150				20.39	24.72	30.12	34.95	40.42	50.38	61.22	73.22	82.06	97.62	116.00		
140			15.26	18.74	22.35	27.00	31.15	35.98	44.74	54.90	65.32	74.97	87.61	99.62	116.48	
130			13.77	16.72	19.93	23.59	27.09	31.06	39.25	48.79	59.09	67.15	78.05	88.13	104.06	
120			12.75	14.93	17.47	21.02	23.98	27.10	34.30	43.79	53.90	60.87	68.70	79.70	89.24	104.44
110		8.27	11.33	13.69	15.55	18.45	21.43	24.29	30.93	38.87	47.56	54.58	61.19	70.87	80.78	92.79
100		7.05	9.83	11.73	13.92	16.54	19.12	21.84	27.64	34.59	41.57	48.56	55.45	64.35	73.76	83.50
90		6.29	8.52	10.29	12.41	14.65	17.06	19.40	24.66	30.30	36.38	43.04	49.29	57.04	65.96	77.12
80		5.32	7.28	9.04	10.89	12.68	14.90	17.25	21.87	26.61	32.56	37.93	44.04	51.49	58.16	67.57
70		4.43	6.17	7.80	9.47	10.96	12.98	14.98	18.71	23.08	28.50	33.09	38.73	44.52	51.22	59.58
60		3.65	5.08	6.50	7.80	9.16	10.89	12.83	16.18	19.78	24.47	28.58	33.63	38.70	44.59	51.33
50		2.96	4.12	5.32	6.36	7.53	8.84	10.41	13.47	16.76	20.60	24.31	28.68	32.81	38.14	43.67
40	1.92	2.37	3.30	4.27	5.07	5.96	7.16	8.39	11.08	13.94	17.03	20.31	24.24	28.07	32.23	37.09
30	1.55	1.81	2.53	3.28	4.05	4.71	5.72	6.66	8.78	11.22	13.67	16.52	19.78	23.34	26.56	30.94
20	0.91	1.25	1.81	2.42	3.06	3.66	4.30	5.05	6.59	8.55	10.63	13.09	15.65	18.25	20.87	24.39
10	0.62	0.82	1.28	1.64	2.09	2.46	2.85	3.28	4.67	5.94	7.97	9.56	11.63	13.45	15.69	18.58
5	0.39	0.58	0.98	1.30	1.81	2.26	2.59	2.93	3.95	5.02	6.64	8.02	9.79	11.28	12.86	15.39
2	0.33	0.52	0.81	1.02	1.39	1.98	2.15	2.62	3.79	4.72	6.11	7.15				

Figure 3.2 Exhaust Power

When comparing the exhaust gas power to the fuel power it is best to look at the true fuel power. This true power includes the consideration that combustion is not 100% and also uses different calorific values depending on fuel equivalence ratio.

$$\text{True Fuel Power} = 0.98 \times \dot{m} \times \text{Actual Calorific Value}$$

Equation 3.6 True Fuel Power

0.98 is the combustion efficiency factor (Bosch, 2004) and $\dot{m}$ is the fuel flow rate in Kg/s.

	Exhaust Power % of True Fuel Power															
	800	1000	1250	1500	1750	2000	2250	2500	3000	3500	4000	4500	5000	5500	6000	6500
FL		21.13%	23.93%	26.51%	30.30%	34.59%	35.72%	38.10%		40.53%	41.28%	41.69%	43.83%	43.82%	43.28%	41.79%
200						34.60%	36.22%	38.03%		40.93%						
190						33.87%	35.07%	37.18%	39.08%	40.23%						
180					29.93%	33.14%	34.63%	36.40%	38.24%	38.76%	40.62%					
170				26.46%	27.65%	32.48%	34.00%	35.54%	37.32%	36.88%	39.67%	40.45%				
160				26.13%	27.35%	31.37%	32.62%	34.41%	36.36%	37.40%	38.71%	39.59%	42.72%			
150				25.68%	27.21%	30.28%	31.46%	33.02%	35.01%	36.40%	37.68%	38.16%	40.34%	42.25%		
140			24.15%	25.48%	26.83%	29.06%	30.27%	31.29%	33.36%	35.35%	36.57%	37.49%	38.55%	39.95%	42.80%	
130			23.90%	25.14%	26.57%	27.83%	28.78%	29.83%	32.07%	34.17%	35.55%	36.30%	37.04%	38.79%	41.54%	
120			24.06%	24.87%	24.94%	27.01%	27.85%	28.57%	30.78%	32.29%	34.89%	35.10%	35.52%	37.37%	38.37%	41.03%
110		21.09%	23.80%	24.87%	24.37%	26.21%	27.36%	27.98%	30.16%	32.46%	33.52%	34.32%	34.46%	35.68%	37.55%	39.04%
100		19.48%	23.52%	23.33%	24.53%	26.11%	27.16%	27.67%	29.43%	31.46%	32.41%	33.56%	34.03%	35.23%	36.60%	38.22%
90		20.17%	23.32%	23.47%	24.79%	26.43%	27.30%	27.61%	29.34%	30.90%	31.91%	33.03%	33.42%	34.47%	35.75%	37.36%
80		19.85%	22.47%	23.56%	24.90%	25.70%	26.89%	27.61%	29.40%	30.65%	31.87%	32.58%	33.02%	34.32%	35.01%	36.38%
70		19.23%	21.75%	23.13%	24.22%	25.01%	26.18%	26.97%	28.45%	29.98%	31.41%	32.19%	32.75%	33.48%	34.53%	35.73%
60		18.38%	20.66%	22.07%	23.04%	24.10%	25.25%	26.21%	27.97%	29.31%	30.91%	31.76%	32.45%	33.10%	34.03%	35.01%
50		17.59%	19.55%	21.00%	21.94%	22.91%	23.95%	25.07%	27.05%	28.67%	30.21%	31.14%	32.21%	32.66%	33.57%	34.41%
40	15.14%	16.81%	18.57%	19.92%	20.70%	21.46%	22.88%	24.00%	26.30%	27.91%	29.31%	30.44%	31.75%	32.46%	33.34%	34.14%
30	14.29%	15.64%	17.48%	18.94%	19.92%	20.57%	22.04%	23.18%	25.29%	27.02%	28.50%	29.81%	31.17%	32.26%	32.94%	33.83%
20	12.23%	14.09%	15.96%	18.08%	19.32%	20.29%	21.42%	22.56%	24.47%	26.30%	27.89%	29.53%	30.74%	31.49%	31.91%	32.93%
10	11.55%	13.27%	16.13%	17.22%	18.44%	19.22%	20.19%	20.91%	23.52%	25.41%	27.50%	28.85%	30.02%	30.35%	30.81%	31.85%
5	8.84%	11.32%	15.14%	16.92%	18.86%	21.08%	21.67%	22.10%	24.57%	25.78%	27.40%	28.25%	29.44%	29.49%	29.66%	30.97%
2	8.42%	10.94%	14.53%	14.91%	17.31%	20.85%	20.55%	21.98%	25.51%	26.91%	27.42%	28.08%				

Figure 3.3 Exhaust Power Percentage

At high load and high RPM the exhaust gas power is over 40% of the true fuel power. At high RPM the efficiency if the engine is reducing by much Figure 3.1, but because there is less time for heat transfer less of the power is getting absorbed into the coolant system therefore it is released down the exhaust.

$$\text{Power Into Oil and Water} = \text{True Fuel Power} - \text{Exhaust Power} - \text{Brake Power}$$

Equation 3.7 Oil and Water Power

Oil + Water Power % of True Fuel Power																
	800	1000	1250	1500	1750	2000	2250	2500	3000	3500	4000	4500	5000	5500	6000	6500
FL		50.35%	47.26%	44.82%	40.11%	34.98%	33.90%	31.41%		27.46%	27.39%	25.89%	24.66%	24.80%	24.98%	25.64%
200						34.94%	33.07%	31.37%		28.15%						
190						35.37%	33.96%	31.81%	28.67%	27.21%						
180					40.30%	35.67%	33.99%	32.25%	29.61%	28.68%	28.05%					
170				44.75%	43.22%	35.99%	34.34%	32.80%	31.02%	30.57%	28.55%	27.10%				
160				44.78%	42.83%	37.12%	35.67%	33.56%	31.55%	30.16%	29.34%	27.41%	24.87%			
150				44.68%	42.51%	38.09%	36.68%	34.89%	32.23%	30.88%	29.89%	28.97%	27.19%	26.28%		
140			46.99%	44.73%	42.36%	39.20%	37.71%	36.85%	33.89%	31.60%	30.77%	29.52%	29.18%	27.72%	24.88%	
130			46.61%	44.11%	41.67%	40.06%	38.64%	37.47%	34.58%	32.46%	31.67%	30.57%	30.63%	28.26%	25.81%	
120			46.23%	43.77%	43.65%	40.86%	39.35%	38.33%	35.36%	33.26%	32.49%	32.29%	31.96%	30.24%	29.21%	26.87%
110		50.40%	45.93%	43.74%	44.05%	41.09%	39.58%	38.82%	36.13%	33.88%	34.18%	33.08%	33.07%	32.44%	30.27%	29.43%
100		52.05%	45.14%	45.38%	43.20%	40.55%	39.30%	39.15%	37.12%	35.24%	34.78%	33.85%	33.79%	33.21%	32.16%	30.60%
90		49.86%	44.47%	44.32%	42.30%	39.72%	38.79%	38.90%	37.05%	35.41%	35.12%	34.41%	34.56%	34.21%	33.60%	32.96%
80		48.86%	45.29%	43.66%	41.62%	40.30%	39.13%	38.84%	36.90%	35.56%	35.38%	35.01%	35.60%	34.99%	34.71%	34.31%
70		49.13%	45.85%	44.38%	43.04%	41.61%	40.58%	40.05%	38.11%	36.67%	36.16%	35.68%	36.28%	36.21%	35.82%	35.68%
60		50.12%	47.35%	45.74%	44.59%	43.00%	42.00%	41.64%	39.48%	38.09%	37.41%	36.78%	37.25%	37.33%	37.18%	37.15%
50		51.34%	49.39%	47.93%	46.34%	45.11%	44.16%	43.41%	41.35%	40.00%	39.17%	38.71%	38.38%	38.65%	38.79%	38.81%
40	57.57%	54.04%	52.06%	50.57%	49.56%	48.34%	47.04%	46.12%	44.03%	42.78%	41.82%	41.35%	40.80%	40.83%	40.73%	40.81%
30	59.93%	57.66%	55.62%	54.26%	52.96%	52.12%	50.74%	49.44%	47.58%	46.41%	45.29%	44.70%	44.14%	43.85%	43.61%	43.81%
20	65.00%	63.57%	61.31%	59.41%	57.46%	56.45%	55.15%	54.14%	52.30%	51.08%	50.11%	49.23%	48.75%	48.59%	48.75%	48.79%
10	71.98%	72.52%	68.08%	66.52%	65.32%	64.68%	63.67%	63.17%	59.81%	58.87%	58.04%	57.38%	56.49%	56.21%	56.04%	55.67%
5	81.10%	80.05%	75.75%	73.49%	69.67%	69.31%	69.08%	67.92%	66.74%	64.79%	64.10%	63.46%	62.82%	62.05%	62.71%	61.48%
2	87.81%	82.87%	82.85%	80.78%	78.12%	73.73%	74.53%	73.71%	70.03%	68.90%	68.95%	68.17%				

Figure 3.4 Oil and Water Power Percentage

Figure 3.4 shows that at lower RPM and lower loads the power into the oil and water increases. This is due to the fact that there is more time for heat transfer to occur. At lower RPM everything is happening much slower. At low load the volumetric efficiency is poor therefore there is more distance between fuel molecules meaning combustion slows down.

The power into the oil and water can be broken down further into how much has come from the heat due to friction leaving the rest of the power coming from combustion.

From FMEP data

$$\text{Work Done in 2 Revs} = \text{FMEP} \times \text{Engine Displacement}$$

$$\text{Work Done per Second} = \frac{\text{Work Done in 2 Revs}}{2} \times \frac{\text{RPM}}{60}$$

	Friction Power % of Oil and Water Power															
	800	1000	1250	1500	1750	2000	2250	2500	3000	3500	4000	4500	5000	5500	6000	6500
FL		4.84%	4.21%	3.91%	4.49%	5.49%	6.13%	7.05%		9.91%	8.71%	10.21%	11.72%	12.63%	13.25%	16.26%
200						5.61%	6.36%	6.99%		7.36%						
190						5.62%	6.42%	7.13%	6.15%	8.17%						
180					4.54%	5.59%	6.45%	7.05%	7.46%	6.56%	9.72%					
170				3.92%	4.15%	5.60%	6.41%	6.69%	7.55%	8.69%	9.96%	10.10%				
160				4.11%	4.41%	5.54%	6.07%	6.76%	7.34%	7.55%	9.65%	10.14%	12.05%			
150				4.17%	4.62%	5.56%	6.14%	6.65%	7.03%	7.66%	9.54%	9.83%	10.52%	12.07%		
140			4.22%	4.39%	4.81%	5.46%	6.15%	6.46%	7.69%	9.66%	9.92%	10.13%	11.68%		14.47%	
130			4.44%	4.75%	5.20%	5.47%	6.10%	6.59%	6.94%	7.51%	9.83%	9.70%	10.02%	12.07%	14.08%	
120			4.70%	5.11%	5.03%	5.69%	6.33%	6.78%	7.20%	7.76%	9.91%	9.46%	10.13%	11.74%	13.03%	15.37%
110		4.91%	5.02%	5.42%	5.42%	6.08%	6.62%	7.07%	7.39%	8.23%	9.78%	9.79%	10.45%	11.25%	12.83%	14.56%
100		4.76%	5.50%	5.50%	5.98%	6.72%	7.15%	7.51%	7.57%	8.39%	9.96%	10.13%	11.14%	11.20%	13.28%	14.91%
90		5.55%	6.12%	6.27%	6.72%	7.31%	7.93%	8.19%	8.31%	9.28%	10.85%	10.98%	12.12%	12.08%	12.92%	14.02%
80		6.08%	6.50%	6.98%	7.50%	8.01%	8.43%	8.88%	9.15%	10.15%	11.74%	11.81%	12.31%	12.85%	13.38%	15.05%
70		6.60%	6.85%	7.23%	7.59%	8.27%	8.72%	9.32%	9.83%	10.80%	12.53%	12.82%	13.34%	13.83%	14.42%	16.31%
60		7.08%	7.21%	7.66%	7.98%	8.42%	9.17%	9.66%	10.24%	11.29%	13.44%	14.09%	14.71%	15.05%	15.73%	17.02%
50		7.88%	7.77%	8.40%	8.64%	9.17%	9.73%	10.23%	10.63%	11.92%	13.78%	15.06%	16.50%	16.68%	17.16%	18.60%
40	9.46%	8.69%	8.73%	9.06%	9.21%	9.76%	10.39%	11.06%	11.13%	12.50%	14.61%	16.22%	17.96%	18.21%	19.08%	20.43%
30	9.80%	9.69%	9.77%	10.07%	10.37%	10.84%	11.34%	12.26%	12.20%	13.40%	16.18%	17.62%	19.69%	19.84%	21.22%	22.19%
20	10.03%	11.15%	11.19%	11.72%	11.96%	12.71%	13.13%	13.84%	14.14%	15.20%	18.23%	20.00%	21.89%	22.34%	22.78%	24.62%
10	11.87%	13.62%	13.85%	14.37%	14.71%	15.64%	15.76%	16.50%	16.34%	18.34%	21.29%	22.58%	25.01%	24.52%	25.81%	26.89%
5	12.50%	14.38%	14.84%	15.88%	15.37%	16.30%	17.42%	18.16%	18.08%	20.23%	22.84%	23.83%	26.20%	27.87%	26.72%	29.29%
2	12.21%	14.33%	14.96%	16.03%	16.41%	17.29%	18.40%	18.85%	18.89%	20.90%	23.74%	23.87%				

Figure 3.5 Friction Power Percentage of Oil and Water Power

	Friction Power % of Engine Power															
	800	1000	1250	1500	1750	2000	2250	2500	3000	3500	4000	4500	5000	5500	6000	6500
FL		8.55%	6.90%	6.11%		6.32%	6.85%	7.26%	5.68%	8.50%	7.61%	8.15%	9.17%	9.98%	10.43%	12.80%
200						6.44%	6.85%	7.16%	6.68%	6.70%						
190						6.46%	7.04%	7.32%	5.47%	6.83%						
180					6.15%	6.39%	6.99%	7.25%	6.87%	5.78%	8.70%					
170				6.09%	6.16%	6.40%	6.95%	6.93%	7.40%	8.16%	8.94%	8.43%				
160				6.33%	6.33%	6.53%	6.82%	7.08%	7.22%	7.02%	8.86%	8.42%	9.25%			
150				6.29%	6.49%	6.69%	7.07%	7.23%	6.91%	7.23%	8.79%	8.66%	8.81%	10.08%		
140			6.88%	6.58%	6.61%	6.75%	7.25%	7.47%	7.19%	7.35%	9.10%	8.88%	9.16%	10.01%	11.14%	
130			7.02%	6.81%	6.82%	6.83%	7.24%	7.55%	7.19%	7.31%	9.49%	8.95%	9.50%	10.35%	11.13%	
120			7.31%	7.13%	6.99%	7.23%	7.60%	7.85%	7.52%	7.72%	9.87%	9.37%	9.96%	10.96%	11.75%	12.86%
110		8.68%	7.62%	7.56%	7.56%	7.64%	7.93%	8.27%	7.92%	8.28%	10.35%	9.93%	10.64%	11.45%	12.08%	13.59%
100		8.70%	7.92%	7.98%	8.00%	8.17%	8.38%	8.87%	8.40%	8.87%	10.56%	10.53%	11.70%	11.78%	13.67%	14.64%
90		9.23%	8.46%	8.63%	8.63%	8.58%	9.07%	9.51%	9.16%	9.75%	11.56%	11.61%	13.08%	13.19%	14.16%	15.57%
80		9.49%	9.13%	9.30%	9.32%	9.49%	9.71%	10.28%	10.02%	10.68%	12.69%	12.76%	13.97%	14.66%	15.33%	17.63%
70		10.25%	9.70%	9.87%	9.98%	10.31%	10.64%	11.32%	11.20%	11.87%	13.98%	14.24%	15.62%	16.52%	17.42%	20.36%
60		11.27%	10.67%	10.88%	10.99%	11.01%	11.76%	12.51%	12.42%	13.20%	15.87%	16.47%	18.09%	19.00%	20.31%	22.71%
50		13.02%	12.35%	12.96%	12.63%	12.93%	13.48%	14.09%	13.91%	15.21%	17.62%	19.34%	21.53%	22.46%	24.08%	26.96%
40	19.96%	16.11%	15.48%	15.53%	15.34%	15.62%	16.25%	17.07%	16.51%	18.24%	21.16%	23.78%	26.70%	27.84%	29.95%	33.27%
30	22.78%	20.94%	20.21%	20.40%	20.25%	20.69%	21.14%	22.13%	21.40%	23.41%	27.95%	30.91%	35.20%	36.41%	39.48%	43.49%
20	28.63%	31.73%	30.20%	30.93%	29.59%	30.85%	30.91%	32.15%	31.83%	34.33%	41.53%	46.37%	52.00%	54.50%	57.42%	65.71%
10	51.86%	69.52%	59.68%	58.78%	59.19%	62.81%	62.19%	65.49%	58.62%	68.66%	85.44%	94.10%	104.80%	102.58%	110.01%	119.91%
5	100.71%	133.40%	123.31%	121.76%	93.38%	117.57%	130.21%	123.70%	138.84%	138.89%	172.33%	182.32%	212.59%	204.33%	219.46%	238.51%
2	284.10%	191.65%	472.08%	300.16%	280.15%	235.14%	279.02%	322.10%	296.87%	343.28%	450.95%	433.17%				

Figure 3.6 Friction Power Percentage of Engine Power

At low load and low rpm the friction power is a large percentage of the engine power. This is because at lower RPM everything in the engine is moving slower oil films can't be built up and there is likely to be surface contact between components.

4 Turbocharger model

Turbochargers have become commonly used on passenger car engines due to the downsizing trend to help pass emissions tests. To restore the power lost by downsizing the engine turbochargers are fitted to force more air into the cylinder and improve the volumetric efficiency of the engine which increases the inlet density (Dave Hancock, 2008).

Turbochargers extract heat energy from the exhaust to power a turbine which is coupled to a compressor that raises the pressure of the intake system to force extra charge into the cylinder. The power of the turbine uses can be calculated from the steady flow energy equation seen previously

$$\dot{W} = \dot{m}(h_{in} - h_{out})$$

Equation 4.1 Turbocharger Output Power

Where $\dot{W}$ is the shaft work that is used to power the compressor.

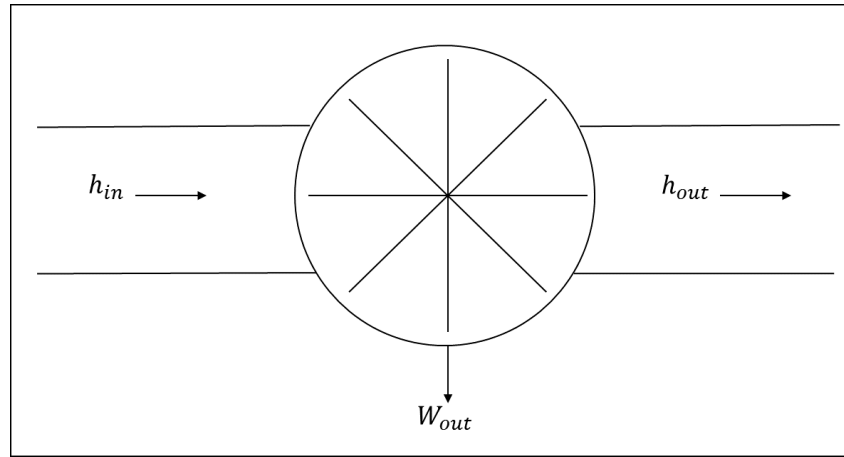


Figure 4.1 Simple Turbocharger Model

This is the model that is used by (Lumley, 1999), however the model from (Heywood, 1988) includes kinetic energy. For this model the kinetic energy will be calculated and used and then compared with the enthalpy power to see if in fact it can be ignored and a simple model is sufficient. Adding the kinetic energy into the model makes Equation 4.1 become

$$\dot{W} = \dot{m}[(h_{in} + KE_{in}) - (h_{out} + KE_{out})]$$

Equation 4.2 Turbocharger Power Output with Kinetic Energy

$$Kinetic\ Energy = \frac{1}{2}mv^2$$

Equation 4.3 Kinetic Energy

For 1 Kg of gas it can be simplified to

$$Kinetic\ Energy = \frac{v^2}{2}$$

Equation 4.4 Specific Kinetic Energy

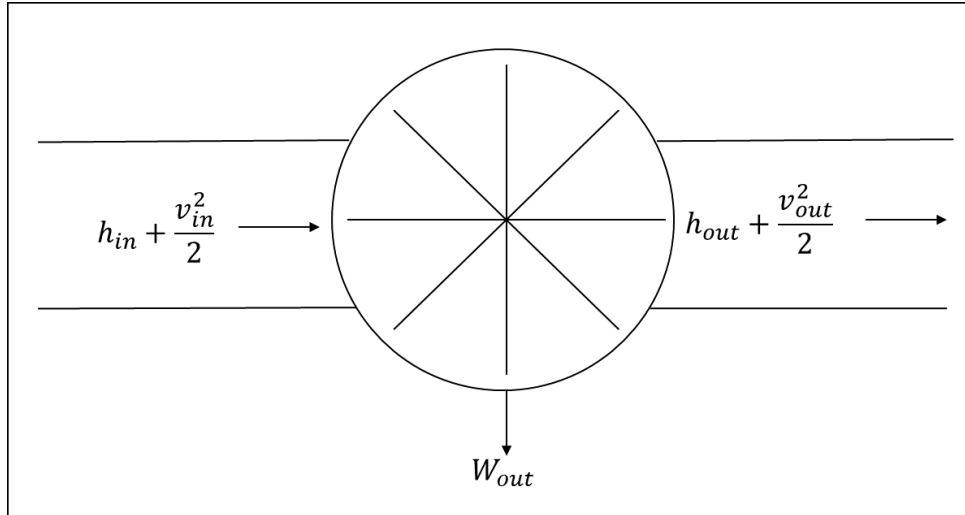


Figure 4.2 Turbocharger Model with Kinetic Energy

To calculate the velocity of the gas the mass flow rate equation can be used

$$\text{Mass Flow Rate} = \rho \cdot A \cdot v$$

Equation 4.5 Mass Flow Rate

Where ρ is the density, A is the area and v is the velocity.

This can then be rearranged for velocity giving

$$\text{Velocity} = \frac{\text{Mass Flow Rate}}{\rho \cdot A}$$

Equation 4.6 Exhaust Gas Velocity

The mass flow rate can be calculated from the fuel flow rate given in the data and the air/ fuel ratio

$$\text{Mass Flow Rate} = \frac{\text{Fuel Mass Flow Rate} + (\text{Air Fuel Ratio} \times \text{Fuel Mass Flow Rate})}{3600}$$

The fuel mass flow rate is given in Kg/hr therefore it is divided by 3600 to convert to Kg/seconds. This is the total mass flow rate through the engine. When calculating the mass flow rate through the turbocharger the waste gate needs to be considered. Not all of the mass flows through the turbine some of the gas is directed through the waste gate depending on the engines desired boost requirements.

The Area of the inlet and outlet of the turbocharger can be measured leaving the density to be calculated.

The ideal gas law can be used to calculate the density

$$PV = m \cdot R \cdot T$$

Equation 4.7 Ideal Gas Law

Where P is pressure, V is volume, m is mass, r is the specific gas constant and T is the temperature.

$$\rho = \frac{m}{V}$$

Equation 4.8 Density

Therefore the ideal gas law equation can be rearranged to

$$\frac{m}{V} = \frac{P}{R.T}$$

$$\rho = \frac{P}{R.T}$$

Equation 4.9 Density from Ideal Gas Law

The pressure and temperature of the inlet and outlet are given in the data and

$$R = \frac{\text{Universal Gas Constant}}{\text{Average Molecular weight}}$$

Equation 4.10 Specific Gas Constant

4.1 Flow through Wastegate

The wastegate on a turbocharger is used to bypass the turbine to regulate the level of boost required. It can be seen that during low load which would be the region the drive cycle operates in that the wastegate is around its highest opening point to minimise pumping losses. This is because the turbine acts as a restriction in the exhaust system making it difficult for the engine to push the exhaust gases out. By opening the wastegate this bypasses the turbine allowing a much less restrictive path for the gas to travel through at high loads. The wastegate then opens up again at high RPM and load to prevent over boosting by the compressor. In order to find out how much of the gas is bypassing the turbine to gain an accurate mass flow rate the wastegate was tested on a flow bench.

	Waste Gate Position mm															
	800	1000	1250	1500	1750	2000	2250	2500	3000	3500	4000	4500	5000	5500	6000	6500
FL		0.15	0.18	0.44	0.78	1.09	2.02	2.59	3.31	4.16	5.88	6.54	6.53	6.73	7.23	8.39
200						1.09	2.05	2.60	3.43	5.00						
190						1.17	2.08	2.77	3.84	5.35						
180					0.83	1.29	2.18	3.03	4.29	5.66	6.16					
170				0.45	0.98	1.44	2.57	3.77	4.84	5.87	6.51	6.77				
160				0.52	1.11	1.60	3.04	4.50	5.32	6.40	6.87	7.22	6.94			
150				0.63	1.26	1.93	3.90	5.36	6.03	6.77	7.35	7.67	7.57	7.21		
140			0.21	0.78	1.54	2.66	5.04	6.14	6.52	7.24	7.91	8.15	8.10	7.94	7.74	
130			0.39	1.06	2.17	4.33	5.96	6.69	7.24	8.27	8.61	9.02	8.71	8.59	8.41	
120			0.63	1.51	3.33	4.13	5.43	6.98	7.71	8.93	9.43	10.07	9.62	9.27	9.27	8.84
110		0.18	1.00	2.06	2.95	5.16	6.19	7.74	8.70	9.66	10.06	10.61	10.50	10.01	9.84	9.71
100		0.83	0.90	2.39	4.74	6.91	8.02	9.10	11.99	11.99	11.99	11.64	11.34	10.93	10.63	10.67
90		2.13	1.41	7.15	7.93	9.37	11.38	11.99	11.99	11.99	11.99	11.99	11.99	11.99	11.82	11.52
80		6.97	5.35	10.56	10.35	11.98	12.00	11.99	11.99	11.99	11.99	11.99	11.99	11.99	12.00	12.00
70		12.00	12.00	12.00	11.90	12.00	12.00	11.99	11.99	11.99	11.99	11.99	11.99	11.99	12.00	12.00
60		12.00	12.00	12.00	12.00	12.00	12.00	11.99	11.99	11.99	11.99	11.99	11.99	11.99	12.00	12.00
50		12.00	12.00	12.00	12.00	12.00	12.00	11.99	11.99	11.99	11.99	11.99	11.99	11.99	12.00	12.00
40	1.54	0.09	0.17	12.00	12.00	12.00	12.00	11.99	11.99	11.99	11.99	11.99	11.99	11.99	12.00	12.00
30	0.05	0.07	0.14	12.00	12.00	12.00	12.00	11.99	11.99	11.99	11.99	11.99	11.99	11.99	12.00	12.00
20	0.01	0.06	0.10	12.00	12.00	12.00	12.00	11.99	11.99	11.99	11.99	11.99	11.99	11.99	12.00	12.00
10	0.02	0.05	0.09	12.00	12.00	12.00	12.00	11.99	11.99	9.02	9.00	10.62	11.99	11.99	12.00	12.00
5	0.01	0.05	0.09	8.19	12.00	9.98	9.05	9.14	9.70	5.93	3.20	4.04	6.17	7.37	8.61	11.54
2	0.03	0.05	0.09	0.09	0.16	1.62	0.07	0.35	3.89	2.20	0.24	0.48				

Figure 4.3 Wastegate Position

It is assumed that the wastegate position is measured from the control arm and not the waste gate valve itself due to its rotational opening mechanism. This is how the wastegate position was measured on the flow bench by replace the control arm with a threaded bar with a 1mm pitch giving a known lift for each revolution turned. The flow through the waste gate was measured in 0.5mm lift increments. A polynomial equation was then generated to interpolate the flow at every 0.01mm.

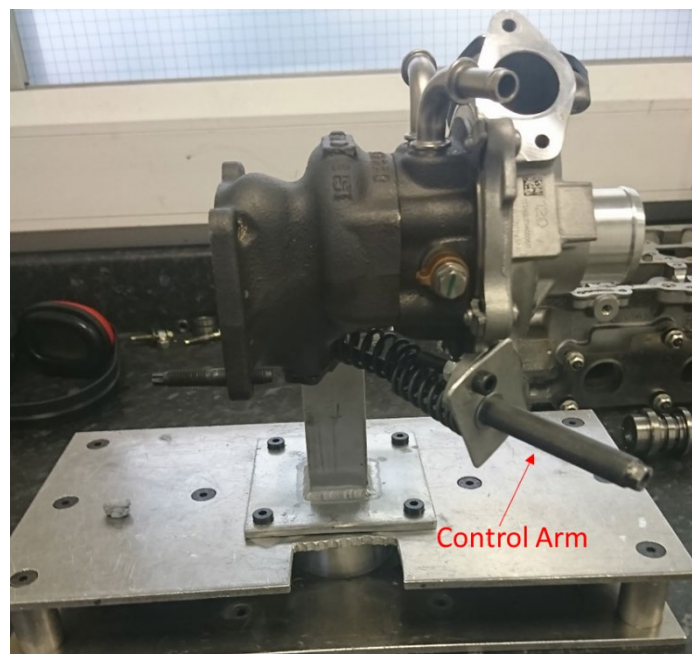


Figure 4.4 Wastegate Control Arm

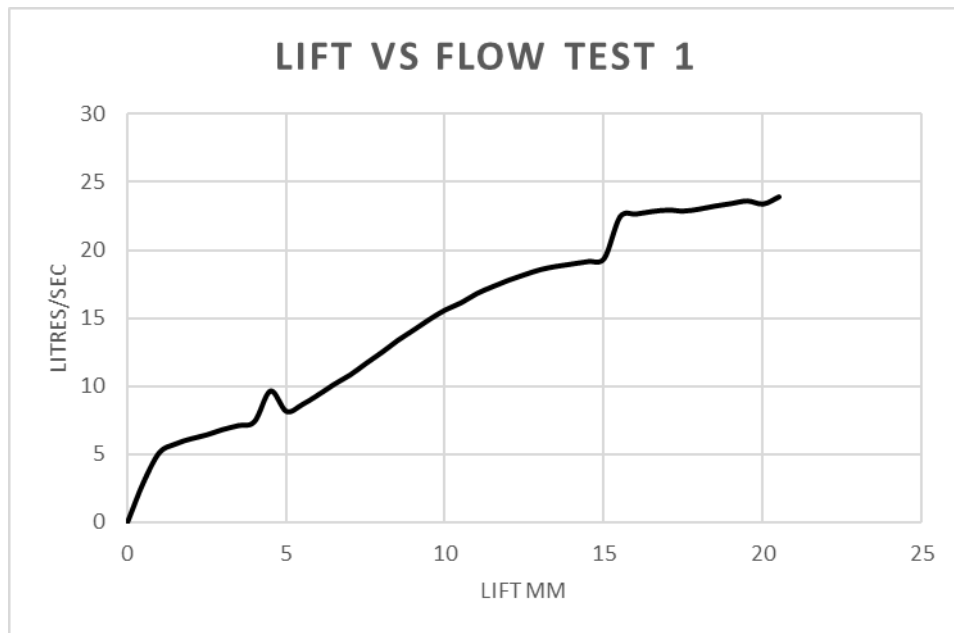


Figure 4.5 First Flow Bench Results

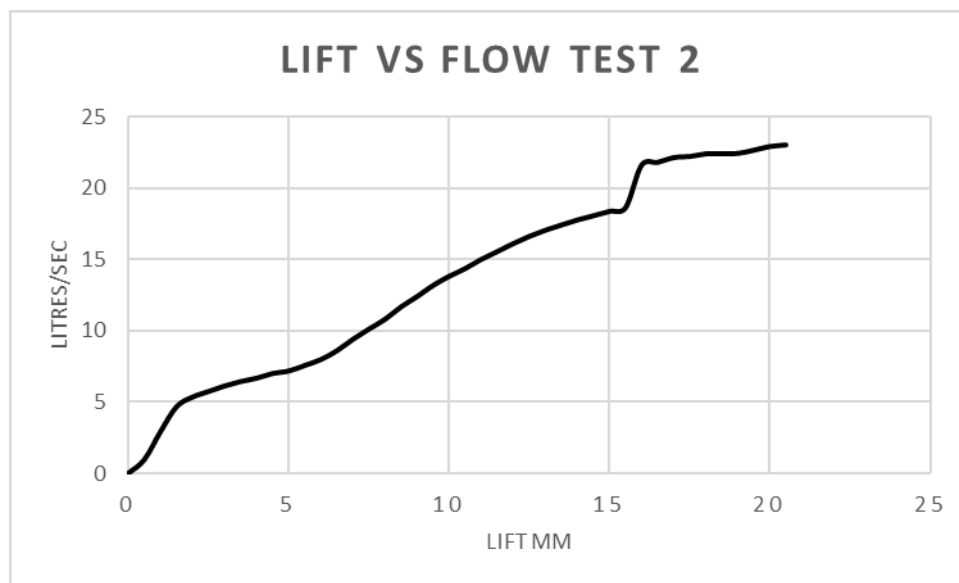


Figure 4.6 Second Flow Bench Results

Two flow tests were carried due to irregularities with the first test. The first irregularity at around 4mm lift did not happen again in the second test this could have been due to a human error in recording the flow percentage. However the jump in flow at around 15mm lift remains. Although they occur at slightly different lift they both share a common theme that it happens when the flow

range was change from 1 to 2 on the flow bench therefore it is deemed at a fault with the flow bench. The flow data obtained can still be used as the maximum wastegate lift from the data is 12mm therefore the irregularity falls out of this range.

The flow data collected from the flow bench now has to be applied to what is happening across the turbocharger. The pressure drop on the flow bench was set to 6229pa and the ambient conditions meant the density was around 1.2 Kg/m³. Across the turbo these conditions are much different with varying pressure differences and temperatures meaning the density will also vary therefore a ratio of what happened on the flow bench to what happens across the turbocharger needs to be created. This can be done by comparing the velocities.

From Bernoulli theorem neglecting the potential energy

$$P_1 + \frac{1}{2}\rho v_1^2 = P_2 + \frac{1}{2}\rho v_2^2$$

Equation 4.11 Bernoulli's equation

By assuming the $v_1 = 0$ the equation can be simplified to

$$P_1 = P_2 + \frac{1}{2}\rho v_2^2$$

The equation can be rearranged for v_2

$$v_2 = \sqrt{\frac{2}{\rho}(P_1 - P_2)}$$

This can then be applied to both the turbocharger and the flow bench

$$\frac{v_2 \text{ Turbo}}{v_2 \text{ Flow Bench}} = \frac{\sqrt{\frac{2}{\rho}(P_1 - P_2)_{\text{Turbo}}}}{\sqrt{\frac{2}{\rho}(P_1 - P_2)_{\text{Flow Bench}}}} = \sqrt{\frac{\rho_{\text{Flow Bench}} \times (P_1 - P_2)_{\text{Turbo}}}{\rho_{\text{Turbo}} \times (P_1 - P_2)_{\text{Flow Bench}}}}$$

The solution can then be applied to the flow rate measure on the flow bench to calculate the flow rate to the through the waste gate in operating conditions.

$$\text{Flow Rate Through Wastegate} = \frac{v_2 \text{ Turbo}}{v_2 \text{ Flow Bench}} \times \text{Flow Rate Measured on Flow Bench}$$

Converting this flow rate to mass flow rate and subtracting from the total mass flow rate will give the actual mass flow rate through the turbocharger.

To convert to Kg/s

$$\dot{m} \text{ Wategate} = \frac{\text{Flow Rate Through Wastegate}}{1000} \times \text{Average Density}$$

Equation 4.12 Wastegate Mass Flow Rate

$$\text{Actual } \dot{m} \text{ Through Turbine} = \dot{m} \text{ Total} - \dot{m} \text{ Wategate}$$

Equation 4.13 Turbine Mass Flow Rate

Actual Flow Through Turbine kg/s																
	800	1000	1250	1500	1750	2000	2250	2500	3000	3500	4000	4500	5000	5500	6000	6500
FL		0.013	0.020	0.027	0.032	0.037	0.037	0.040		0.049	0.056	0.055	0.062	0.064	0.063	0.051
200						0.037	0.037	0.040		0.053						
190						0.035	0.035	0.037	0.044	0.047						
180					0.031	0.032	0.032	0.034	0.040	0.044	0.052					
170				0.027	0.028	0.029	0.029	0.032	0.038	0.042	0.047	0.052				
160				0.025	0.025	0.027	0.027	0.029	0.035	0.038	0.043	0.047	0.055			
150				0.022	0.023	0.024	0.024	0.026	0.031	0.035	0.039	0.042	0.048	0.056		
140			0.020	0.020	0.020	0.021	0.022	0.024	0.028	0.031	0.034	0.037	0.042	0.048	0.055	
130			0.017	0.017	0.017	0.018	0.020	0.021	0.024	0.027	0.030	0.032	0.037	0.042	0.048	
120			0.015	0.014	0.015	0.017	0.018	0.019	0.021	0.024	0.027	0.028	0.032	0.037	0.040	0.046
110		0.013	0.012	0.012	0.014	0.015	0.016	0.016	0.018	0.021	0.024	0.026	0.028	0.033	0.036	0.040
100		0.009	0.011	0.011	0.012	0.012	0.013	0.014	0.016	0.019	0.022	0.023	0.026	0.029	0.033	0.036
90		0.006	0.009	0.007	0.009	0.009	0.010	0.012	0.014	0.018	0.020	0.022	0.025	0.028	0.029	0.032
80		0.003	0.006	0.005	0.007	0.008	0.010	0.011	0.012	0.016	0.018	0.020	0.022	0.025	0.028	0.031
70		0.002	0.003	0.004	0.006	0.007	0.008	0.008	0.010	0.013	0.016	0.017	0.020	0.022	0.024	0.027
60		0.002	0.003	0.003	0.005	0.006	0.007	0.006	0.008	0.011	0.013	0.015	0.017	0.019	0.021	0.023
50		0.002	0.002	0.002	0.004	0.005	0.005	0.004	0.006	0.009	0.011	0.013	0.015	0.017	0.018	0.020
40	0.002	0.005	0.006	0.002	0.003	0.004	0.002	0.002	0.004	0.007	0.009	0.011	0.013	0.014	0.015	0.017
30	0.004	0.004	0.005	0.001	0.002	0.002	0.001	0.002	0.003	0.005	0.006	0.008	0.010	0.011	0.013	0.014
20	0.003	0.003	0.004	0.001	0.001	0.000	0.001	0.001	0.002	0.004	0.004	0.006	0.008	0.008	0.011	0.011
10	0.002	0.002	0.003	0.001	0.000	0.001	0.001	0.002	0.002	0.002	0.003	0.004	0.005	0.006	0.008	0.009
5	0.002	0.002	0.002	0.000	0.000	0.001	0.001	0.001	0.001	0.003	0.004	0.005	0.006	0.006	0.007	0.007
2	0.001	0.002	0.002	0.002	0.003	0.002	0.004	0.004	0.002	0.003	0.007	0.008				

Figure 4.7 Actual Flow through Turbine

Turbine Mass Flow Rate % of Total																
	800	1000	1250	1500	1750	2000	2250	2500	3000	3500	4000	4500	5000	5500	6000	6500
FL		96.88%	95.92%	88.25%	82.69%	78.22%	68.50%	65.43%		63.59%	61.76%	59.81%	60.59%	60.10%	58.70%	51.45%
200						78.26%	68.11%	65.44%		63.32%						
190						77.38%	68.35%	65.15%	64.43%	61.24%						
180					81.86%	76.01%	68.07%	64.55%	63.45%	60.96%	60.56%					
170				87.95%	78.36%	74.46%	66.36%	63.60%	63.31%	61.42%	59.27%	58.82%				
160				86.05%	76.56%	73.00%	64.95%	62.93%	62.47%	59.30%	57.82%	56.93%	57.68%			
150				83.24%	74.75%	70.30%	63.59%	61.81%	61.04%	58.23%	55.97%	54.95%	54.83%	56.72%		
140			94.91%	80.20%	71.92%	66.19%	62.46%	60.20%	59.75%	56.70%	54.01%	52.71%	52.47%	53.74%	54.65%	
130			89.09%	75.49%	66.96%	63.15%	60.87%	58.41%	57.44%	53.05%	51.55%	49.20%	49.76%	51.10%	51.52%	
120			82.56%	69.59%	62.55%	63.07%	61.76%	57.19%	55.33%	50.91%	49.58%	46.88%	47.22%	48.59%	48.28%	49.50%
110		95.62%	74.18%	64.17%	63.72%	61.80%	60.43%	54.79%	51.12%	49.26%	48.38%	46.15%	45.81%	46.76%	47.10%	46.85%
100		76.55%	76.07%	62.69%	61.21%	56.94%	53.60%	50.31%	47.80%	49.53%	49.99%	45.19%	45.26%	45.74%	46.12%	45.27%
90		57.12%	66.66%	48.73%	51.85%	48.32%	47.20%	50.53%	46.95%	51.17%	49.34%	48.65%	48.08%	47.65%	44.03%	44.33%
80		36.84%	51.76%	33.91%	42.81%	44.79%	49.14%	48.67%	44.94%	50.48%	49.59%	48.27%	47.56%	47.80%	47.19%	47.23%
70		21.94%	30.59%	31.97%	39.92%	43.97%	46.93%	40.26%	41.84%	48.95%	48.72%	47.61%	47.09%	47.31%	46.67%	46.76%
60		29.63%	28.51%	30.74%	39.27%	43.33%	43.07%	33.22%	38.63%	47.09%	47.40%	47.95%	46.34%	46.88%	46.34%	45.50%
50		34.55%	22.76%	27.06%	35.60%	42.39%	35.29%	24.40%	33.72%	43.74%	44.74%	47.22%	47.20%	46.82%	44.96%	44.77%
40	51.37%	99.30%	96.54%	24.74%	32.65%	37.18%	18.95%	16.17%	28.95%	40.03%	42.00%	45.04%	46.72%	45.20%	44.78%	43.18%
30	100.00%	100.00%	97.50%	19.12%	29.30%	24.59%	10.38%	16.75%	25.11%	35.85%	37.77%	42.32%	45.16%	43.11%	45.61%	42.60%
20	100.00%	100.00%	98.77%	16.44%	15.53%		14.12%	17.72%	20.27%	30.12%	32.85%	37.98%	42.88%	39.53%	45.94%	42.28%
10	100.00%	100.00%	98.98%	14.93%	-6.31%	21.83%	17.74%	26.98%	24.16%	25.88%	32.80%	31.94%	36.60%	37.13%	46.60%	41.96%
5	100.00%	100.00%	98.89%			13.63%	24.96%	23.35%	25.38%	38.99%	48.11%	50.84%	45.64%	46.04%	45.93%	39.35%
2	100.00%	100.00%	98.73%	98.75%	93.92%	45.08%	100.00%	88.05%	41.13%	48.27%	93.02%	85.41%				

Figure 4.8 Turbine Mass Flow Rate Percentage of Total

In the low load and low RPM region the amount of flow through the turbine is around 15 to 40% showing that large amount is being bypassed through the wastegate.

4.2 Turbocharger Transfer Efficiency

Now the mass flow rate through the turbine has been calculated the power the turbine produces can be calculated from the change in enthalpy and change in kinetic energy across the turbine. This is also applied to the compressor to calculate the power compressor is using.

$$\dot{W} = \dot{m}[(h_{in} + KE_{in}) - (h_{out} + KE_{out})]$$

The kinetic energy converted from joules to Megajoules within the equation to match the enthalpy. This then gives the answer in MJ/S and then multiplying by 10^3 gives the answer in Kw.

Turbine Power Kw																
	800	1000	1250	1500	1750	2000	2250	2500	3000	3500	4000	4500	5000	5500	6000	6500
FL		0.944729	1.530583	2.058763	3.274153	5.304029	5.265665	5.812771		7.669906	8.198026	7.892318	10.02842	10.45055	10.17971	6.899147
200						5.307513	5.091516	5.734698		8.058523						
190						4.673227	4.385407	4.927454	6.224411	6.906245						
180					2.988166	3.9563	3.819776	4.278281	5.19085	5.909497	7.300015					
170				2.050418	1.876874	3.302862	3.168021	3.518172	4.371277	4.824967	5.864791	7.000256				
160				1.784846	1.588298	2.798265	2.618135	2.864708	3.599229	3.970291	4.675495	5.590351	8.1497			
150				1.427354	1.374704	2.149844	2.016273	2.144124	2.655858	3.166775	3.619305	4.33521	5.921301	8.384557		
140			1.49882	1.193557	1.115693	1.509428	1.46036	1.593583	2.052346	2.48847	2.706944	3.174528	4.147804	5.694761	8.106465	
130			1.161939	0.986447	0.85699	1.005473	1.110201	1.193264	1.538527	1.722389	2.033895	2.266108	2.881641	4.108902	6.093515	
120			0.974811	0.868256	0.679619	0.942449	1.080925	1.07495	1.24377	1.366649	1.562897	1.628749	1.998081	2.770742	4.017125	5.648047
110		0.976757	0.819422	0.748872	0.730864	0.856565	1.00259	0.928227	1.000076	1.095242	1.246162	1.26487	1.484327	1.895679	3.119787	3.957974
100		0.634914	0.92153	0.703336	0.668846	0.714113	0.793248	0.72241	0.582027	0.745845	0.751475	0.969788	1.185837	1.40842	1.927911	2.947338
90		0.382671	0.718348	0.362888	0.462687	0.488952	0.518257	0.535062	0.522763	0.716592	0.776106	0.861872	0.952305	0.957455	1.296871	1.766954
80		0.168758	0.39241	0.17645	0.259046	0.29187	0.446755	0.467907	0.483565	0.677076	0.797438	0.869841	0.88998	0.873026	1.037764	1.252145
70		0.076456	0.138361	0.131656	0.168293	0.234651	0.384846	0.332181	0.407807	0.628521	0.739409	0.837302	0.904861	0.920309	0.975933	1.129245
60		0.126124	0.140961	0.126489	0.154551	0.245081	0.329904	0.257535	0.370197	0.579454	0.687811	0.812387	0.894455	0.962756	1.008374	1.150076
50		0.164847	0.118922	0.126884	0.163543	0.24942	0.262155	0.183659	0.326071	0.537339	0.643474	0.788515	0.91306	0.975358	1.036239	1.186779
40	0.288122	0.615893	0.646884	0.13031	0.169198	0.233893	0.150232	0.129919	0.301344	0.489439	0.603935	0.758694	0.913696	0.943289	1.029589	1.140643
30	0.539611	0.574573	0.638837	0.122257	0.184542	0.18508	0.094219	0.160095	0.292557	0.461534	0.575136	0.72927	0.908442	0.929925	1.050486	1.108248
20	0.380035	0.464754	0.595384	0.116651	0.121805	-0.03575	0.142276	0.190444	0.266239	0.423895	0.528161	0.700502	0.905823	0.864108	1.036632	1.043827
10	0.296458	0.357701	0.527447	0.083216	-0.03855	0.146612	0.131419	0.215627	0.251485	0.327376	0.498176	0.522547	0.683454	0.692362	0.916432	0.891533
5	0.151992	0.229381	0.385426	-0.02079	-0.03329	0.107693	0.208686	0.204684	0.259487	0.453215	0.66057	0.761809	0.866195	0.707875	0.785246	0.603082
2	0.059215	0.192699	0.255	0.291031	0.375154	0.309132	0.642545	0.69301	0.427526	0.568415	1.239853	1.226158				

Figure 4.9 Turbine Power

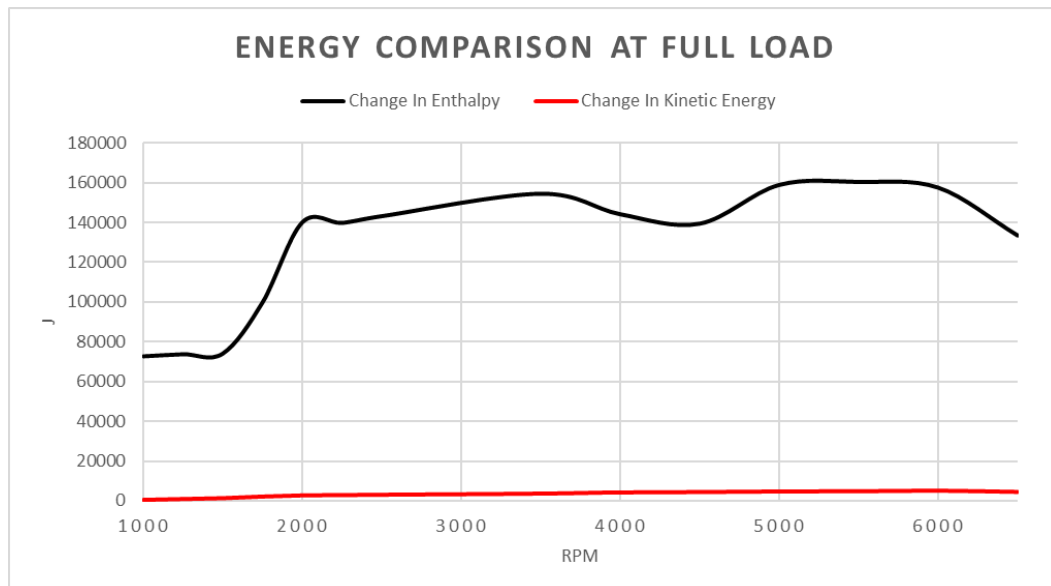


Figure 4.10 Energy Comparison

Change in Kinetic Energy % of Change in Enthalpy																
	800	1000	1250	1500	1750	2000	2250	2500	3000	3500	4000	4500	5000	5500	6000	6500
FL		0.70%	1.29%	1.77%	1.98%	1.71%	1.62%	1.62%		1.79%	2.21%	2.32%	2.24%	2.26%	2.39%	2.16%
200						1.71%	1.64%	1.64%		2.03%						
190						1.79%	1.78%	1.77%	1.97%	1.85%						
180					2.03%	1.84%	1.88%	1.84%	2.05%	2.06%	2.24%					
170				1.76%	2.39%	1.90%	1.95%	2.02%	2.37%	2.52%	2.45%	2.39%				
160				1.75%	2.40%	1.90%	1.98%	2.18%	2.52%	2.61%	2.68%	2.58%	2.14%			
150				1.75%	2.35%	1.96%	2.15%	2.49%	2.90%	2.88%	2.96%	2.82%	2.39%	2.15%		
140			1.31%	1.73%	2.30%	2.03%	2.46%	2.73%	3.11%	3.11%	3.32%	3.17%	2.80%	2.55%	2.18%	
130			1.24%	1.54%	2.07%	2.22%	2.50%	2.73%	3.23%	3.45%	3.63%	3.40%	3.18%	2.87%	2.31%	
120			1.06%	1.16%	1.70%	1.88%	2.08%	2.27%	2.94%	3.41%	3.98%	3.91%	3.68%	3.43%	2.65%	2.29%
110		0.63%	0.76%	0.90%	1.32%	1.55%	1.76%	1.95%	2.49%	3.31%	3.96%	4.24%	4.15%	4.09%	2.97%	2.64%
100		0.40%	0.54%	0.67%	1.04%	1.21%	1.29%	1.63%	3.25%	4.60%	6.94%	4.58%	4.58%	4.85%	4.28%	3.05%
90		0.22%	0.35%	0.49%	0.75%	0.86%	1.11%	1.87%	2.63%	4.07%	4.84%	5.75%	6.44%	7.98%	5.09%	4.55%
80		0.09%	0.22%	0.25%	0.55%	0.84%	1.07%	1.42%	1.84%	3.05%	3.75%	4.27%	5.32%	7.37%	7.24%	7.55%
70		0.03%	0.09%	0.20%	0.49%	0.69%	0.76%	0.77%	1.20%	2.12%	2.82%	3.16%	3.86%	5.09%	5.86%	6.55%
60		0.03%	0.05%	0.12%	0.32%	0.40%	0.44%	0.37%	0.72%	1.39%	1.93%	2.41%	2.73%	3.53%	4.23%	4.42%
50		0.02%	0.02%	0.05%	0.14%	0.23%	0.18%	0.12%	0.34%	0.78%	1.13%	1.62%	2.01%	2.46%	2.69%	2.98%
40	0.01%	0.07%	0.13%	0.02%	0.06%	0.09%	0.03%	0.03%	0.14%	0.41%	0.62%	0.93%	1.30%	1.56%	1.86%	1.96%
30	0.03%	0.04%	0.07%	0.01%	0.02%	0.02%	0.00%	0.01%	0.05%	0.18%	0.27%	0.47%	0.71%	0.86%	1.25%	1.30%
20	0.01%	0.02%	0.03%	0.00%	0.00%	0.00%	0.00%	0.01%	0.01%	0.06%	0.10%	0.19%	0.33%	0.38%	0.73%	0.78%
10	0.01%	0.01%	0.01%	0.00%	0.00%	0.00%	0.00%	0.01%	0.01%	0.02%	0.04%	0.06%	0.12%	0.18%	0.42%	0.46%
5	0.00%	0.01%	0.01%	0.00%	0.00%	0.00%	0.00%	0.00%	0.01%	0.02%	0.06%	0.10%	0.10%	0.20%	0.27%	0.34%
2	0.01%	0.01%	0.01%	0.01%	0.02%	0.00%	0.03%	0.03%	0.01%	0.03%	0.17%	0.20%				

Figure 4.11 Change in Kinetic Energy Percentage of Enthalpy

It can be seen from Figure 4.11 that the change in kinetic energy is mostly only a very small percentage of the enthalpy energy. By looking at the maximum change in velocity across the turbine it can be seen as to why the kinetic energy is so low in comparison to the enthalpy energy.

Maximum Velocity Change = 54 m/s

$$KE = \frac{1}{2}mv^2$$

For 1 Kg of gas

$$KE = \frac{54^2}{2} = 1458J$$

Comparing this to the change in enthalpy that ranges from a low of 30,000J and a peak of 200,000J shows how small the change in kinetic energy actually is.

In order to see how efficient the turbocharger is the work done by the compressor can be calculated and compared. The same process is used.

$$\text{Turbocharger Efficiency} = \frac{\text{Compressor Power}}{\text{Turbine Power}}$$

Equation 4.14 Turbocharger Efficiency

Turbine to Compressor Efficiency																
	800	1000	1250	1500	1750	2000	2250	2500	3000	3500	4000	4500	5000	5500	6000	6500
FL		47%	87%	128%	114%	102%	111%	122%		115%	122%	122%	116%	116%	113%	121%
200						101%	118%	121%		119%						
190						100%	116%	120%	112%	115%						
180					114%	99%	113%	117%	116%	115%	122%					
170				127%	151%	99%	114%	117%	118%	120%	127%	123%				
160				125%	145%	97%	111%	116%	117%	124%	132%	126%	120%			
150				125%	135%	100%	114%	122%	122%	125%	137%	133%	128%	122%		
140			86%	119%	131%	111%	121%	125%	122%	124%	141%	147%	145%	132%	119%	
130			83%	104%	131%	119%	116%	122%	118%	132%	147%	157%	164%	141%	124%	
120			76%	84%	118%	105%	100%	103%	109%	128%	149%	166%	175%	167%	138%	130%
110		43%	63%	74%	89%	85%	80%	88%	101%	119%	142%	168%	178%	186%	142%	140%
100		39%	42%	56%	70%	69%	66%	82%	97%	104%	146%	160%	171%	194%	184%	143%
90		42%	36%	61%	63%	67%	65%	68%	87%	80%	102%	121%	142%	191%	204%	192%
80		62%	41%	88%	83%	81%	56%	65%	78%	68%	83%	95%	123%	170%	178%	183%
70		84%	65%	90%	101%	82%	56%	77%	76%	58%	73%	76%	95%	123%	150%	162%
60		45%	55%	80%	89%	65%	56%	85%	72%	50%	54%	57%	71%	89%	109%	122%
50		34%	64%	74%	74%	57%	57%	97%	66%	42%	43%	43%	50%	64%	78%	87%
40	24%	11%	16%	60%	62%	48%	83%	112%	57%	37%	35%	32%	36%	49%	56%	66%
30	11%	9%	12%	54%	47%	49%	112%	72%	45%	31%	27%	23%	25%	35%	38%	50%
20	11%	8%	9%	44%	55%		60%	49%	38%	26%	22%	17%	17%	27%	25%	37%
10	11%	8%	7%	49%		40%	52%	33%	33%	29%	22%	20%	16%	26%	20%	33%
5	20%	11%	9%			54%	32%	34%	32%	22%	19%	17%	16%	33%	24%	50%
2	56%	13%	12%	15%	15%	20%	12%	12%	23%	20%	14%	15%				

Figure 4.12 Turbine to Compressor Efficiency

It can be seen in Figure 4.12 there are many data points that show an efficiency of over 100%. This is clearly incorrect and impossible to achieve. Efficiencies are not expected to exceed 90% at best due to frictional losses. Initial thoughts were that the change in temperature across the compressor measured is not a true representation of the work done by the compressor. As the turbine and compressor are connected to each other therefor very close this could mean that heat transfer from the hot exhaust gas within the turbine could be heating the air as it passes through the compressor. There could also be a slight increase in temperature by turbulence when the air hits the compressor fins. An outlet temperature can be calculated for an adiabatic reversible process that could be a better representation of the compressor work.

$$T_{2s} = T_1 \left(\frac{P_2}{P_1} \right)^{\frac{\gamma-1}{\gamma}}$$

Equation 4.15 Adiabatic Compressor Temperature

Where

$$\gamma = \frac{\text{Specific Heat at Constant Pressure}}{\text{Specific Heat at Constant Volume}} = \frac{C_p}{C_v} = \sim 1.4 \text{ For Air}$$

Equation 4.16 Ratio of Specific Heats

The adiabatic temperatures are lower than the measured temperatures. This means that the enthalpy would decrease as it is temperature dependant and also the kinetic energy because the density increases with lower temperature and therefor reducing the velocity. Calculating the adiabatic temperatures allows the isentropic efficiency to be defined.

$$\eta_{isen} = \frac{\text{Adiabatic Power}}{\text{Actual Power}} = \frac{T_{2s} - T_1}{T_2 - T_1}$$

Equation 4.17 Compressor Isentropic Efficiency

Compressor Isentropic Efficiency																
	800	1000	1250	1500	1750	2000	2250	2500	3000	3500	4000	4500	5000	5500	6000	6500
FL		61%	69%	71%	70%	70%	72%	73%	75%	76%	77%	77%	76%	74%	72%	70%
200						70%	72%	73%	75%	77%						
190						70%	72%	74%	76%	77%						
180					70%	71%	73%	74%	76%	77%						
170				71%	72%	71%	73%	74%	76%	77%	77%	77%				
160				71%	72%	71%	73%	75%	76%	77%	77%	77%	76%			
150				71%	72%	72%	73%	75%	76%	77%	77%	76%	76%	75%		
140			69%	71%	72%	72%	73%	75%	76%	76%	76%	76%	75%	74%	73%	
130			68%	70%	72%	71%	72%	73%	74%	75%	75%	74%	74%	73%	72%	
120			67%	68%	69%	70%	71%	71%	73%	73%	73%	71%	71%	71%	69%	69%
110		62%	64%	66%	67%	67%	68%	68%	70%	71%	71%	69%	68%	66%	66%	65%
100		55%	60%	61%	63%	62%	62%	62%	53%	59%	60%	65%	64%	64%	63%	61%
90		44%	54%	47%	51%	52%	49%	44%	50%	53%	57%	57%	56%	54%	59%	59%
80		28%	39%	32%	41%	39%	39%	41%	48%	53%	58%	56%	57%	54%	51%	49%
70		7%	13%	20%	30%	34%	34%	33%	44%	49%	56%	55%	57%	55%	53%	51%
60		3%	9%	16%	23%	25%	14%	16%	22%	23%	28%	28%	29%	29%	33%	52%
50		-1%	2%	5%	12%	9%	9%	13%	18%	20%	25%	25%	28%	27%	27%	26%
40	5%	9%	15%	2%	5%	5%	5%	8%	14%	15%	20%	20%	24%	23%	24%	25%
30	3%	5%	9%	-1%	1%	0%	2%	3%	7%	10%	14%	12%	18%	18%	20%	21%
20	-3%	0%	2%	-3%	-3%	-3%	-2%	-1%	-1%	1%	8%	4%	8%	9%	11%	14%
10	-3%	-1%	-1%	-3%	-5%	-4%	-3%	-3%	-4%	0%	7%	2%	1%	0%	3%	3%
5	-3%	-1%	-1%	-3%	-4%	-4%	-3%	-3%	-3%	3%	14%	11%	12%	12%	10%	6%
2	-2%	-1%	-1%	0%	2%	0%	4%	7%	3%	7%	21%	23%				

Figure 4.13 Compressor Isentropic Efficiency

The compressor efficiency does not show anything unusual in the areas that efficiency is over 100% and is in the expected region.

Turbine to Compressor Efficiency With Calculated Temperatures																
	800	1000	1250	1500	1750	2000	2250	2500	3000	3500	4000	4500	5000	5500	6000	6500
FL		29%	61%	92%	80%	72%	81%	91%		89%	95%	95%	88%	87%	82%	86%
200						72%	86%	90%		92%						
190						71%	85%	90%	86%	90%						
180					80%	71%	84%	88%	89%	90%	95%					
170				92%	109%	72%	84%	89%	91%	94%	99%	96%				
160				91%	105%	70%	83%	89%	90%	97%	103%	98%	92%			
150				92%	99%	74%	86%	94%	95%	97%	106%	102%	98%	92%		
140			61%	87%	96%	82%	91%	96%	94%	96%	108%	112%	110%	100%	88%	
130			58%	76%	98%	88%	86%	93%	90%	101%	111%	118%	124%	105%	91%	
120			53%	60%	87%	76%	73%	77%	82%	96%	110%	120%	127%	122%	99%	91%
110		27%	42%	51%	64%	60%	57%	64%	73%	86%	102%	118%	123%	129%	98%	95%
100		22%	27%	36%	48%	47%	44%	57%	58%	64%	93%	106%	114%	130%	124%	93%
90		19%	21%	33%	38%	40%	38%	39%	51%	45%	60%	71%	84%	114%	132%	122%
80		18%	19%	39%	45%	43%	29%	37%	45%	37%	50%	55%	73%	103%	105%	104%
70		9%	16%	32%	48%	40%	27%	39%	41%	30%	43%	42%	56%	75%	91%	95%
60		2%	11%	25%	38%	28%	19%	32%	26%	14%	14%	17%	24%	35%	48%	73%
50		1%	8%	14%	25%	17%	17%	34%	22%	11%	10%	11%	16%	25%	31%	35%
40	8%	1%	3%	11%	17%	13%	24%	35%	17%	8%	6%	6%	10%	18%	22%	28%
30	4%	1%	2%	9%	12%	12%	33%	21%	12%	6%	3%	2%	5%	13%	14%	21%
20	4%	1%	2%	9%	15%		20%	14%	11%	5%	2%	0%	2%	9%	8%	16%
10	5%	1%	1%	14%		14%	21%	11%	11%	8%	4%	2%	2%	10%	6%	15%
5	9%	1%	2%			19%	14%	13%	13%	8%	5%	3%	3%	15%	8%	25%
2	26%	1%	3%	5%	6%	8%	5%	5%	10%	8%	5%	5%				

Figure 4.14 Turbine to Compressor Efficiencies with Calculated Temperatures

The efficiencies with the calculated temperatures are less as expected due to the reduced compressor power however there are still many regions where the efficiency is over 100%. In order to get the enthalpy a temperatures between the data given a polynomial equation was used. There was a possibility that the equation was not accurate enough and therefore inaccurate enthalpies were obtained for the exhaust gas. However fig shows that the error of the polynomial is very small and therefore it can be dismissed as a potential problem.

T (k)	N2	N2 Polynomial	Error
300	0.054	0.145	-9.1%
400	2.973	2.960	1.3%
500	5.92	5.867	5.3%
600	8.905	8.854	5.1%
700	11.942	11.915	2.7%
800	15.046	15.044	0.2%
900	18.222	18.235	-1.3%
1000	21.468	21.482	-1.4%
1100	24.77	24.780	-1.0%
1200	28.118	28.125	-0.7%
1300	31.51	31.512	-0.2%
1400	34.939	34.938	0.1%
1500	38.404	38.400	0.4%

T (k)	CO	CO Polynomial	Error
300	0.054	0.058	-0.4%
400	2.979	2.976	0.3%
500	5.943	5.936	0.7%
600	8.955	8.950	0.5%
700	12.029	12.028	0.1%
800	15.176	15.178	-0.2%
900	18.401	18.399	0.2%
1000	21.697	21.689	0.8%
1100	25.046	25.039	0.7%
1200	28.44	28.438	0.2%
1300	31.874	31.875	-0.1%
1400	35.345	35.342	0.3%
1500	38.847	38.839	0.8%

T (k)	H2O	H2O Polynomial	Error
300	0.062	0.094	-3.2%
400	3.548	3.480	6.8%
500	6.947	6.949	-0.2%
600	10.528	10.519	0.9%
700	14.209	14.202	0.7%
800	18.005	18.008	-0.3%
900	21.93	21.942	-1.2%
1000	25.993	26.005	-1.2%
1100	30.191	30.197	-0.6%
1200	34.518	34.514	0.4%
1300	38.963	38.952	1.1%
1400	43.52	43.504	1.6%
1500	48.181	48.163	1.8%

T (k)	CO2	CO2 Polynomial	Error
300	0.069	0.073	-0.4%
400	4.003	4.005	-0.2%
500	8.301	8.302	-0.1%
600	12.899	12.900	-0.1%
700	17.749	17.749	0.0%
800	22.81	22.805	0.5%
900	28.047	28.032	1.5%
1000	33.425	33.402	2.3%
1100	38.911	38.890	2.1%
1200	44.488	44.477	1.1%
1300	50.149	50.147	0.2%
1400	55.882	55.888	-0.6%
1500	61.681	61.691	-1.0%

T (k)	O2	O2 Polynomial	Error
300	0.054	0.037	1.7%
400	3.031	3.003	2.8%
500	6.097	6.084	1.3%
600	9.254	9.264	-1.0%
700	12.503	12.527	-2.4%
800	15.838	15.862	-2.4%
900	19.25	19.260	-1.0%
1000	22.721	22.713	0.8%
1100	26.232	26.212	2.0%
1200	29.775	29.753	2.2%
1300	33.35	33.331	1.9%
1400	36.955	36.942	1.3%
1500	40.59	40.583	0.7%

Figure 4.15 Polynomial Error

It was then decided to use different test data using exactly the same method with measured compressor temperatures to see if the problem was that the original data was the issue.

	Turbine to Compressor Efficiency																
	900	1000	1250	1500	1750	2000	2250	2500	2750	3000	3500	4000	4500	5000	5500	6000	6500
FL	17%	32%	42%		65%		63%	70%	68%	74%	82%	83%	85%	89%		88%	82%
200								77%	81%	74%							
190							69%	77%	82%	81%							
180					63%		68%	74%	80%	79%	80%						
170				60%	60%		68%	72%	81%	85%	89%	95%					
160				58%	57%		64%	68%	79%	82%	86%	92%	95%				
150				55%	53%	56%	61%	68%	76%	79%	83%	90%	92%	79%			
140				49%	49%	52%	57%	68%	74%	76%	79%	86%	92%	80%	78%		
130				43%	41%	47%	57%	64%	69%	68%	73%	81%	90%	77%	77%		
120			38%	37%	37%	50%	54%	57%	59%	58%	67%	76%	84%	73%	76%	73%	73%
110			30%	33%	38%	43%	44%	45%	49%	53%	61%	68%	79%	90%	73%	71%	72%
100		29%	27%	35%	32%	37%	37%	40%	45%	47%	54%	60%	69%	81%	66%	64%	67%
90		34%	33%	39%	47%	47%	37%	36%	41%	42%	44%	50%	60%	69%	56%	54%	59%
80		42%	42%	37%	41%	42%	33%	32%	37%	38%	38%	43%	49%	56%	51%	44%	46%
70		31%	36%	32%	33%	35%	31%	32%	43%	41%	35%	36%	44%	48%	48%	45%	42%
60		21%	28%	30%	32%	33%	28%	31%	40%	37%	35%	33%	39%	44%	45%	41%	42%
50		16%	18%	33%	31%	30%	25%	34%	34%	30%	30%	28%	33%	38%	43%	38%	39%
40	16%	16%	17%	31%	27%	25%	24%	28%	28%	22%	24%	22%	26%	34%	41%	37%	37%
30	13%	11%	14%	19%	17%	18%	18%	19%	19%	15%	20%	18%	21%	25%	32%	31%	34%
20	11%	11%	11%	14%	13%	14%	13%	12%	11%	11%	14%	15%	16%	15%	21%	18%	21%
10	10%	10%	9%	21%	11%	7%					14%	13%	14%	19%	23%		
5	18%										18%	19%	18%				16%
2			25%			12%											

Figure 4.16 New Data Turbine to Compressor Efficiency

The new data shows much more realistic efficiencies with no sites over 100%.

The turbine and compressor powers were compared from both sets of data. Positive values show that the new data power is more than the original data.

	Compressor Power Difference Kw														
	1000	1250	1500	1750	2000	2250	2500	3000	3500	4000	4500	5000	5500	6000	6500
FL	0.08	-0.27		1.47		0.56	-0.04		0.07	-0.69	0.39	0.75		1.32	1.29
200							0.69								
190						1.63	0.96	1.18							
180				1.29		1.21	0.87	0.95	0.45						
170			0.57	1.00		1.02	0.71	0.58	0.99	1.53					
160			0.52	0.89		0.83	0.64	0.41	0.48	0.91	2.15				
150			0.48	0.63	0.56	0.72	0.57	0.45	0.39	0.97	1.00	1.09			
140			0.30	0.50	0.43	0.66	0.59	0.37	0.20	0.53	0.48	0.72	0.30		
130			0.24	0.24	0.35	0.46	0.43	0.29	0.10	0.04	0.31	0.44	0.46		
120		0.13	0.18	0.13	0.15	0.14	0.24	0.13	-0.05	-0.09	0.06	0.23	0.02	-0.27	-0.23
110		0.07	0.06	-0.03	0.02	0.04	0.00	-0.04	-0.09	-0.18	-0.20	-0.11	0.12	-0.32	-0.48
100	0.16	0.05	0.01	-0.09	-0.05	-0.02	-0.11	0.04	-0.03	-0.01	-0.26	-0.26	-0.13	-0.61	-0.47
90	0.06	-0.01	0.04	-0.08	-0.06	-0.03	-0.03	-0.01	-0.02	-0.10	-0.10	-0.19	-0.12	-0.73	-0.86
80	0.04	0.02	0.05	-0.05	0.00	0.01	-0.01	0.00	-0.01	-0.07	-0.09	-0.17	-0.10	-0.47	-0.66
70	0.05	0.07	0.06	-0.02	0.01	0.02	0.01	0.03	0.01	-0.06	-0.02	-0.10	-0.01	-0.30	-0.49
60	0.06	0.06	0.05	-0.01	0.01	0.01	0.01	0.03	0.06	0.03	0.03	0.01	0.26	0.10	-0.08
50	0.05	0.07	0.04	-0.01	0.00	0.02	0.02	0.03	0.05	0.05	0.08	0.03	0.26	0.13	0.11
40	0.02	0.02	0.04	-0.01	0.01	0.02	0.01	0.01	0.05	0.04	0.08	0.08	0.24	0.14	0.11
30	0.01	0.03	0.03	-0.01	0.01	0.01	0.02	0.01	0.04	0.05	0.07	0.07	0.17	0.12	0.10
20	0.01	0.02	0.02	-0.01	0.00	0.01	0.01	0.02	0.03	0.04	0.06	0.05	0.12	0.07	0.00
10	0.01	0.01	0.02	-0.01	0.00	0.01	0.02	0.03	0.02	0.03	0.05	0.08	0.10		
5									0.03	0.03	0.05				-0.05
2		0.00			-0.01										

Figure 4.17 Compressor Power Difference

Looking at the compressor power shows that although there is only a small difference in power between the two therefore it must mean that the turbine power calculated from the new data must have increased.

Turbine Power Difference Kw															
	1000	1250	1500	1750	2000	2250	2500	3000	3500	4000	4500	5000	5500	6000	6500
FL	0.66	1.01		4.73		4.85	4.21		3.13	2.92	3.74	3.80		4.25	4.70
200							4.15								
190						5.24	4.01	3.77							
180				4.45		4.22	3.63	3.61	3.14						
170			3.23	4.53		3.57	3.20	2.30	2.68	3.45					
160			2.98	4.00		3.19	2.94	1.99	2.18	2.93	4.01				
150			2.72	3.32	2.70	2.92	2.50	1.98	1.97	2.86	2.84	4.93			
140			2.28	2.91	2.53	2.76	2.18	1.67	1.57	2.24	2.27	4.18	4.14		
130			1.98	2.50	2.23	1.90	1.71	1.51	1.44	1.58	1.93	3.72	3.91		
120		1.33	1.57	1.82	1.34	1.18	1.26	1.27	1.10	1.29	1.55	2.98	3.25	3.05	3.90
110		1.12	1.12	0.89	0.86	0.89	0.88	0.77	0.82	0.99	1.07	1.21	2.95	2.55	2.94
100	0.78	0.67	0.45	0.50	0.49	0.56	0.46	0.67	0.57	0.99	0.80	0.88	2.41	2.50	2.47
90	0.27	0.05	0.31	-0.01	0.08	0.29	0.37	0.51	0.51	0.55	0.61	0.62	1.97	2.09	2.37
80	0.17	0.03	0.36	0.15	0.26	0.34	0.43	0.51	0.48	0.53	0.55	0.68	1.70	1.96	2.13
70	0.30	0.30	0.43	0.30	0.33	0.35	0.50	0.41	0.44	0.53	0.50	0.58	1.30	1.48	1.91
60	0.47	0.36	0.40	0.25	0.27	0.36	0.47	0.41	0.40	0.49	0.42	0.53	1.47	1.81	1.92
50	0.50	0.67	0.29	0.20	0.21	0.41	0.38	0.49	0.40	0.48	0.42	0.34	1.04	1.33	1.61
40	-0.07	0.05	0.25	0.19	0.27	0.46	0.44	0.51	0.47	0.54	0.45	0.24	0.75	0.87	1.12
30	-0.01	0.09	0.39	0.27	0.38	0.55	0.54	0.72	0.45	0.55	0.42	0.28	0.61	0.57	0.80
20	0.01	0.08	0.42	0.31	0.61	0.56	0.69	0.84	0.59	0.50	0.46	0.48	0.79	0.75	0.74
10	0.01	0.03	0.19	0.47	0.69				0.53	0.55	0.59	0.29	0.53		
5									0.26	0.17	0.22				0.94
2		-0.12			0.16										

Figure 4.18 Turbine Power Difference

Figure 4.18 shows that the power of the turbine from the new data is quite a bit greater than that of the original data. The difference in temperature drop across the turbine can be looked at to see if this is the reason for increased power.

Temperature Drop Difference															
	1000	1250	1500	1750	2000	2250	2500	3000	3500	4000	4500	5000	5500	6000	6500
FL	31.7	42.4		67.2		39.2	34.3	33.8	19.8	24.9	23.3	23.1		18.1	17.1
200							31	37							
190						44	31.9	28.1							
180				66.6		41.3	30.4	31.3	24.9						
170			66.3	78.6		37.3	30.2	18.2	19.6	20.6					
160			65.3	73.5		36.6	32.1	18.5	15.5	19.1	21.8				
150			64.5	65.6	37.5	41.7	35.6	23	16.1	21.1	15.4	34.3			
140			57.6	61.9	44.9	48.5	36.7	22	14	19	15.2	34.7	28.4		
130			53	58.1	50.8	41.2	33.1	22	16.2	15.6	14.2	36.3	29.8		
120		45.9	45.2	51.9	34.4	27.2	23.1	19.2	12.5	14.3	14.3	33.9	29.6	18.2	18.6
110		38.1	38.8	29.4	22.2	20.6	17.1	10.1	9.4	12.5	12	9.9	33.4	18.3	17.3
100	32	21.7	16.7	17.4	14.7	10.8	8.5	14.3	11.3	21.4	11.1	8.2	32.8	28.6	17.9
90	13.4	-4.2	10.4	-8	-4.8	6	8.8	11.4	9.5	10.4	11	9.5	35.7	29.8	28.5
80	9.6	-5.4	16.5	4.5	10.5	13.3	11.4	11.4	10	10.2	9.7	12.4	33.7	35.1	34.2
70	32.3	21.3	30.4	20.3	20.7	14.1	13.6	11.4	11	11.6	9.1	9.7	27.2	28.1	33.3
60	49.2	28.3	35.7	29.4	21.9	13.4	12.6	12.8	11.7	10.5	8.5	8.3	30.5	34.2	33.7
50	36.1	41.9	32.3	30	22.9	13.9	13.7	17.5	13.4	10.7	8.4	3.5	23.6	25.3	26.3
40	4.4	16.3	29.8	32.6	23.8	14.9	15.2	28.8	16.8	14.5	9	-0.8	16.3	16.9	17.2
30	24.6	18.2	35.5	38.9	31.2	21	16.7	34.1	18.5	11.9	9.1	0.9	13.4	9.5	9.5
20	20.4	26.7	7.3	11.1	11.3	12.6	11.7	14.1	16.6	9.2	4.1	7	22.7	19.8	13.6
10	28.1	29.7	-20.8	35.1	49.4	-7.2	-25.9	-15	-3.8	4.3	9.1	-15.3	12.5	22	
5									-28.3	-24.7	-11.6				31.2
2		-28.9			-34.2										

Figure 4.19 Temperature Drop Difference

Figure 4.19 shows that the temperature drop across the turbine from the new data is greater than that of the old data. This is a reason for the increased power of the turbine and therefore a more realistic efficiency.

	Mass Flow Rate Percentage Difference														
	1000	1250	1500	1750	2000	2250	2500	3000	3500	4000	4500	5000	5500	6000	6500
FL	9.6%	-3.3%		23.4%		28.2%	19.8%		5.0%	-3.9%	5.1%	2.3%		8.8%	19.4%
200							20.6%								
190						31.9%	22.4%	12.3%							
180				23.4%		27.8%	23.6%	12.0%	5.6%						
170			16.4%	25.1%		28.0%	23.4%	8.8%	6.7%	10.1%					
160			17.5%	27.3%		29.3%	24.3%	7.4%	8.6%	10.6%	15.0%				
150			20.3%	28.1%	30.3%	27.6%	19.2%	9.2%	9.6%	13.7%	13.5%	12.1%			
140			22.6%	31.1%	30.0%	26.5%	17.3%	8.9%	9.7%	13.7%	12.6%	11.9%	9.1%		
130			27.3%	35.4%	27.0%	17.0%	14.6%	11.8%	11.6%	10.6%	15.7%	13.5%	12.9%		
120		20.3%	28.5%	28.9%	14.4%	7.1%	15.2%	13.0%	11.6%	10.4%	15.0%	13.0%	13.4%	12.4%	13.7%
110		27.3%	20.9%	8.9%	9.2%	4.9%	9.6%	9.3%	8.6%	6.0%	6.6%	11.1%	12.3%	10.3%	10.9%
100	29.6%	18.9%	-2.4%	-1.9%	-7.2%	3.4%	-5.8%	4.9%	-6.3%	-5.8%	-1.7%	5.9%	7.3%	5.9%	9.0%
90	4.5%	-15.4%	-1.2%	-40.9%	-24.0%	-15.6%	-3.0%	4.2%	-1.4%	-8.6%	-12.3%	-9.9%	-6.4%	1.1%	2.5%
80	-17.7%	-54.7%	24.7%	-40.4%	-21.0%	-22.0%	5.0%	10.2%	-2.7%	-9.3%	-14.4%	-13.2%	-7.4%	-7.4%	-7.6%
70	27.7%	5.4%	23.3%	-26.9%	-34.9%	-6.2%	22.7%	-5.6%	-13.9%	-9.0%	-16.5%	-13.6%	-9.7%	-7.8%	-7.0%
60	46.7%	21.1%	19.5%	-111.5%	-51.8%	12.6%	33.6%	-7.6%	-26.8%	-10.0%	-26.1%	-14.3%	1.5%	5.5%	3.2%
50	59.9%	73.2%	-10.9%	-275.3%	-92.5%	29.9%	11.6%	3.6%	-30.9%	-8.5%	-28.3%	-24.5%	-11.8%	0.7%	7.5%
40	-21.5%	-14.0%	-11.5%	-279.2%	-31.7%	47.8%	40.5%	-0.7%	-16.9%	-5.6%	-26.2%	-31.8%	-22.2%	-16.3%	0.1%
30	-25.8%	-5.9%	46.7%	-63.2%	4.3%	66.3%	53.3%	41.5%	-19.7%	4.6%	-30.4%	-43.6%	-31.6%	-31.1%	-9.7%
20	-14.6%	-8.4%	72.5%	41.9%	111.7%	67.6%	70.8%	70.5%	22.1%	-5.4%	-18.8%	-26.3%	-20.0%	-20.7%	-19.7%
10	-17.6%	-13.5%	66.7%	118.6%	75.2%				48.8%	30.3%	18.3%	11.5%	-1.4%		
5									34.2%	15.3%	3.0%				18.3%
2		-34.3%			48.1%										

Figure 4.20 Mass Flow Rate Difference

Figure 4.20 shows that the mass flow rate of the new data is also higher than that of the old data which will furthermore increase the turbine power.

It is noted that within the new data the pre compressor temperatures are incorrect as shown in Figure 4.21. For the new data calculations the pre compressor temperatures from the original data were used.

PRECOMPT deg C	900	1000	1250	1500	1750	2000	2250	2500	2750	3000	3500	4000	4500	5000	5500	6000	6500
FL	**	**	**		**		**	**	**	**	**	**	**	**	**	**	**
200							**	**	**	**	**	**	**	**	**	**	**
190							**	**	**	**	**	**	**	**	**	**	**
180					**		**	**	**	**	**	**	**	**	**	**	**
170				**	**		**	**	**	**	**	**	**	**	**	**	**
160				**	**		**	**	**	**	**	**	**	**	**	**	**
150				**	**	133.7	**	**	**	**	**	**	**	**	**	**	**
140				**	**	88.6	**	**	**	**	**	**	**	**	**	**	**
130				**	**	124.5	78.1	83.4	**	**	**	**	**	**	**	**	**
120			420.4	**	79.2	72.0	64.4	114.8	**	**	**	**	**	**	**	**	**
110			250.5	**	75.5	70.1	58.6	92.0	128.0	136.9	**	**	**	**	**	**	**
100		**	240.2	**	58.6	62.6	63.8	85.0	104.3	121.6	**	**	**	**	**	**	**
90		**	236.7	**	56.0	62.4	60.0	140.1	**	82.1	223.0	**	413.9	**	**	**	**
80		**	202.8	**	60.4	61.6	80.3	166.1	**	133.2	133.8	273.4	326.4	**	**	**	**
70		**	263.2	360.9	53.2	60.5	100.2	**	**	156.5	**	164.8	286.5	**	**	**	**
60		**	**	405.9	54.1	58.7	51.7	60.6	**	156.6	**	87.3	**	**	**	**	**
50		**	**	**	54.4	57.8	26.1	32.6	**	113.7	179.7	**	**	**	**	**	**
40	**	**	**	**	54.2	56.8	27.1	48.1	**	79.8	**	170.2	**	**	**	**	**
30	**	**	**	**	52.8	52.7	**	45.8	41.5	90.9	**	**	**	**	**	**	**
20	**	**	**	**	51.5	51.5	**	55.7	108.3	153.7	**	**	**	**	**	**	**
10	**	**	**	**	51.2	79.8	92.8	69.5	175.0	**	**	**	**	**	**	**	**
5	**										**	**	**				**
2			**			99.6											

Figure 4.21 Pre Compressor Data Error

A final set of data was obtained from a 1.4l supercharged and turbocharged engine which included full load and part load data. This data contained turbocharger mechanical efficiency. The calculations were then applied to this final set of data and compared to the mechanical efficiency within the data. The kinetic energy was ignored for these calculations as the inlet and outlet areas could not be measured as there was no access to the turbocharger.

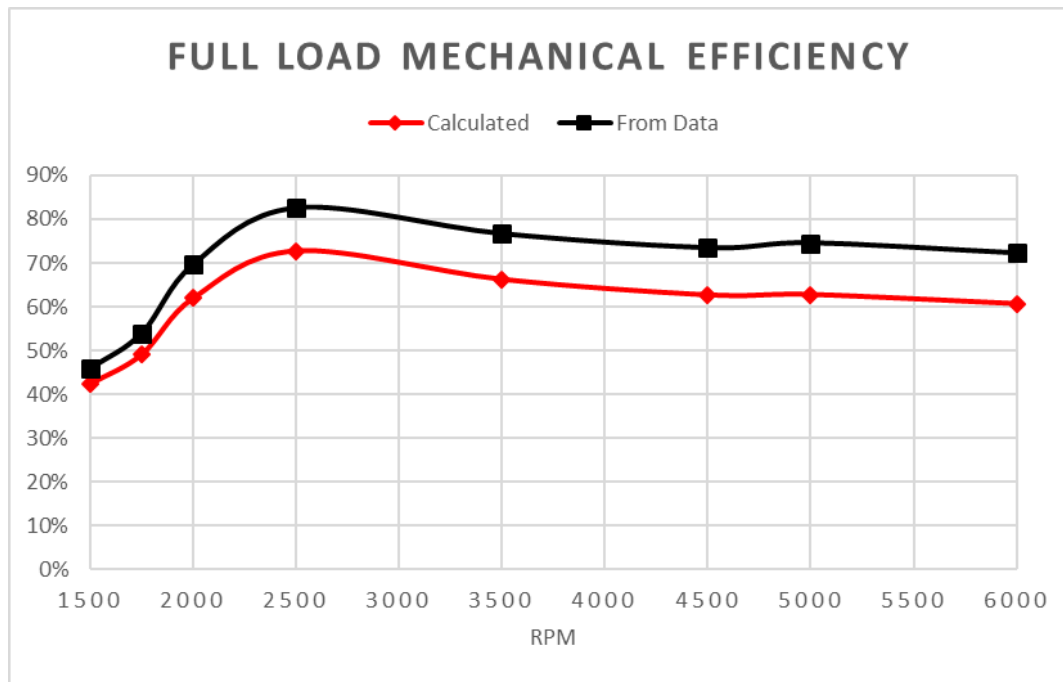


Figure 4.22 Mechanical Efficiency Comparison

Full Load Mechanical Efficiency		
RPM	Calculated	From Data
1500	42.3%	45.8%
1750	49.1%	53.9%
2000	61.9%	69.6%
2500	72.7%	82.5%
3500	66.2%	76.6%
4500	62.7%	73.4%
5000	62.8%	74.5%
6000	60.7%	72.2%

Part Load Mechanical Efficiency		
RPM	Calculated	From Data
2500	38%	42%

Figure 4.22 shows that the mechanical efficiency from the data is slightly higher than the efficiency calculated. This was expected as the actual flow through the turbine was unable to be calculated due to not having the turbocharger to flow bench the wastegate to determine the amount of flow bypassing the turbine through the wastegate, therefore the total mass flow rate was used. However the efficiencies follow the same trend throughout the RPM range.

Having used the method on three other sets of data the author believes that the figures of efficiency over 100% are due to errors in the data and could possibly be down to exhaust gas temperature sensors that are either faulty or have not been calibrated correctly.

Difference in Turbine To Compressor Efficiency																	
	900	1000	1250	1500	1750	2000	2250	2500	2750	3000	3500	4000	4500	5000	5500	6000	6500
FL	0%	1%	1%		2%		2%	2%	2%	2%	1%	2%	1%	1%		2%	3%
200								2%	2%	1%							
190							2%	2%	2%	2%							
180					2%		2%	2%	2%	2%	2%						
170				2%	2%		2%	2%	2%	2%	2%	2%					
160				2%	2%		2%	2%	2%	2%	2%	2%	2%				
150				2%	2%	2%	2%	2%	2%	2%	3%	3%	3%	2%			
140				2%	2%	2%	2%	2%	2%	3%	3%	3%	3%	3%	3%		
130				2%	2%	2%	2%	2%	3%	3%	4%	4%	4%	4%	4%		
120			1%	2%	2%	2%	2%	3%	3%	3%	4%	4%	5%	4%	5%	4%	4%
110			1%	1%	2%	2%	2%	3%	3%	4%	5%	5%	5%	6%	5%	5%	5%
100		1%	1%	1%	2%	2%	2%	3%	4%	4%	5%	5%	6%	7%	6%	7%	6%
90		1%	1%	1%	2%	2%	2%	3%	4%	4%	6%	6%	7%	8%	8%	8%	8%
80		0%	1%	1%	1%	1%	2%	3%	3%	4%	5%	5%	6%	7%	7%	9%	10%
70		0%	0%	1%	1%	1%	2%	2%	2%	2%	4%	5%	5%	6%	6%	7%	9%
60		0%	0%	0%	1%	1%	1%	2%	2%	2%	3%	4%	4%	5%	5%	6%	7%
50		0%	0%	0%	1%	1%	1%	1%	1%	1%	2%	3%	3%	4%	4%	5%	6%
40	0%	0%	0%	0%	0%	0%	1%	1%	1%	1%	1%	2%	3%	3%	3%	4%	5%
30	0%	0%	0%	0%	0%	0%	0%	1%	1%	1%	1%	2%	2%	2%	2%	3%	4%
20	0%	0%	0%	0%	0%	0%	0%	0%	0%	0%	1%	1%	1%	2%	1%	2%	3%
10	0%	0%	0%	0%	0%	0%					0%	0%	1%	1%	1%		
5	0%										0%	0%	0%				1%
2			0%			0%											

Figure 4.23 Difference in Turbine to Compressor Efficiency by Including Kinetic Energy

Figure 4.23 shows that for most of the data sites the difference in transfer efficiency is a very small amount only a 1 or 2% increase. However there are a few sites where it does reach up to 10% which that cannot be considered negligible.

4.3 Increase in Flow Work

Adding a turbocharger to an engine increases the restriction within the exhaust system and raises the pressure and temperature (Huang, 2017). This restriction means that the engine must work harder to push out the exhaust gases resulting in an increase in PMEP. The increase in flow work by adding a turbocharger can be calculated by calculate the flow work either side of the turbocharger. After the turbocharger the conditions similar to what a naturally aspirated engine would be.

$$\text{Flow Work per Second} = \text{Pressure} \times \text{Change in Volume}$$

Equation 4.18 Turbocharger Flow Work

$$\text{Change in Volume} = \frac{\text{Mass Flow Rate}}{\text{Density}}$$

$$\text{Change In Flow Work per Second} = \text{Flow Work In} - \text{Flow Work Out}$$

Change in Flow Work Kw																	
	900	1000	1250	1500	1750	2000	2250	2500	2750	3000	3500	4000	4500	5000	5500	6000	6500
FL	0.15	0.37	0.57		1.73		2.24	2.47	2.83	2.97	3.12	3.28	3.50	4.16		4.52	3.69
200								2.42	2.53	3.00							
190							2.14	2.18	2.28	2.69							
180					1.61		1.85	1.93	2.02	2.40	2.62						
170				1.16	1.39		1.57	1.64	1.69	1.82	2.14	2.69					
160				1.05	1.21		1.36	1.42	1.45	1.54	1.75	2.18	2.81				
150				0.92	1.02	1.07	1.19	1.19	1.21	1.29	1.46	1.86	2.11	3.21			
140				0.77	0.88	0.93	1.04	1.00	1.00	1.04	1.16	1.43	1.61	2.49	2.98		
130				0.66	0.74	0.79	0.80	0.80	0.81	0.85	0.92	1.07	1.26	1.98	2.43		
120			0.52	0.57	0.61	0.61	0.63	0.65	0.67	0.71	0.73	0.86	0.97	1.53	1.84	2.19	2.96
110			0.44	0.48	0.45	0.48	0.54	0.53	0.56	0.53	0.58	0.70	0.75	0.85	1.51	1.78	2.18
100		0.33	0.39	0.34	0.35	0.37	0.41	0.38	0.41	0.39	0.43	0.57	0.59	0.67	1.22	1.42	1.74
90		0.21	0.24	0.22	0.16	0.20	0.28	0.29	0.31	0.32	0.38	0.44	0.50	0.54	0.97	1.12	1.36
80		0.14	0.16	0.18	0.16	0.20	0.27	0.28	0.28	0.31	0.36	0.43	0.48	0.53	0.85	1.00	1.12
70		0.16	0.17	0.19	0.18	0.21	0.24	0.26	0.25	0.28	0.35	0.41	0.46	0.51	0.74	0.82	1.00
60		0.19	0.19	0.19	0.18	0.20	0.22	0.23	0.24	0.28	0.34	0.39	0.43	0.48	0.78	0.90	0.98
50		0.17	0.20	0.18	0.19	0.20	0.22	0.22	0.24	0.30	0.34	0.38	0.42	0.43	0.67	0.77	0.89
40	0.14	0.15	0.18	0.18	0.20	0.21	0.23	0.23	0.25	0.31	0.35	0.39	0.42	0.41	0.59	0.66	0.75
30	0.13	0.69	0.19	0.21	0.22	0.24	0.26	0.26	0.28	0.33	0.34	0.39	0.42	0.43	0.55	0.57	0.65
20	0.50	0.55	0.17	0.96	0.19	0.23	0.25	0.26	0.29	0.29	0.35	0.38	0.42	0.49	0.58	0.60	0.61
10	0.35	0.36	0.14	0.59	0.17	0.20	0.14	0.13	0.15	0.18	0.28	0.34	0.38	0.31	0.41		
5	0.22										0.22	0.25	0.31				0.48
2			0.24			0.11											

Figure 4.24 Change in Flow Work

For the majority of the sites the increase in flow work is low, however at high RPM the flow work does increase to over 4 Kw this will cause back pressure in the system.

5 Exhaust Energy Waste Reduction and Recovery Systems

The efficiency of an engine can be improved by either reducing the amount of exhaust energy wasted or by recovering this energy. The first two systems researched, the miller cycle and ILMOR's 5 Stroke engine work in the basis of reducing the exhaust energy waste by increasing the expansion ratio. The other systems are recovery systems that use the energy to create electricity for charging purposes.

It is believed that if 6% of the exhaust gas energy is able to be recovered then it is possible to reduce the fuel consumption by 10% (Vazquez, et al., 2002)

(Wang, et al., 2013) Show's a method to calculate exhaust energy recovery efficiency with respect to the fuel power. The method uses the exhaust gas power percentage of fuel power and the Chambadal-Novikov efficiency or Curzon-Ahlborn efficiency also referred to as the maximum power cycle in (Kaminski, 2005) which is based on the Carnot cycle efficiency but gives a more realistic efficiency.

The equation for exhaust energy recovery efficiency is

$$\eta_{er} = \eta_{ex} \cdot \eta_{mp}$$

Equation 5.1 Exhaust Energy Recovery Efficiency

Where

$$\eta_{ex} = \frac{\text{Exhaust Power}}{\text{Fuel Power}}$$

And

$$\eta_{mp} = 1 - \sqrt{\frac{T_{cold}}{T_{hot}}}$$

Equation 5.2 Maximum Power Cycle Efficiency

If you have an engine that wastes 33% of its power down the exhaust and the temperature of this gas is around 800°C assuming a naturally aspirated engine and the recovery device is able to drop the temperature of the gas by 200°C the efficiency can be calculated.

$$\eta_{mp} = 1 - \sqrt{\frac{600 + 273}{800 + 273}} = 0.1$$

$$\eta_{er} = 0.33 \times 0.1 = 0.03 = 3\%$$

By dropping the temperature of the exhaust gas by 200°C it shows that it is possible to recover 3% of the fuel power.

5.1 Waste Heat Reduction Systems

The main aim for waste reduction systems is to make the expansion ratio larger than the compression ratio. By doing this the engine can extract more energy from combustion.

5.1.1 Over Expansion Cycles

In a traditional Otto cycle engine the inlet valve closes when the piston is around BDC to maximise the amount of air and fuel in the cylinder. The modern Atkinson and Miller cycles work by leaving the inlet valve open for longer so as the piston is coming up on its compression stroke some air and fuel is then being pushed back out of the cylinder. This reduces the effective compression ratio. The performance is reduced but efficiency is increased. The overexpansion means that the exhaust gas temperatures are lower, an advantage of this is that a closer to stoichiometric fuel equivalence ratio can be used at high loads without damaging the catalytic converter (Zhao, 2017). The difference between the modern Atkinson cycle and the Miller cycle is that the Miller cycle makes use of a supercharger to maintain power. The modern Atkinson cycle is commonly used on hybrid vehicle because the loss of performance is not an issue as efficiency is more important and the electric motor can make up for the loss in engine power.

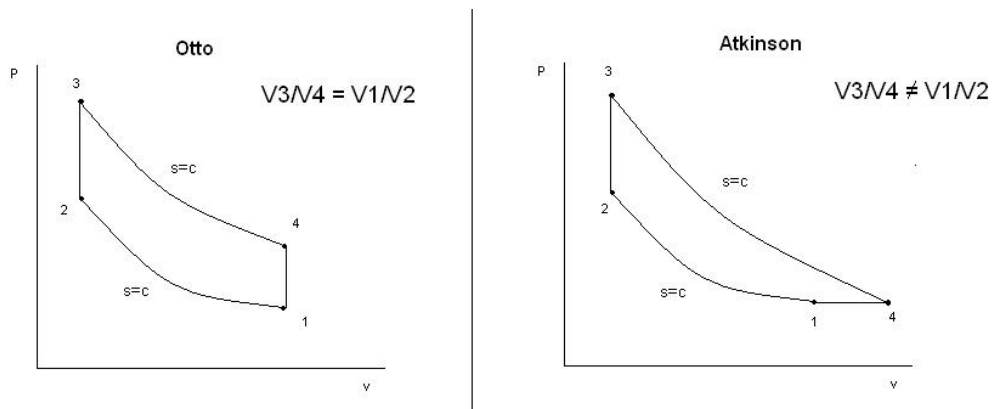


Figure 5.1 Otto Cycle VS Atkinson Cycle

If a recovery system was used on an over expansion engine where the efficiency is higher therefore the exhaust power will be lower and could be around 20% of the fuel power as more has been converted to useful work. This means the temperature of the exhaust gas will be lower too. Assuming the exhaust gas comes out at around 600°C.

$$\eta_{mp} = 1 - \sqrt{\frac{400 + 273}{600 + 273}} = 0.12$$

$$\eta_{er} = 0.2 \times 0.12 = 0.014 = 2.4\%$$

Although the drop in temperature is the same the exhaust power is less therefore the efficiency has reduced.

5.1.2 Ilmor 5 Stroke Engine

The Ilmor 5 stroke engine is a 3 cylinder engine with 2 high pressure firing cylinders and a low pressure expansion cylinder between them. The exhaust ports from each of the firing cylinders feed into the expansion cylinder. The expansion cylinder is timed so that when a firing cylinder is at BDC then the expansion cylinder is at TDC. As the firing cylinder piston pushes the exhaust gas into the expansion cylinder this then pushes the expansion piston down resulting in work. This effectively makes the expansion ratio twice the compression ratio and the exhaust gas will go through two expansions. The optimum configuration for a 5 stroke engine is 2 combustion cylinders and 1 expansion cylinder because this allows the expansion cylinder to operate on a 2 stroke cycle (Melin, et al., 2015). A downside to the 5 stroke engine is that there is now an extra cylinder in the engine that will cause friction, the expansion cylinder in the ILMOR engine has a larger bore, therefore a larger piston and this is extra weight for the engine to move.

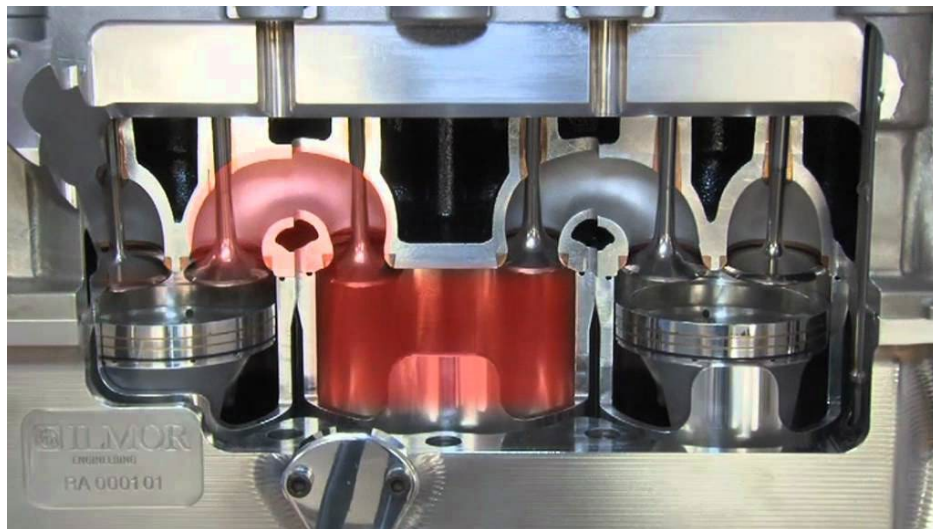


Figure 5.2 Ilmor 5 Stroke Engine

5.2 Energy Recovery Systems

5.2.1 MGU-H and Electric Turbo Compounding

As seen from the turbocharger model the particular turbocharger can generate up to 10 Kw of power. This power could be used to generate electricity to charge a battery on a hybrid vehicle. In formula 1 the turbine and compressor are split and have a device known as a MGU-H between them that can work both ways so that as the turbine is spinning it can store the energy, but then when the turbine speed is low it can act as a motor to spin the turbine faster to reduce turbo lag. On a production vehicle turbine could be added. The disadvantage of adding a turbine is that this creates a restriction in the exhaust system therefore increasing the pumping work the engine has to do.

Electric Turbo Compounding can come in two different configurations series and parallel. In series a second turbine is placed downstream of the main turbocharger. In parallel the main turbocharger turbine is designed produce more power than is needed by the compressor, the excess energy is the extracted by an electric generator coupled to the turbine (Pasini, et al., 2016).

Electric Turbo Compounding is best suited to larger vehicles that mostly drive motorway cycles. This is because at these higher loads it is able to produce more power, and that at low loads the system uses more electricity to keep the turbine spinning fast enough to negate turbo lag (Pasini, et al., 2016) .

5.2.2 Rankine cycle based Recovery

The Rankine cycle based recovery system is similar to the compound turbocharger system where by the power out of the system is from the turbine or expander in this case. However the way the turbine is powered is slightly different. Exhaust gas is used to heat a working fluid to turn to steam that in turn powers the turbine. The advantages of this is that there is no extra restriction in the exhaust system (Huang, 2017). However as seen in Figure 5.3 a pump is needed to move the working fluid around the system. This pump needs to be powered which may come from the engine causing more friction or it could be powered by part of the power generated by the system once it is up and running but this will then reduce the net power output. After the steam has left the expander it then needs to be cooled back down to a liquid and is done so by passing through a condenser. The condenser needs to be supplied with cooling water therefore more power will be needed for this and could increase friction.

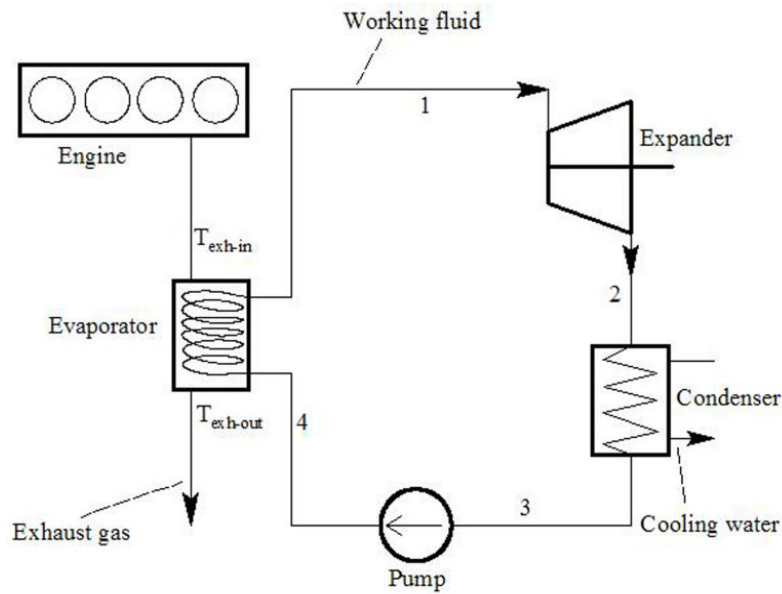


Figure 5.3 Rankine Cycle Example

(Arias, et al., 2006) Conducted a theoretical analysis into three different configurations of a Rankine cycle based waste heat recovery. The tests varied by increasing the temperature of the working fluid between the pump and the evaporator. The first test was a standard baseline test with no additional heat added between these points. The second test used a heat exchanger that exchanged heat from the hot coolant from the engine to the working fluid. The final test eliminated the cooling system from the engine and circulates the working fluid through the engine to use the waste heat to heat the fluid and also using the exhaust gas for the evaporator. Figure 5.4 to Figure 5.6 show the configurations of the three tests.

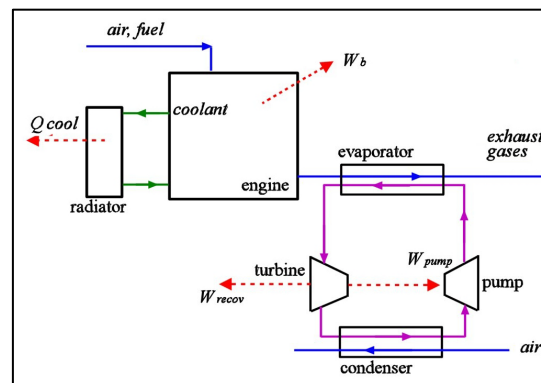


Figure 5.4 First Configuration

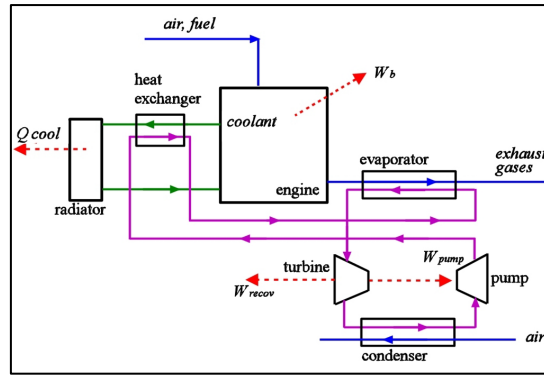


Figure 5.5 Second Configuration

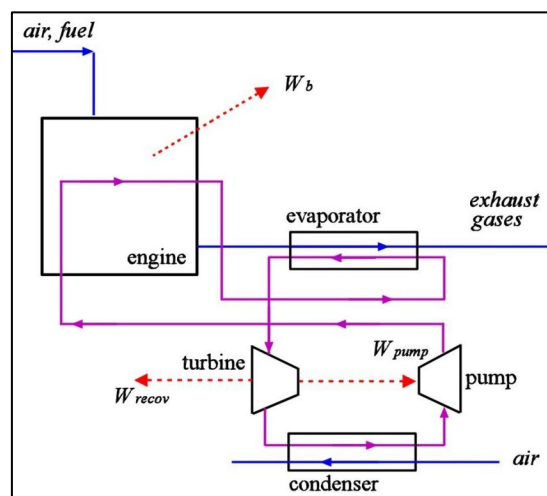


Figure 5.6 Third Configuration

The results show that the third system was most efficient at recovering the waste heat from the engine with managing to recover up to 8% of the wasted heat. This is because it is using the power lost to the engine block as well as the exhaust heat therefore it has more power available.

Rankine cycle based recovery systems tend to have a low efficiency and need lots of space (Huang, 2017). This means that for passenger vehicle use this recovery type may not be practical.

5.2.3 Thermoelectric Generator

Jaguar Land Rover have designed a thermoelectric generator that works by exploiting the seebeck effect. The thermoelectric generator consists of a number of modules that the exhaust gas passes through. The modules contain two semiconductor metals, one positively charged and one negatively charged. As the exhaust gas heats the pair of semiconductors at one end direct electric current is generated (Crosse, 2017).

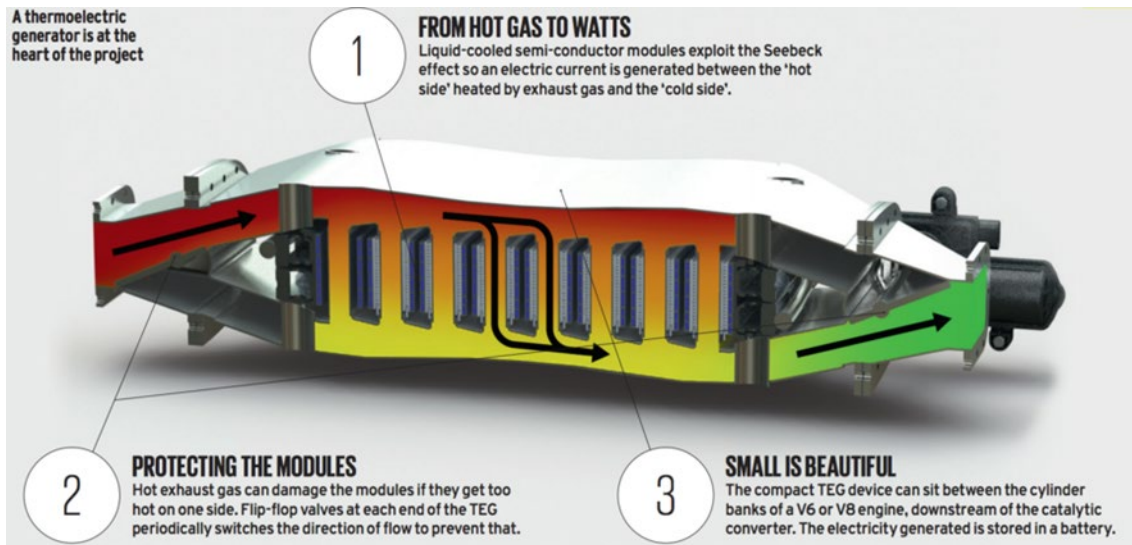


Figure 5.7 JLR Thermoelectric Generator Example

(Vazquez, et al., 2002) Explains how the efficiency of thermoelectric generators are low because the efficiency depends on certain material properties and these are contradictory. The maximum efficiency of thermoelectric material is determined by a dimensionless figure of merit ZT .

$$ZT = \frac{S^2 \sigma}{k} T$$

Equation 5.3 Thermoelectric Efficiency Figure of Merit

$$S = \frac{\text{Open Circuit Voltage}}{\text{Temperature Difference}} = \frac{V}{K}$$

Equation 5.4 Seebeck Coefficient

Where S is the Seebeck coefficient σ is the electrical conductivity, $S^2 \sigma$ is the power factor and k is the thermal conductivity. It can be seen from Equation 5.3 that ideally the material would have a high electrical conductivity to maximise the electricity being generated but also to have a low thermal conductivity to keep the temperature gradient that is key to the process. However materials that tend to be good conductors are also good thermal conductors therefore the efficiency is limited.

An advantage of using a thermoelectric generator as an energy recovery system is that it has no moving parts or no chemical reactions therefore there is little maintenance required due to wear or corrosion (Orr, et al., 2016).

6 Conclusion and Further Work

The friction model in chapter 1 demonstrated how a simplified friction model can get reasonable accuracy in terms of trend and proportions however the friction coefficient is a big factor and it is not easily calculated. It may be a case that the standard coefficient used in simple models of 0.1 should be reduced as more efforts are being taken to reduce friction. It also shows how much of the FMEP is from the piston assembly, although the model over predicted the comparison with real data shows it was still around 40% of the total friction.

Chapter 2 shows that by offsetting the crankshaft the side force reduced greatly. And for the simplified model to friction reduced too. However research suggests the piston secondary motion is effected and this could negate the reduction in side force therefore a more complex model would need to be created.

The thermal efficiency of the Ford Fox Ecoboost engine was calculated in chapter 3 and it shows just how much potential power there is to be recovered in the exhaust.

The turbocharger model showed kinetic energy is only a very small percentage of the enthalpy energy therefore it can be neglected if just looking at either the compressor or turbine. However if the data is available to allow the kinetic energy to be calculated its quite a simple task and if comparing the compressor to turbocharger efficiency then it may have a bigger role. 3 out of the 4 data sets the model was applied to provided reasonable results therefore it was determined that bad data was to blame for the efficiencies over 100%.

Research into systems to either reduce the heat waste or recover the heat waste showed that current ways of recovering the heat are rather inefficient. However with the introduction of more hybrid vehicles the modern Atkinson and miller cycles are starting to be used to increase efficiency. As these cycles mean a lower exhaust gas temperature they may be ideal to use with thermoelectric generators as they will have less of a damaging effect on the materials used.

The further work the author would like to carry would be to combine the efficiency increasing technologies discussed in the paper and calculate the potential efficiency this engine would have.

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8 Appendix

8.1 Calculating Average Enthalpy per Kg

In order to calculate the enthalpy of the exhaust gas at certain temperatures enthalpy of formation tables are used for each of the products of the exhaust gas H_2O , CO_2 , N_2 , O_2 and CO . Plotting these tables in excel allows a polynomial equation to be obtained to calculate the enthalpy formation between the data points.

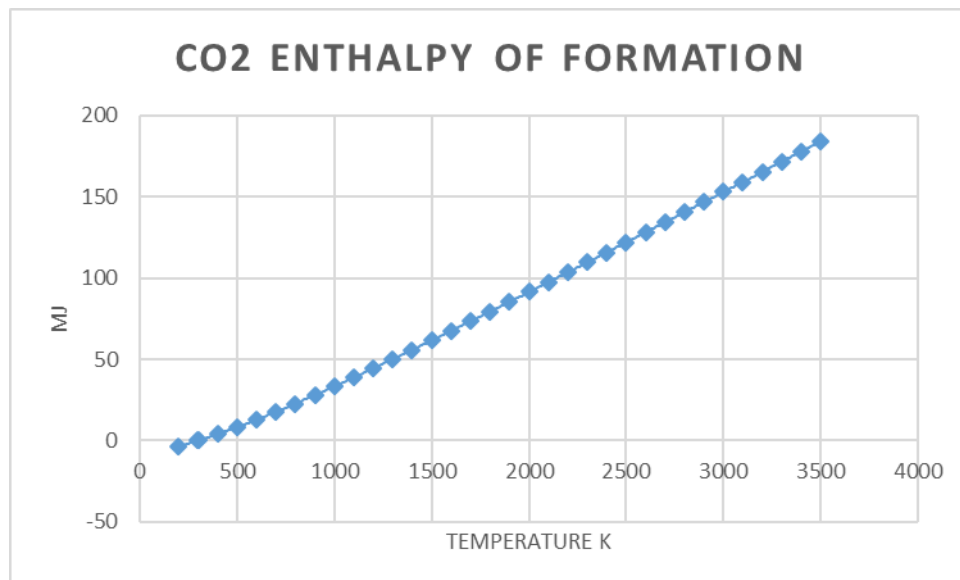
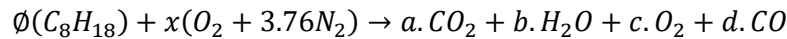


Figure 8.1 CO2 Enthalpy of Formation

$$y = 0.0000000000000000085657339772x^6 - 0.0000000000000001148039281120120x^5 + 0.000000000006379447143132750000x^4 - 0.000000019337110269941800000000x^3 + 0.000035818423405687800000000000x^2 + 0.020409522356917800000000000000x - 8.800283726107930000000000000000$$

The air fuel ratio will determine the number of Kmols products of the exhaust gas.

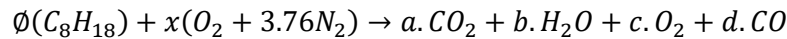


Where ϕ is the fuel equivalence ratio.

The sum of the Kmols multiplied by their enthalpy of formations gives the total enthalpy of formation.

Dividing the total enthalpy by the total number of Kmols gives the average enthalpy per Kmol. Then dividing the average enthalpy per Kmol by the average atomic weight of the products gives the average enthalpy per Kg.

8.2 Calculating Calorific Values



To calculate the calorific values the net enthalpy of formation needs to be calculated.

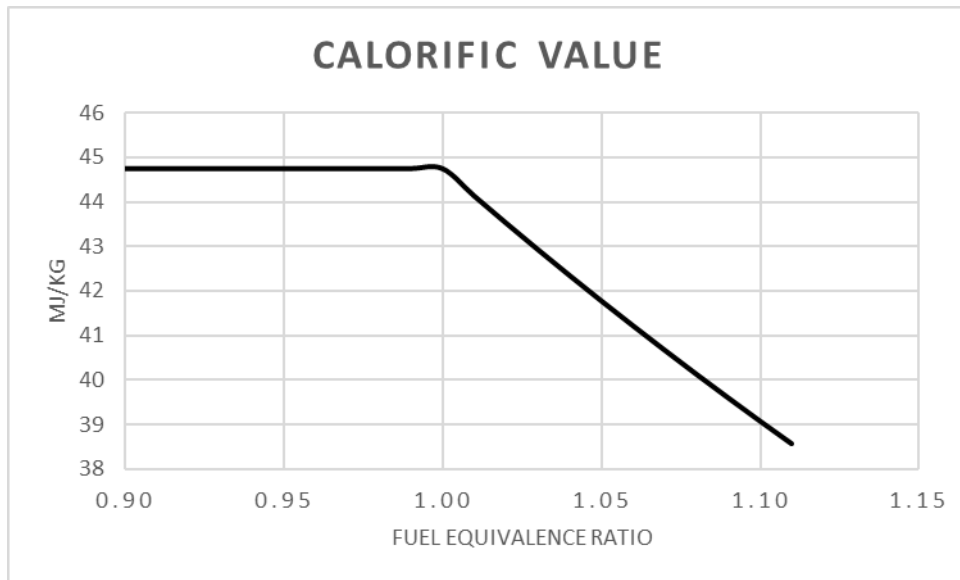
$$\text{Net Enthalpy of Formation} = \text{Enthalpy of Products} - \text{Enthalpy of Reactants}$$

By multiplying the number of Kmols of each products and reactants by its enthalpy formation the net enthalpy of formation can be found.

Enthalpies of formation	MJ/Kmol
O_2	0
N_2	0
H_2	0
C	0
CO_2	-393.5
H_2O	-241.8
C_8H_{18}	-224.1
CO	-110.5

$$\text{Calorific Value} = \frac{\text{Net Enthalpy of Formation}}{\text{Weight of Fuel}}$$

By multiplying the number of carbons and hydrogens in the fuel by their atomic weights give the weight of the fuel.



8.3 Camshaft Lift to Valve Lift Calculation

Due to the tappet clearance and camshaft not contacting directly on the centre of the bucket the valve lift differs from the camshaft lift. In order to calculate the valve lift the highest point of the camshaft needs to be calculated for each degree the camshaft has turned. Then the vertical component X needs to be calculated.

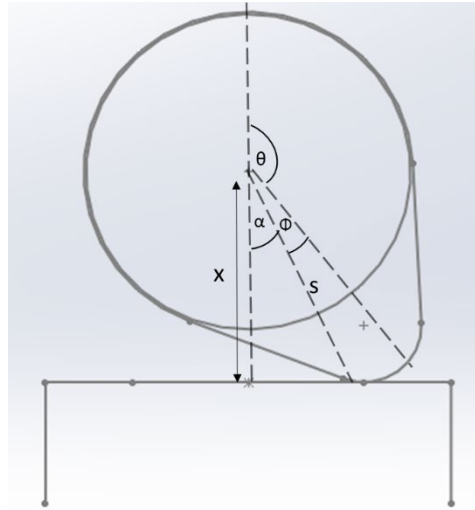


Figure 8.2 Camshaft Lift to Valve Lift

It can be seen that

$$\theta + \phi + \alpha = 180^\circ$$

Therefore

$$\alpha = 180 - (\theta + \phi) \quad (1)$$

And

$$\cos. \alpha = \frac{X}{S}$$

Therefor

$$X = S \times \cos. \alpha \quad (2)$$

By substituting (1) into (2)

$$X = S \times \cos. (180 - (\theta + \phi))$$

S is the base circle radius + camshaft, θ is the angle the camshaft has turned and ϕ is the angle of maximum lift from the camshaft nose.

A 360x360 matrix needs to be created for angles θ and ϕ and the X equation is applied to all cells. The maximum value of ϕ for every angle θ will give the maximum value of X.

Subtracting the base circle radius and tappet clearance from X gives the valve lift

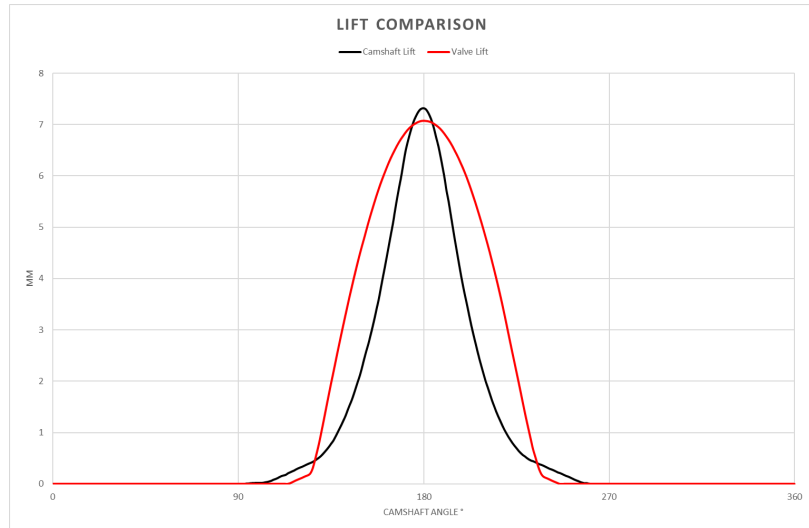


Figure 8.3 Camshaft Lift to Valve Lift

8.4 Calculated Data

8.4.1 Transfer Efficiency

Turbine to Compressor Efficiency Just Enthalpy Energy																	
	900	1000	1250	1500	1750	2000	2250	2500	2750	3000	3500	4000	4500	5000	5500	6000	6500
FL	15%	33%	42%		66%		63%	63%	60%	62%	63%	63%	63%	66%		63%	59%
200							69%	69%	70%	61%							
190							68%	68%	70%	66%							
180					64%		65%	66%	68%	63%	61%						
170				61%	61%		64%	64%	69%	68%	68%	71%					
160				59%	58%		61%	61%	66%	65%	67%	70%	70%				
150				55%	54%	56%	56%	59%	64%	63%	65%	69%	69%	59%			
140				50%	50%	50%	52%	57%	61%	60%	62%	66%	69%	59%	58%		
130				43%	42%	44%	48%	52%	56%	55%	58%	62%	68%	58%	57%		
120			38%	36%	34%	42%	43%	46%	47%	46%	52%	58%	63%	55%	57%	54%	53%
110			31%	29%	31%	35%	35%	35%	39%	41%	47%	51%	58%	67%	54%	52%	52%
100		29%	26%	27%	25%	27%	28%	29%	34%	35%	40%	44%	50%	60%	49%	47%	49%
90		25%	24%	27%	30%	31%	25%	27%	31%	32%	34%	37%	43%	50%	41%	40%	43%
80		24%	26%	26%	25%	28%	22%	24%	28%	29%	29%	32%	35%	40%	38%	32%	34%
70		18%	21%	21%	20%	22%	22%	23%	28%	28%	25%	26%	31%	34%	35%	33%	31%
60		15%	17%	19%	16%	19%	20%	23%	25%	24%	23%	24%	27%	31%	33%	31%	31%
50		15%	17%	18%	14%	16%	18%	20%	21%	19%	19%	20%	22%	26%	30%	28%	29%
40	14%	15%	16%	16%	12%	14%	15%	16%	16%	13%	15%	15%	17%	23%	27%	25%	26%
30	12%	11%	13%	11%	8%	10%	11%	12%	12%	10%	12%	12%	13%	16%	21%	21%	23%
20	10%	10%	10%	11%	7%	8%	9%	9%	9%	9%	9%	9%	10%	10%	13%	13%	14%
10	9%	10%	9%	15%	6%	7%	12%	16%	15%	14%	10%	9%	9%	14%	16%		
5	16%										13%	14%	13%				12%
2			24%			12%											

Figure 8.4 Transfer Efficiency Just Enthalpy Energy

Turbine to Compressor Efficiency Enthalpy Energy with Corrected Mass Flow Rate																	
	900	1000	1250	1500	1750	2000	2250	2500	2750	3000	3500	4000	4500	5000	5500	6000	6500
FL	17%	33%	42%		66%		64%	71%	69%	76%	83%	84%	86%	90%		89%	84%
200								78%	83%	75%							
190							71%	78%	83%	82%							
180					64%		70%	75%	81%	80%	81%						
170				61%	61%		70%	73%	83%	87%	91%	97%					
160				59%	58%		66%	69%	81%	83%	88%	94%	96%				
150				55%	54%	57%	62%	69%	78%	81%	86%	92%	95%	81%			
140				50%	50%	53%	58%	69%	76%	78%	82%	89%	95%	82%	81%		
130				43%	42%	48%	58%	66%	70%	70%	76%	84%	94%	80%	80%		
120			38%	38%	38%	51%	55%	59%	60%	60%	70%	79%	88%	76%	79%	77%	76%
110			31%	33%	39%	44%	45%	46%	51%	55%	64%	72%	83%	95%	77%	74%	76%
100		29%	27%	36%	32%	37%	38%	41%	47%	49%	57%	63%	74%	87%	70%	69%	71%
90		35%	33%	39%	47%	48%	38%	37%	43%	44%	46%	53%	64%	75%	60%	59%	64%
80		42%	43%	38%	42%	43%	34%	33%	38%	39%	40%	45%	53%	60%	55%	48%	51%
70		31%	36%	32%	33%	36%	31%	32%	44%	42%	36%	38%	46%	51%	51%	49%	46%
60		21%	28%	30%	32%	33%	28%	32%	41%	38%	36%	35%	41%	46%	47%	44%	45%
50		17%	18%	33%	31%	31%	25%	35%	35%	30%	30%	29%	35%	40%	44%	40%	42%
40	16%	16%	17%	31%	27%	25%	24%	28%	28%	22%	24%	22%	27%	36%	42%	38%	39%
30	13%	11%	14%	19%	17%	18%	18%	19%	19%	15%	20%	18%	21%	25%	33%	32%	35%
20	11%	11%	11%	14%	13%	14%	13%	12%	11%	11%	14%	15%	16%	15%	21%	19%	22%
10	10%	10%	9%	21%	11%	7%	12%	16%	15%	14%	14%	13%	14%	19%	23%		
5	18%										18%	19%	18%				16%

Figure 8.5 Transfer Efficiency with Corrected Mass Flow Rate

Turbine to Compressor Efficiency																	
	900	1000	1250	1500	1750	2000	2250	2500	2750	3000	3500	4000	4500	5000	5500	6000	6500
FL	17%	32%	42%		65%		63%	70%	68%	74%	82%	83%	85%	89%		88%	82%
200								77%	81%	74%							
190							69%	77%	82%	81%							
180					63%		68%	74%	80%	79%	80%						
170				60%	60%		68%	72%	81%	85%	89%	95%					
160				58%	57%		64%	68%	79%	82%	86%	92%	95%				
150					55%	53%	56%	61%	68%	76%	79%	83%	90%	92%	79%		
140					49%	49%	52%	57%	68%	74%	76%	79%	86%	92%	80%	78%	
130					43%	41%	47%	57%	64%	69%	68%	73%	81%	90%	77%	77%	
120					38%	37%	37%	50%	54%	57%	59%	58%	67%	76%	84%	73%	76%
110					30%	33%	38%	43%	44%	45%	49%	53%	61%	68%	79%	90%	73%
100		29%	27%	35%	32%	37%	37%	40%	45%	47%	54%	60%	69%	81%	66%	64%	67%
90		34%	33%	39%	47%	47%	37%	36%	41%	42%	44%	50%	60%	69%	56%	54%	59%
80		42%	42%	37%	41%	42%	33%	32%	37%	38%	38%	43%	49%	56%	51%	44%	46%
70		31%	36%	32%	33%	35%	31%	32%	43%	41%	35%	36%	44%	48%	48%	45%	42%
60		21%	28%	30%	32%	33%	28%	31%	40%	37%	35%	33%	39%	44%	45%	41%	42%
50		16%	18%	33%	31%	30%	25%	34%	34%	30%	30%	28%	33%	38%	43%	38%	39%
40	16%	16%	17%	31%	27%	25%	24%	28%	28%	22%	24%	22%	26%	34%	41%	37%	37%
30	13%	11%	14%	19%	17%	18%	18%	19%	19%	15%	20%	18%	21%	25%	32%	31%	34%
20	11%	11%	11%	14%	13%	14%	13%	12%	11%	11%	14%	15%	16%	15%	21%	18%	21%
10	10%	10%	9%	21%	11%	7%					14%	13%	14%	19%	23%		
5	18%										18%	19%	18%				
2			25%			12%											16%

Figure 8.6 Turbine to Compressor Efficiency with Kinetic Energy and Corrected Mass Flow Rate

8.4.2 Kinetic Energy

Kinetic Energy Loss Across Turbo Kw																	
	900	1000	1250	1500	1750	2000	2250	2500	2750	3000	3500	4000	4500	5000	5500	6000	6500
FL	0.0003	0.0096	0.0231		0.1327		0.2106	0.1680	0.1847	0.1705	0.1491	0.1636	0.1751	0.2027		0.2266	0.2172
200								0.1643	0.1727	0.1695							
190							0.2006	0.1555	0.1588	0.1510							
180					0.1257		0.1591	0.1457	0.1444	0.1318	0.1338						
170				0.0796	0.1187		0.1415	0.1407	0.1327	0.1206	0.1300	0.1565					
160				0.0702	0.1062		0.1265	0.1235	0.1105	0.1015	0.1191	0.1502	0.1704				
150				0.0582	0.0919	0.1117	0.0997	0.0879	0.0900	0.0879	0.1089	0.1376	0.1498	0.1911			
140				0.0491	0.0778	0.0807	0.0782	0.0654	0.0684	0.0735	0.0938	0.1170	0.1284	0.1592	0.1809		
130				0.0422	0.0650	0.0521	0.0428	0.0466	0.0554	0.0646	0.0767	0.0895	0.1098	0.1361	0.1644		
120			0.0208	0.0288	0.0334	0.0275	0.0279	0.0388	0.0466	0.0539	0.0642	0.0752	0.0903	0.1084	0.1430	0.1556	0.1963
110			0.0177	0.0145	0.0145	0.0196	0.0210	0.0252	0.0357	0.0344	0.0462	0.0534	0.0608	0.0815	0.1107	0.1262	0.1479
100		0.0077	0.0107	0.0050	0.0074	0.0075	0.0122	0.0105	0.0196	0.0220	0.0266	0.0362	0.0400	0.0623	0.0866	0.1022	0.1205
90		0.0011	0.0019	0.0020	0.0013	0.0023	0.0038	0.0092	0.0144	0.0160	0.0272	0.0276	0.0320	0.0423	0.0620	0.0746	0.0948
80		0.0001	0.0003	0.0012	0.0005	0.0014	0.0027	0.0079	0.0111	0.0126	0.0188	0.0219	0.0230	0.0311	0.0505	0.0619	0.0787
70		0.0001	0.0002	0.0006	0.0004	0.0007	0.0025	0.0058	0.0024	0.0041	0.0087	0.0157	0.0158	0.0226	0.0346	0.0447	0.0618
60		0.0002	0.0001	0.0003	0.0001	0.0003	0.0022	0.0034	0.0011	0.0021	0.0037	0.0097	0.0092	0.0154	0.0313	0.0433	0.0518
50		0.0005	0.0009	0.0000	0.0000	0.0001	0.0014	0.0003	0.0007	0.0012	0.0018	0.0056	0.0057	0.0091	0.0156	0.0254	0.0399
40	0.0001	0.0002	0.0005	0.0000	0.0000	0.0001	0.0003	0.0002	0.0003	0.0004	0.0012	0.0031	0.0034	0.0047	0.0073	0.0109	0.0203
30	0.0001	0.0001	0.0004	0.0000	0.0000	0.0000	0.0001	0.0002	0.0004	0.0008	0.0005	0.0018	0.0015	0.0020	0.0033	0.0052	0.0098
20	0.0000	0.0001	0.0002	0.0001	0.0000	0.0000	0.0001	0.0006	0.0009	0.0015	0.0005	0.0004	0.0008	0.0014	0.0019	0.0042	0.0046
10	0.0000	0.0000	0.0001	0.0000	0.0000	0.0003	0.0000	0.0000	0.0000	0.0000	0.0004	0.0006	0.0006	0.0011	0.0012		
5	0.0000										0.0004	0.0006	0.0008				0.0040
2			0.0000			0.0001											

Figure 8.7 Kinetic Energy Loss across Turbine

Kinetic Energy Increase Across Compressor Kw																	
	900	1000	1250	1500	1750	2000	2250	2500	2750	3000	3500	4000	4500	5000	5500	6000	6500
FL	9.38022E-05	0.001227	0.002351		0.001768		0.000944	-0.00227	-0.00197	-0.00129	0.00105	0.007826	-0.01358	-0.01094		0.017509	0.09966
200								-0.00503	-0.00711	-0.00072							
190								-0.00066	-0.00091	-0.00154	0.00043						
180					0.002624		0.00315	0.003411	0.004201	0.008123	0.015193						
170				0.003408	0.003252		0.005007	0.00665	0.009066	0.01059	0.010661	-0.00659					
160				0.00351	0.003998			0.006845	0.008944	0.012264	0.015275	0.021352	0.016601	0.003326			
150					0.003561	0.004495	0.005762	0.007906	0.010457	0.014433	0.018046	0.026696	0.027532	0.03408	0.050287		
140					0.003729	0.004688	0.006106	0.008395	0.011112	0.015293	0.019209	0.029797	0.037179	0.050703	0.071969	0.089325	
130					0.003619	0.004658	0.006316	0.008822	0.011813	0.01586	0.019978	0.031312	0.042143	0.05749	0.080442	0.101301	
120			0.002247	0.003365	0.004638	0.006293	0.008857	0.012025	0.016195	0.020377	0.031015	0.042676	0.061491	0.087935	0.115041	0.128115	0.137311
110			0.001988	0.003009	0.004201	0.005917	0.008341	0.011413	0.015522	0.019399	0.029307	0.040945	0.059787	0.081078	0.116049	0.133668	0.165558
100		0.000989	0.001762	0.002614	0.003647	0.005111	0.007388	0.010077	0.013977	0.017534	0.026271	0.038293	0.055278	0.077141	0.109661	0.134532	0.163115
90		0.000816	0.001334	0.00223	0.003111	0.004184	0.005811	0.007572	0.010845	0.012629	0.020833	0.032395	0.048101	0.068588	0.101318	0.125642	0.155664
80		0.000591	0.000997	0.00148	0.002143	0.002936	0.004017	0.005854	0.007421	0.008992	0.014237	0.023171	0.034118	0.046947	0.076424	0.099583	0.13863
70		0.00035	0.000698	0.001028	0.001626	0.002139	0.002879	0.004407	0.005489	0.006506	0.011244	0.017626	0.025234	0.034473	0.051517	0.070134	0.096973
60		0.000208	0.000456	0.000655	0.000935	0.001346	0.001972	0.002941	0.003718	0.004946	0.008076	0.01266	0.018144	0.027297	0.041671	0.059316	0.074255
50		0.000138	0.000232	0.000422	0.000649	0.000853	0.001344	0.001866	0.002468	0.003189	0.005026	0.008	0.01244	0.016838	0.026765	0.038057	0.052947
40	4.95549E-05	7.21E-05	0.000144	0.000257	0.000381	0.000574	0.000839	0.001104	0.001444	0.00136	0.003188	0.004862	0.007783	0.011197	0.018003	0.024034	0.033099
30	2.85503E-05	2.92E-05	0.0001	0.000143	0.00022	0.000325	0.00045	0.000647	0.000817	0.000765	0.001658	0.002836	0.00423	0.006204	0.009825	0.014103	0.020826
20	1.30867E-05	1.68E-05	3.85E-05	6.34E-05	9.23E-05	0.00015	0.000201	0.000289	0.000387	0.000353	0.000826	0.00143	0.002341	0.003305	0.004841	0.006867	0.00976
10	4.90024E-06	5.08E-06	1.12E-05	1.51E-05	3.38E-05	4.18E-05	5.79E-05	8.73E-05	0.000116	0.000161	0.000361	0.000597	0.000929	0.001111	0.001711		
5	1.94048E-06										0.000243	0.00044	0.000682				0.003157
2			2.44E-06			1.7E-05											

Figure 8.8 Kinetic Energy Increase across Compressor

8.4.3 Flow Work

Flow Work Before Turbine KW																	
	900	1000	1250	1500	1750	2000	2250	2500	2750	3000	3500	4000	4500	5000	5500	6000	6500
FL	1.05	3.44	5.24		13.52		17.72	19.89	21.98	24.18	27.52	29.77	33.63	36.39		40.40	37.17
200								20.91	23.23	24.14							
190							18.22	19.77	22.06	24.27							
180					12.87		16.57	18.26	20.48	22.47	24.94						
170				9.77	11.83		15.29	16.69	18.82	21.03	25.29	30.61					
160				9.06	10.89		13.86	15.14	17.10	18.87	22.60	27.42	32.85				
150				8.23	9.79	10.99	12.61	13.78	15.41	17.07	20.50	25.27	28.57	31.73			
140				7.41	8.85	9.88	11.50	12.69	13.91	15.24	18.12	21.82	25.01	28.30	31.56		
130				6.61	7.81	8.87	10.22	11.35	12.48	13.64	15.99	18.68	22.34	25.12	28.65		
120			4.91	5.88	6.94	8.05	9.13	10.25	11.15	12.22	14.23	16.77	19.56	22.06	25.45	28.58	33.38
110			4.35	5.20	6.15	7.12	8.14	8.89	9.97	10.75	12.76	14.83	17.14	19.90	22.93	25.68	29.20
100		3.11	3.98	4.61	5.38	6.13	7.07	7.70	8.77	9.51	11.12	13.26	15.27	17.76	20.37	23.01	26.17
90		2.70	3.35	4.08	4.67	5.26	6.00	6.57	7.56	8.11	9.84	11.64	13.70	15.72	17.95	20.30	23.14
80		2.31	2.91	3.42	4.03	4.60	5.25	5.96	6.53	7.08	8.53	10.35	12.09	13.75	15.99	17.97	20.43
70		1.87	2.54	2.97	3.66	4.08	4.51	5.25	5.75	6.20	7.55	9.02	10.59	12.24	13.93	15.83	18.06
60		1.50	2.14	2.37	2.77	3.19	3.73	4.39	4.86	5.46	6.64	7.92	9.20	10.73	12.42	14.31	15.79
50		1.22	1.53	1.94	2.37	2.64	3.20	3.67	4.13	4.60	5.55	6.68	8.06	9.10	10.67	12.33	14.02
40	0.82	0.94	1.25	1.58	1.93	2.26	2.66	2.99	3.35	3.31	4.67	5.56	6.77	7.78	9.18	10.45	11.91
30	0.67	0.69	1.11	1.31	1.60	1.87	2.14	2.47	2.72	2.70	3.67	4.56	5.39	6.33	7.37	8.61	10.08
20	0.50	0.55	0.79	0.96	1.17	1.42	1.61	1.86	2.10	2.05	2.88	3.57	4.36	5.10	5.76	6.70	7.65
10	0.35	0.36	0.51	0.59	0.84	0.93	1.07	1.22	1.39	1.58	2.13	2.62	3.14	3.40	3.87		
5	0.22										1.89	2.37	2.82				5.01
2			0.24			0.63											

Figure 8.9 Flow Work before Turbine

Flow Work After Turbine KW																	
	900	1000	1250	1500	1750	2000	2250	2500	2750	3000	3500	4000	4500	5000	5500	6000	6500
FL	0.89	3.07	4.67		11.79		15.48	17.42	19.15	21.21	24.40	26.49	30.14	32.23		35.88	33.49
200								18.50	20.69	21.15							
190							16.08	17.58	19.78	21.58							
180					11.26		14.72	16.33	18.45	20.07	22.32						
170				8.61	10.45		13.72	15.05	17.13	19.21	23.15	27.92					
160				8.02	9.68		12.51	13.72	15.65	17.33	20.85	25.24	30.04				
150				7.31	8.77	9.92	11.42	12.59	14.20	15.78	19.04	23.41	26.46	28.52			
140				6.64	7.97	8.96	10.47	11.69	12.91	14.20	16.96	20.38	23.40	25.81	28.57		
130				5.95	7.07	8.08	9.41	10.55	11.66	12.78	15.08	17.61	21.08	23.14	26.22		
120			4.39	5.31	6.33	7.44	8.50	9.60	10.48	11.51	13.51	15.91	18.59	20.52	23.62	26.39	30.42
110			3.92	4.72	5.70	6.64	7.60	8.36	9.41	10.22	12.18	14.13	16.39	19.04	21.42	23.89	27.02
100		2.79	3.59	4.27	5.03	5.76	6.66	7.32	8.36	9.12	10.69	12.69	14.67	17.09	19.15	21.59	24.43
90		2.49	3.11	3.86	4.51	5.06	5.72	6.29	7.25	7.78	9.46	11.20	13.20	15.19	16.98	19.18	21.78
80		2.17	2.75	3.24	3.88	4.40	4.98	5.68	6.25	6.77	8.17	9.91	11.61	13.22	15.14	16.98	19.31
70		1.71	2.38	2.78	3.48	3.87	4.26	5.00	5.50	5.92	7.19	8.61	10.13	11.74	13.19	15.01	17.06
60		1.31	1.96	2.18	2.59	2.98	3.51	4.16	4.62	5.18	6.30	7.53	8.77	10.24	11.64	13.41	14.81
50		1.04	1.33	1.75	2.18	2.44	2.99	3.44	3.89	4.30	5.21	6.30	7.63	8.67	10.00	11.56	13.13
40	0.68	0.79	1.07	1.40	1.73	2.05	2.44	2.76	3.10	3.01	4.33	5.17	6.35	7.37	8.59	9.80	11.16
30	0.54	0.00	0.92	1.10	1.38	1.63	1.88	2.21	2.44	2.37	3.32	4.18	4.97	5.90	6.82	8.04	9.42
20	0.00	0.00	0.62	0.00	0.98	1.20	1.36	1.59	1.81	1.76	2.53	3.18	3.94	4.61	5.18	6.10	7.04
10	0.00	0.00	0.37	0.00	0.67	0.74	0.92	1.09	1.23	1.40	1.85	2.28	2.76	3.09	3.47		
5	0.00										1.67	2.11	2.52				4.53

Figure 8.10 Flow Work after Turbine

8.5 Engine Specifications

Engine Specifications		
Connecting Rod length	0.1375	m
Stroke	0.082	m
Bore Diameter	0.0719	m
Gudgeon Pin Offset	0.0005	m
Crank offset	0.008	m
cylinder Displacement	0.000333	m ³
Number of Cylinders	3	

Figure 8.11 Engine Internal Specifications

Reciprocating Mass		
Piston	0.22	Kg
Gudgeon pin	0.067	Kg
Circlip	0.00166	kg
Connecting Rod	0.355	Kg

Figure 8.12 Reciprocating Mass

Compression Ring		
Thickness	1	mm
InnerDiameter	68	mm
Inner Area	213.6283	mm^2

Figure 8.13 Compression Ring Specifications

Crankshaft Bearings		
Diameter	46.735	mm
Length	25.165	mm
Clearance	0.023	mm
Surface Area	0.004	m^2
Dynamic Oil Viscosity	11.877	Mpa.s
Number of Bearings	4	

Figure 8.14 Crankshaft Bearing Specifications

Camshaft and Valve Specifications		
Valve Spring	0.02	kg
Collets	0.002	kg
Inlet valve	0.028	kg
Exhaust valve	0.024	kg
Bucket	0.02	kg
Retainer	0.008	kg
Intake Spring Rate	14	N/mm
Initial Compression	11	mm
Exhaust Spring Rate	13	N/mm
Base Circle Radius	15	mm
Intake Lift	7.10	mm
Exhaust Lift	6.79	mm
Number of Lobes per Camshaft	6	

Figure 8.15 Camshaft and Valve Specifications

Oil Pump		
Pressure	3	bar
Flow Rate	30	L/min

Figure 8.16 Oil Pump Specifications

Water Pump		
Pressure	1	bar
Flow Rate	80	L/min

Figure 8.17 Water Pump Specifications