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Design Considerations For Reducing Engine Friction
And Investigation Into Lubrication Regimes.

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Advanced Engine Design

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Major Project

Abstract

This paper will begin by showing the processes involved in developing engine friction models. These models will then be used to analyse the effects of crankshaft offset on engine friction. A suitable engine will then be modified to incorporate the optimum offset calculated from the models. To advance on this a detailed literature review will be carried out into the coefficient of friction and how it changes in respect to speed and load in the hope that an accurate value for the coefficient of friction of certain components can be found and applied to the models.

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Introduction

In today's world the internal combustion engine is becoming subject to much controversy due to the increasing awareness of global warming. Limits are being set every couple of years on the amount of emissions a vehicle can generate with many manufacturers looking for new ways of powering their vehicles. This is leading to the demise of the internal combustion engine with its extinction becoming more and more imminent each year. The reason for this is not only due to its emissions but also its low efficiency. All internal combustion engines are incredibly inefficient at turning fuel into usable energy. The efficiency by which they do so is measured in terms of thermal efficiency with most petrol combustion engines averaging around 20%. Diesels are typically higher with some approaching 40%. For engine efficiency improvement, various approaches include improvements in advanced combustion systems, component system design and handling i.e. down-sizing, boosting, and electrification, as well as waste heat recovery systems etc. Among these approaches, engine friction reduction is a key and relatively low cost approach, which has been receiving significant attention from tribologists and lubrication engineers alike.

The friction in an engine means not all the work transferred to the piston from the gases contained inside the cylinder is available at the wheels. This proportion of the work transferred that is not available to power the vehicle is named the friction work. It is created in a variety of ways within the engine and within the engine's auxiliary components. John B Haywood, Internal combustion engine fundamentals states that the friction work is a sufficiently large fraction of the engine's indicated work and varies between 10% at full load and 100% at no load or idle. Not only does the friction generate wear within the engine but also affects the maximum brake torque and minimum brake specific fuel consumption. John B Haywood also states that often the difference between a good engine and an average engine is the difference in its frictional losses. The frictional work is generated in several ways including the pumping work, overcoming resistance to motion of all the engine's moving parts, and the power required to drive the engine accessories. As a means of reducing this friction the aim of this report is to analyse the benefits of a crankshaft offset and determine the optimum amount of offset for a chosen engine.

To begin the report the friction of several engines will be examined with detailed friction model spreadsheets built initially for the piston skirt. A study will then be conducted into the friction coefficient of engines and how it may change with respect to speed and load. Methods of incorporating this change into the models will then be researched and assessed. Following this the total friction of an engine will be calculated through its torque data which will then be compared with friction models built for each component individually. The amount of error in

the original value used for the coefficient of friction will then become obvious. The coefficient of friction in the models will then be altered until the total matches that of the calculated total engine friction and measured values stated by trusted sources. The models will then be altered to allow for a crankshaft offset. This will then be analysed and discussed until an optimum offset is found. The offset will then be incorporated into a current engine through the use of a Solidworks model. This will allow understanding of the complications involved when re designing the relevant components.

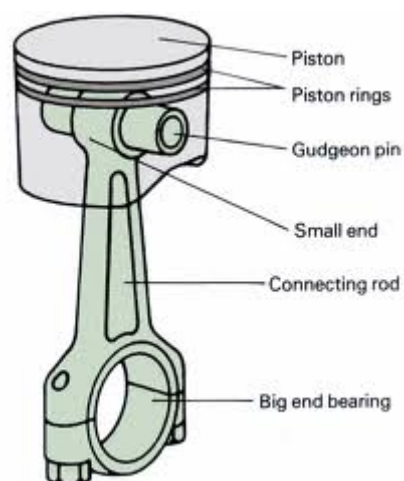
‘This thesis is the result of the author’s original research. It has been composed by the author and has not been previously submitted for examination which has lead to the award of a degree.’

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Chapter 1:

Piston Friction model

The piston assembly is the main source of rubbing friction in an engine. The components that contribute to this friction are the piston rings, piston skirt and piston pin. The forces that act on these components come from the gas forces within the combustion chamber which depend on the engine load, and the inertia forces that depend on engine speed and mass. The piston-assembly contributes to about half of the total friction. The remainder of the friction comes from the crankshaft, connecting rod, camshaft bearings, valvetrain, other oscillating parts and axillary components.



To begin, the piston skirt friction will be calculated with this section detailing the steps taken to produce a friction model. Creating these friction models will give understanding of the amount of friction generated, and how it differs depending on the amount of load and the type of engine i.e. N/A, Turbo, 3 cylinder.

Using acquired data for the cylinder pressures of a Ford 1.6 Sigma, 1.6 Ecoboost and 1.0 litre Fox the bore area of each engine was then measured. From this the gas force at each crank angle is calculated from $F_{\text{gas}} = \text{cyl. Pressure} \times \text{bore area}$. Following this, the inertia at each crank angle is calculated. For this several measurements had to be taken from the engine, including the mass in kilograms of the piston, rings and gudgeon pin; the crank throw (equal

to half the stroke) and the length of the conrod. The angular velocity is also required and is calculated from:

$$\omega = 2\pi \frac{Rpm}{60} \quad (\text{Bosch automotive Handbook pg. 421})$$

The Rpm's used are included in the appendices along with the individual angular velocity values.

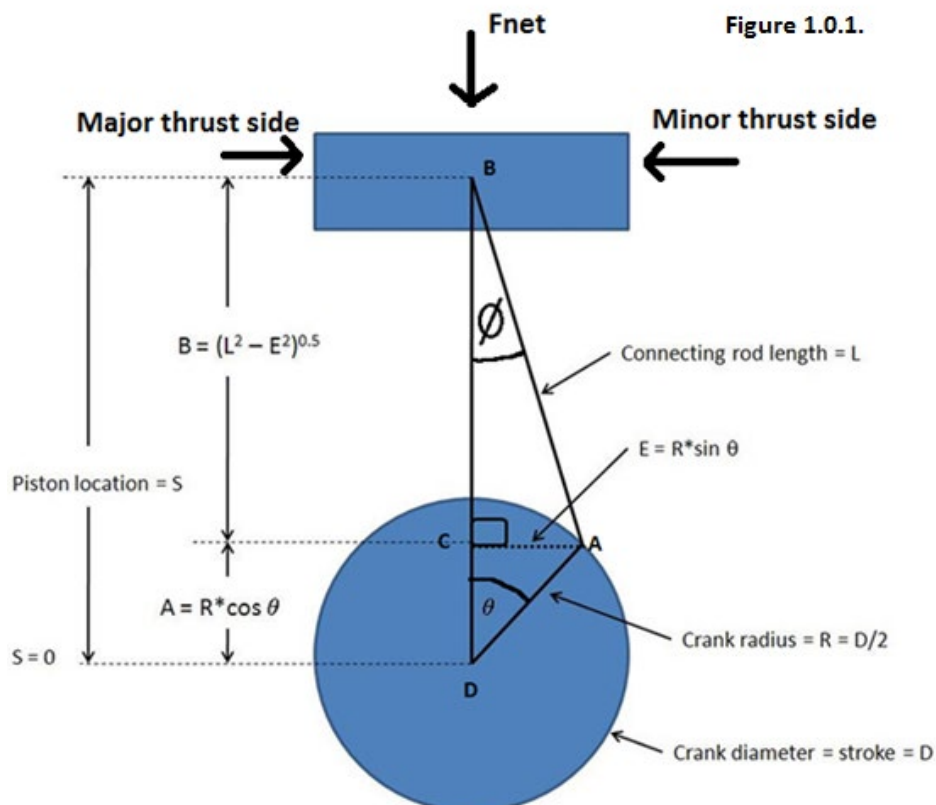
Using the following equation the inertia on the rod at each crank angle (in radians) is calculated.

$$F_{Inertia} = \underbrace{M \cdot R \omega^2 \cdot \cos \theta}_{1st \text{ Order}} + \underbrace{\frac{R}{L} \cdot M \cdot R \omega^2 \cdot \cos 2\theta}_{2nd \text{ Order}}$$

	Piston mass	R	L
1.6 Ecoboost	0.34574	0.04	0.134
Sigma	0.3194	0.04	0.137
1.0 Ecoboost	0.279	0.041	0.137

Fnet or the maximum net downforce is then found from Fgas – Finertia.

Now the angle of the conrod needs to be calculated 'ø'. Figure 1.0.1. illustrates how this is done.



DC is adjacent of the right-angled triangle DAC

CB is adjacent of right angled triangle ABC.

Therefore, looking at triangle ABC:

$AC = L \cdot \sin\phi$ and then triangle DAC:

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

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

$$\sin\phi = \frac{R}{L} \cdot \sin\theta$$

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

Rod reaction is along the length of the rod only because the end bearings are slippery. The side force acting on the piston skirt is a result of the rod angle.

Resolving vertically – $F_{net} = F_{rod} \cdot \cos\phi$ - 1

Resolving horizontally- $F_{side} = F_{rod} \cdot \sin\phi$ - 2

Dividing equation 2 by equation 1 gives:

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

F_{rod} cancels so we are left with:

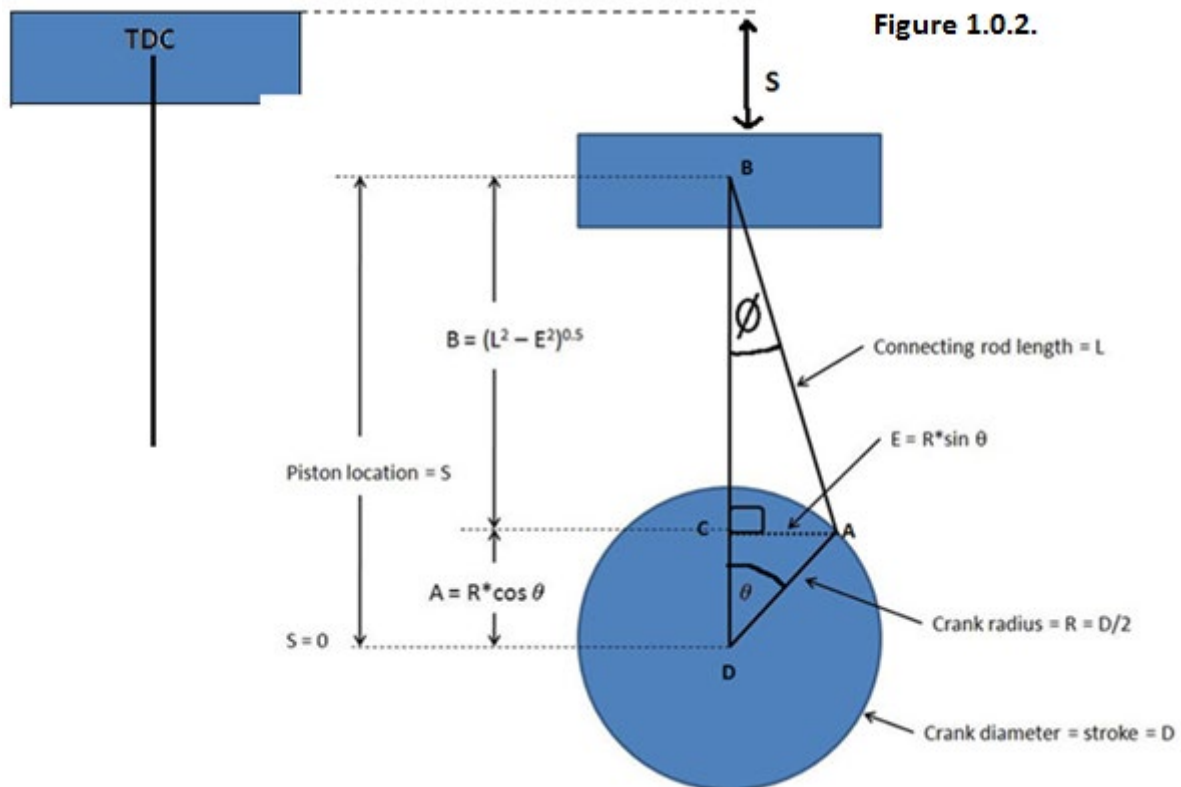
$$\frac{F_{side}}{F_{net}} = \frac{\sin\phi}{\cos\phi} = \tan\phi$$

So $F_{side} = F_{net} \cdot \tan\phi$ and $F_{rod} = \frac{F_{net}}{\cos\phi}$

The friction force is then at right angles to the side force so $F_{friction} = \mu \times F_{side}$

Initially μ will be given a value of 0.1. This will then be subject to further investigation in Chapter 2. The work done per degree due to friction can then be calculated seeing as work done is equal to a force multiplied by a distance.

The distance 'S' or piston travel is found using the triangles in figure 1.0.2.



$$S = R + L - (DB)$$

$$S = R + L - (DC + CB)$$

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

Now that this distance is known the work done can be calculated.

Finding the sum of these values across 720 degrees of crank rotation can then be used to find the FMEP of the piston skirt by dividing the sum of the work done by the engine displacement in cubic metres and then multiplying by the number of cylinders, however the values for work done may yield negative results. In this case the values must be squared and then square rooted to obtain the value as a positive number.

$$\sqrt{W \cdot D^2}$$

For the 1.6 Ecoboost and sigma the engine displacement is equal to 0.0016 m^3 and the Fox 0.001 m^3 . Multiplying by the number of cylinders then gives the total piston friction. Dividing the calculated value for FMEP by 10^5 converts it to Bar.

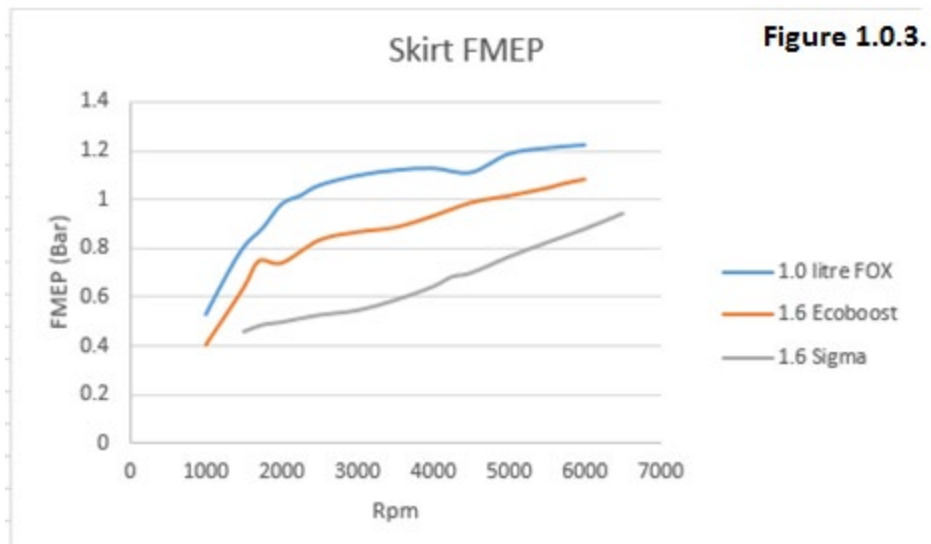
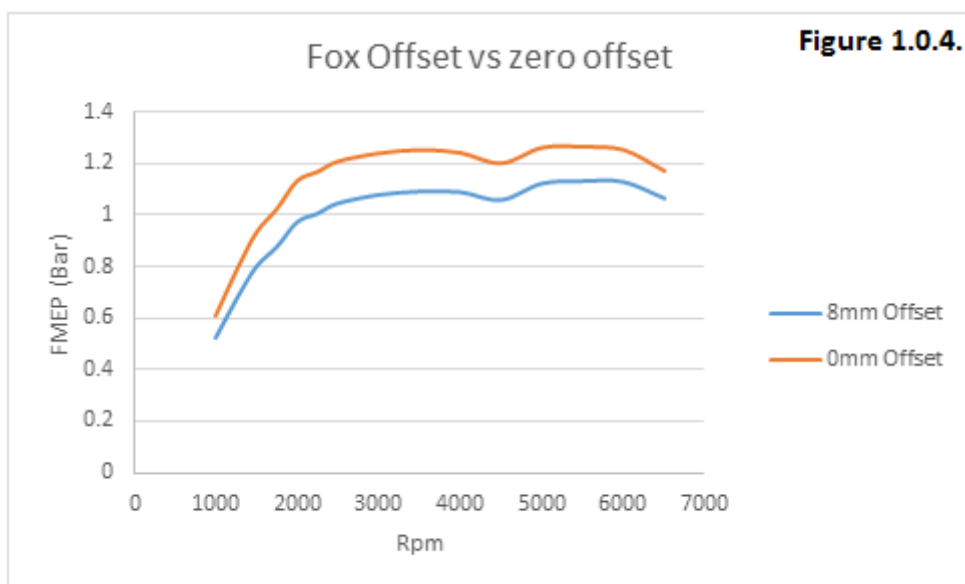


Figure 1.0.3. shows the skirt FMEP for all three engines. It is clear that the turbocharged engines generate a greater skirt friction due to the higher cylinder pressures. The 1.0 litre Fox has greater maximum cylinder pressures and lower inertia due to the lower piston mass than the 1.6 Ecoboost giving it the highest friction mean effective pressure. The 1.0 litre Fox also uses a crank offset of 8mm which reduces the FMEP. Without this skirt friction would be even greater. (figure 1.0.4)

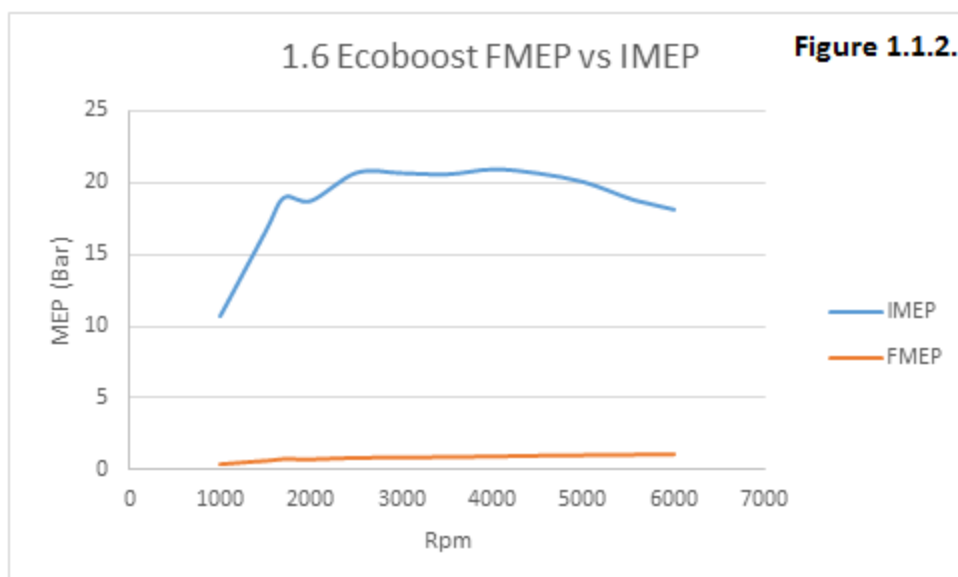
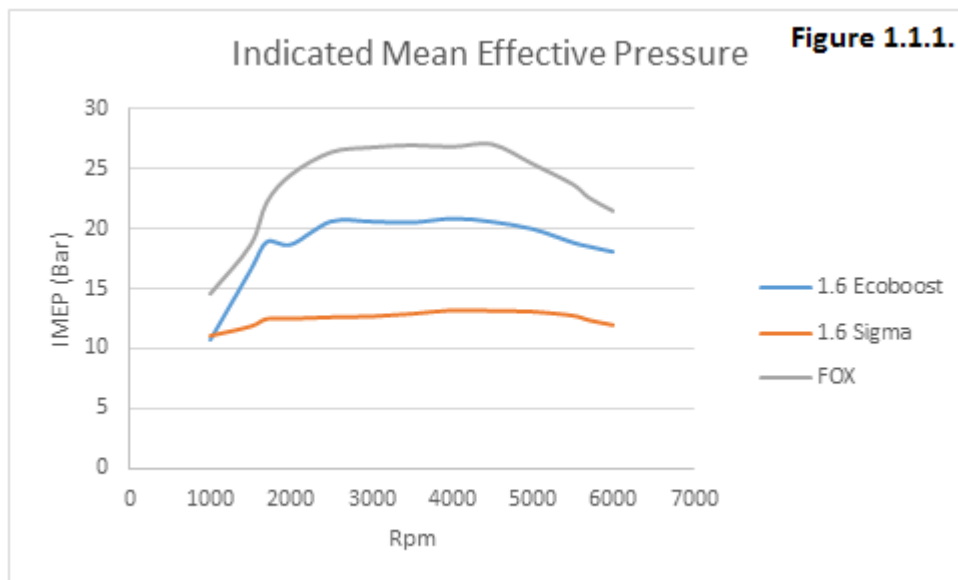


1.1. Indicated mean effective pressure (IMEP)

This is the work delivered to the piston over the entire four strokes of the cycle per unit displaced volume and is equal to the entire engine friction mean effective pressure subtracted from the engine brake mean effective pressure. From the pressure data the cylinder volume in cubic centimetres at each crank angle is also given. Converting these values to cubic metres the work done on the piston per degree can be calculated.

Work done on the piston = *Change in pressure (Psi) x Change in volume (m²)*

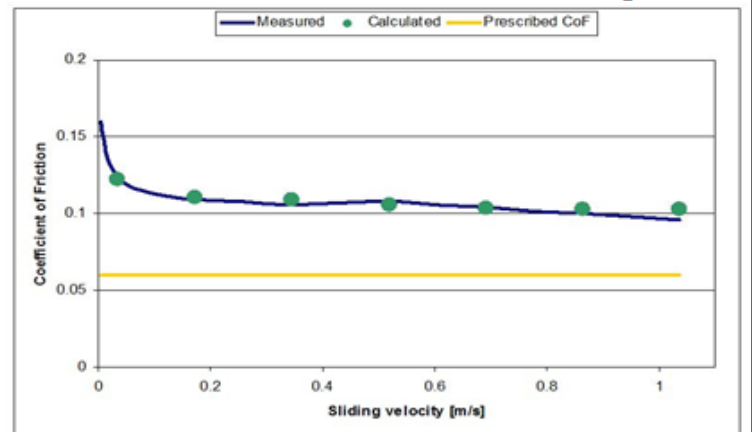
Obtaining the sum of these values allows the IMEP to be calculated just like the FMEP. This time values are not corrected to become positive. (note that by calculating the IMEP in this manner the pumping work friction is taken into account and effectively reduces the IMEP) figure 1.1.2. shows the Piston skirt FMEP as a proportion of the IMEP for the 1.6 Ecoboost engine.



Chapter 2:

Coefficient of friction

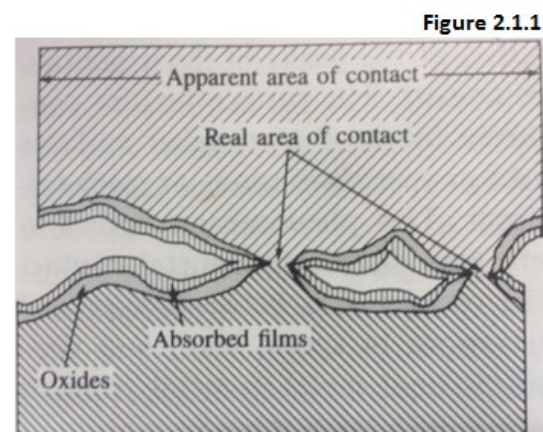
During the process of building a piston skirt friction model a value had to be chosen to represent the coefficient of friction. In the model a single value is used throughout the piston speed and engine speed range, however the following literature review explains how in reality this would not be the case.



The main problem in understanding lubricated friction between surfaces in engines is the wide variation in the magnitude of the forces and velocities involved. This leads to various regimes of lubrication. Figure 2.0.1. shows how as piston velocity increases the coefficient of friction reduces (Phil Carden, Calculation of Friction in High Performance Engines, 2010)

Advances in Powertrain Tribology, SAE

2.1. Boundary lubrication



This type of lubrication is close to solid friction where the lubrication film is only a few molecular layers thick and cannot prevent metal to metal contact. Under this type of lubrication, the friction between two surfaces in relative motion is determined by not only the lubricant properties but also the surface properties of the materials: these being the roughness, hardness, plasticity, elasticity, shearing strength, thermal conductivity and wettability. The lubricant properties are surface ones or chemical ones which dictate the ability of the lubricant molecules to attach themselves to the solid surfaces. Figure 2.1.1. from (John B. Haywood, Internal Combustion Engine Fundamentals, Pg. 717) shows two surfaces under boundary lubrication and how the real contact area is much less than the apparent area of contact. John B. Haywood also states that under boundary lubrication conditions the coefficient of friction is independent of speed and occurs between engine parts such as bearings, pistons and rings, and during normal running at the piston ring/ cylinder interface at top and bottom dead centre crank positions where the piston speeds diminish to zero. It can also occur between heavily loaded parts.

2.2. Hydrodynamic lubrication

This type of lubrication occurs when the shape and relative motion of the sliding surfaces form a film in which there is sufficient pressure to keep the surfaces separated. Unlike boundary lubrication, resistance to motion occurs from the shear forces within the film. Hydrodynamic lubrication is independent of the material or roughness of the parts with the viscosity of the lubricant being the only lubricant property involved. This type of lubrication is formed between two converging surfaces moving at relatively high speed in relation to each other and withstanding a limited load. Hydrodynamic lubrication occurs between engine bearings, piston skirts, cylinder liners and piston rings. (John B. Haywood, Internal Combustion Engine Fundamentals, Pg. 717)

Some of the components of hydrodynamic friction will be dependent on mean piston speed rather than rotational speed. For conventional geometries, a crankshaft's rotational speed is usually used to scale the total friction data rather than mean piston speed although more detailed models must include both. (SAE International, Advances in Powertrain Tribology, 2000)

Hydro elastic and elasto-hydrodynamic are other forms of hydrodynamic lubrication. Hydro elastic occurs when a fluid is pressurized between two surfaces to create a film and so does not require the two surfaces to be in relative motion to one another. Elasto-hydrodynamic lubrication occurs between two surfaces that are in rolling contact such as gears or rolling bearings.

2.3. Mixed Lubrication

This occurs when the hydrodynamic lubrication breaks down due to the film becoming about the same thickness as the height of the surface asperities. To the hydrodynamic friction is then added metal to metal solid friction at the peaks of the asperities and so in a sense both hydrodynamic and boundary lubrication co-exist. The roughness of the surface controls this transition with rougher surfaces making this transition at lower speeds. Sudden variations in load and speed may also cause it to occur. (Advances in Tribology, SAE International, 2000) states that Mixed lubrication is commonly found at the beginning and end of the stroke (first and last 30 degrees).

While cylinder friction varies with speed and load, results show that it represents only 20-30% of overall engine friction. This is contrary to literature findings which suggest cylinder friction is responsible for 60-75% of overall engine friction. (SAE International, Advances in Powertrain Tribology, 2000, Pg. 7)

2.4. Literature review of Stribeck curve and Duty Parameter

It seems a common method of illustrating these different regimes of friction is through a Stribeck diagram (fig 2.4.1.), whereby for a journal bearing the coefficient of friction is plotted against a dimensionless duty parameter equal to the viscosity x shaft speed / loading force per unit area. Initially it was thought that this could be applied to the piston skirt in order to obtain a varying coefficient of friction for the friction model. However, it seems while all papers and books use the same Stribeck curve, none of them give details on how to mathematically relate it to one's own work and when the duty parameter is created within the model, values do not correspond with what would be expected. The next section discusses relevant papers and books on the matter and also demonstrates why the duty parameter does not correspond with what is expected.

- Phil Carden, Calculation of Friction in High Performance Engines, Ricardo UK, 2010

Initially this article seemed ideally suited to the subject with a Stribeck curve showing duty parameter against coefficient of friction for a motored reciprocating test figure 2.4.1. Data for the duty parameter was already known so it was calculated for each crankshaft degree thus giving a value for the friction coefficient at each crankshaft degree. However once completed a duty parameter no larger than 1 was achieved. The calculation below shows how at relatively average piston speeds and loads the duty parameter calculates to be excessively low.

Dynamic viscosity = 0.011619 Pas.s

Velocity = 8 m/s

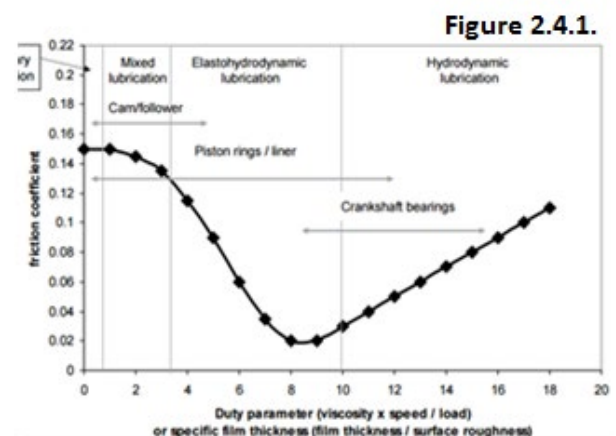
Load = 2000 N

Unit length = skirt length = 40mm

Duty parameter = $0.011619 \times 8 / (2000/0.04) = 1.859 \times 10^{-6}$

Initially it was taken as a possibility that when an

engine is under fired conditions with large side forces, that the piston may never leave boundary lubrication. Further Analysis of the equation shows that perhaps under some part load condition where the side force becomes very low due to the gas and inertia forces cancelling, the duty parameter may yield a value comparable to those illustrated in the graphs. However, this would mean that hydrodynamic lubrication is only reached at this point, which is unlikely as there would be excessive wear at all the other operating conditions.



Further research into the matter led to the author of the paper being contacted (Phil Carden at Ricardo) with a query about whether he had considered the possibility of the skirt never leaving boundary lubrication, and a request for confirmation of the units used in the duty parameter equation - although the units used were already confirmed from other sources. The reply from Ricardo is shown below.

“Abraham

The Stribeck curve I presented in that paper was for illustration only and should not be applied to any particular case. The x-axis was actually intended to relate to specific film thickness so the numbers are non-dimensional.”

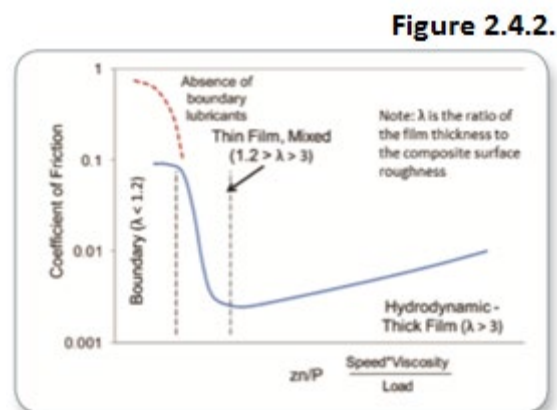
After this reply, additional sources of information were analysed for a solution to the problem. The sources and conclusions of each are listed below.

- Yeow-Chong Tan & Zaidi Mohd Ripin, Technique to Determine Instantaneous Piston Skirt Friction During Piston Slap, Tribology international, 2014,

Computes piston skirt friction force using a motored test whereby the force excitation was measured using a force transducer of an impact hammer and the response measured using an accelerometer. Shows how friction force changes with crank angle but gives no analytical method of relating it to the coefficient of friction.

- Bruker Nano Surfaces Division, Steve Shaffer, Generating a Stribeck curve in a reciprocating test.

Describes the type of test rig required to generate a Stribeck curve for reciprocating motion. Illustrates the same Stribeck curve as other papers but does not put values against the duty parameter figure 2.4.2.



- Tejaskumar Chaudhari & Bharatkumar Sutaria, Investigation of Friction Characteristics in Segmented Piston Ring Liner Assembly of ICE, Sciencedirect, 2016.

Describes an experimental test procedure for measuring reciprocating friction but gives no way of analytically relating piston speed and load to the coefficient of friction.

- Dinu Taraza and Naeim Henein, Friction Losses in Multi Cylinder Diesel Engines, 2017, SAE International.

Again, illustrates the same Stribeck curve but gives no values for duty parameter. Later states that the coefficient of friction varies linearly with the square root of the duty parameter. However, once the duty parameter is calculated and square rooted, values much higher than that given by the graph are produced. This can only mean that the graph must have also been created under motored conditions and thus shows no correspondence.

- Richard Stanley, Dinu Taraza & Naeim Henein, A simplified Friction Model of the Piston Ring Assembly.

Uses the same Stribeck curve with no values put to it. States that the log of the friction coefficient increases linearly with the log of the duty parameter. Once again must have been produced under motored conditions as values do not match.

- L. L. Ting, Ford motor Company, Development of a Reciprocating Test Rig, 2017

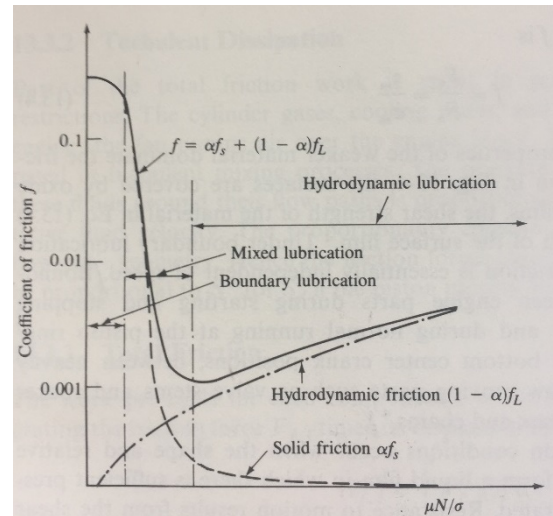
Explains the operation and function of a reciprocating test rig which generates data for friction loss computations. Shows the same Stribeck curve but again does not provide a means of calculating the friction coefficient through analytical methods.

- Richard C. Rosenberg, General Friction Considerations for Engine Design, General Motors Research Laboratories

Describes the different kinds of friction and uses the Stribeck curve to illustrate this. Provides a means of calculating the coefficient of friction but only for a bearing which cannot be applied to reciprocating motion.

- John B haywood, Internal Combustion Engine Fundamentals, 1998.

Once again shows the Stribeck curve depicting where the different regimes of friction occur. States that the “coefficient of friction for a journal bearing is plotted against a dimensionless duty parameter ($\mu N/\sigma$) and that for sliding surfaces the dimensionless duty parameter becomes $\frac{\mu U}{\sigma b}$. Where ‘U’ is the relative velocity of the two surfaces and ‘b’ is the width of the sliding pad in the direction of motion.” This corresponds with Phil Carden’s paper but once again does not go any further into the matter with no comparison of duty parameter results to the coefficient of friction value.



Another possible method of relating the coefficient of friction comes from the fact that the coefficient of friction for the engine’s crank bearing can be calculated from: $\frac{\pi^2 D_b}{h} \times \frac{\mu N}{\sigma}$ with ‘Db’ being the bearing diameter, ‘h’ the mean radial clearance, ‘μ’ the dynamic viscosity, ‘N’ the rotational speed and ‘σ’ the loading per unit projected area equal to $W/(L_b D_b)$. Calculating this and then calculating the duty parameter allows comparison with the piston skirt duty parameter. Knowing that bearings operate hydrodynamically, the coefficient of friction should be around the same number as the bearings. Calculation for the coefficient of friction of the bearing ranges from 0.009 at 1000 rpm to 0.03 at 6000 rpm with duty parameter ranging from 7.48963E-07 to 2.99E-06 respectively. The piston skirt duty parameter gives values ranging from 0.000106413 at 1000 rpm and 0.000280731 at 6000 rpm, meaning a difference in duty parameter at 1000 rpm of 1.056×10^{-4} . Comparing this to the difference in bearing duty parameter from 1000 to 6000 rpm of 2.24×10^{-6} which gives a 0.021 increase in the coefficient of friction means this proportion of increase occurs 46 times to reach the duty parameter of the piston skirt. Perhaps this is down to the shape of the Stribeck curve which is not linear. However such a large difference in duty parameter means the only conclusion can once again be that the duty parameter cannot be related to the piston skirt and that currently there is no mathematical method to calculate its coefficient of friction.

Other papers and books that have been examined and mention the Stribeck curve but again do not give a mathematical method for calculating the coefficient of friction include:

- Bosch Automotive Handbook, 9th edition, 2014
- Bosch Automotive Handbook, 8th Edition, 2012
- SAE International, Advances in Powertrain Tribology, 2000
- F.T. Barwell, Bearing Systems: Principles and practice, 1979
- Colin R. Furguson & Allan T. Kirkpatrick, Internal Combustion Engines, 2nd edition, 2001
- J Halling, Principles of Tribology, 1978
- James j. O'Connor, Standard handbook of lubrication engineering, 1985

So as there is currently no way of calculating the coefficient of friction for a reciprocating motion from load and velocity data without the use of a reciprocating test rig means a more approximate approach is needed. From the work carried out on friction mean effective pressure and the standard value of 0.1 used for the coefficient of friction it seems the only way to validate this value is to calculate entire engine friction and then calculate friction produced by individual components. Adding the friction up for the individual components and comparing to the entire engine friction will show how far off a coefficient of friction of 0.1 is from real life. The coefficient of friction can then be varied for individual components until the sum of the FMEP matches that calculated for the entire engine.

Chapter 3: Calculation of Entire engine friction

3.1. Calculation Of Total FMEP from Torque data

To calculate entire engine friction, data from Ford's 1.6 Ecoboost engine will be used which includes torque values at different rpms. By creating another spreadsheet for the 1.6 Ecoboost engine with the torque at each rpm included allows the BMEP to be calculated through:

Work done in 2 revolutions = torque x 4π = BMEP x Engine Displacement

$$BMEP = Torque \times \left(\frac{4 \times \pi}{Engine\ displacement} \right)$$

In this case the engine displacement is equal to 0.0016 m³.

The aim is to calculate the IMEP and subtract the BMEP. This will yield the total FMEP. To calculate IMEP we need the sum of the work done per degree on the piston. To calculate the work done at each crank angle:

$$\text{work done} = \text{Average pressure (psi)} \times \text{chnage in volume (m}^3\text{)}$$

Cylinder volume data was not provided with the pressure data. For this reason the Cylinder volume per degree had to be calculated differently to previous. Knowing the engine has a compression ratio of 10 the volume at TDC can be found.

$$\frac{V_{bdc}}{V_{tdc}} = 10$$

Volume per cylinder = 400 cc

$$V_{bdc} = V_{tdc} + 400$$

$$\frac{V_{tdc} + 400}{V_{tdc}} = 10$$

$$V_{tdc} + 400 = 10 \times V_{tdc}$$

$$400 = (10 - 1) \times V_{tdc}$$

$$400 = 9 V_{tdc}$$

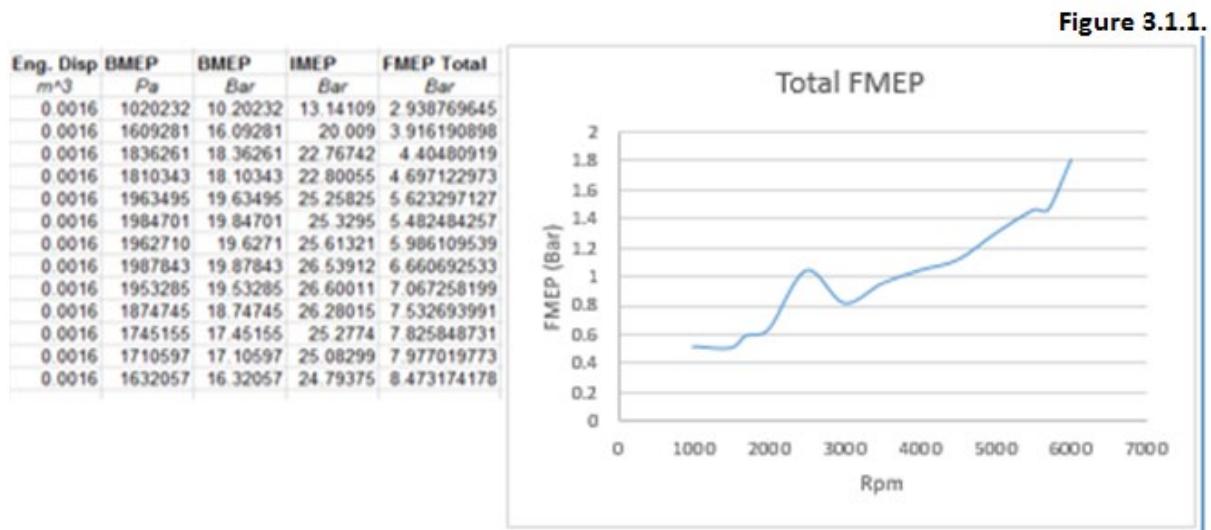
$$\frac{400}{9} = V_{tdc}$$

$$V_{tdc} = 44.444 \text{ cc}$$

Then using the area of the bore and the piston distance moved (S) the change in volume per degree can be added to the volume at TDC.

$$Vol = 44.44 + S \times \text{Bore area}$$

These values then need to be converted to cubic metres and inserted into the work done equation. Calculated total FMEP at 0m offset is shown below. (Figure 3.1.1.)



The aim now is to calculate complete engine friction and see how the values correspond with the total FMEP calculated. It will then be possible to vary the coefficient of friction so that the values match. The FMEP of the piston skirt is already known. A value of 0.1 for the coefficient of friction will be used initially.

The spike created at 2500 rpm in figure 3.1.1. is due to a sudden increase in cylinder pressure. This can only be put down to irregularities in the engine's spark timing or inaccuracies in the practical test and cannot be compensated for in the model. For this reason, the sum of the individual components' friction will be matched with a line of best fit.

3.2. Ring Friction

The piston rings are used to control the lubrication between the liner and the piston lands and skirt. Most engines use three piston rings named the compression rings and oil control ring. The tension in the piston rings holds them out against the cylinder liner and hence generates friction. The gas pressure from the cylinder acts behind the rings and increases their radial force. These pressures are controlled by the engine speed and load.

(John B. Haywood book, *Internal Combustion Engine Fundamentals*, pg. 729)

The friction mean effective pressure is found from first calculating the radial ring force which is equal to the cylinder pressure in Psi multiplied by the ring area.

Fr_{radial} = Pressure x Area

The ring area is found from multiplying the inner diameter by Pi to obtain the circumference and then multiplying this by the ring thickness. Units are in cubic metres.

The ring friction force is then found from:

$$= \mu \times F_{radial}$$

To find the work done due to friction of the ring the friction force is again multiplied by the piston distance travelled.

$$W.D. = ring\ friction \times Piston\ distance\ moved$$

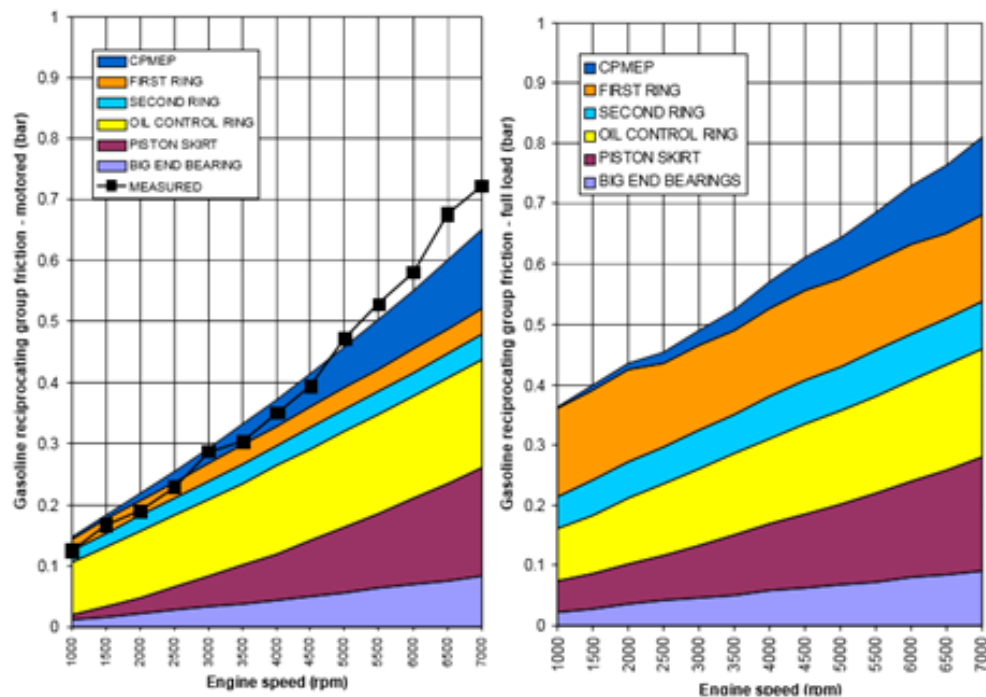
Obtaining the sum of these values after the values have been changed to positive allows the top ring FMEP to be calculated.

This method is only viable for the top ring so the second ring and oil control ring friction must be calculated through practical testing. Difficulty arises due to the ring being subjected to different kinds of loads that change over time, such as: gas pressure loads; inertia loads due to piston acceleration and deceleration; friction loads due to ring contact with the cylinder liner; hydrodynamic forces and oil damping forces; forces due to asperities contact; tangential forces due to the pretension of the ring; contact forces between the piston ring and the piston and the elastic distortion of the piston and liner which alters the effective geometry of the ring face and cylinder liner contact. This then causes a non-uniform distribution of the contact pressure between the cylinder liner and the piston ring face. It is because of these many factors that the required equipment to carry out such a test could not be obtained at the time of writing.

(Jozef Dlugos, Brno University of technology, Computational modelling of piston ring dynamics, 2014)

To cater for this a data set from Ricardo on the friction of a 2.0 litre TFSI engine is used to give an approximate value of total ring friction. Reading from figure 3.2.1. at full load a value of approximately 0.3 bar is given for the total friction produced by the first, second and oil control ring throughout the engine operating speed.

Figure 3.2.1.



3.3 Valve Friction

The valve train of an engine carries high loads over the entire engine speed range. Loads acting on the valvetrain at low engine speed are primarily due to the spring force while at high engine speeds the inertia of the components dominates. The total valve train friction is created by the valve-tappet contact and the camshaft bearings. In this chapter, the valve to tappet friction will be calculated.

Firstly, the valve lift is required to calculate the contact force of the cam on the tappet. The valve lift has been measured using a degree wheel and DTI gauge. The cam is rotated a degree at a time and the lift read off the DTI gauge (figure 3.3.1.).

Figure 3.3.1.



The valve lift has also been calculated from the cam profile as this was known from a previous study. A comparison can then be made on the accuracy of results. To calculate the valve lift from the cam profile the base radius and tappet clearance are required. For the 1.6 Ecoboost this is 15.05 and 0.3mm respectively. Adding the lift at each degree to the base radius gives a value 'S' which is the distance from the midpoint of the cam to a certain lift on the lobe. Starting with the cam facing upwards at full lift the cam angle is set to zero figure 3.3.2. A value 'OB' is then calculated for each degree from vertical through the equation:

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

With θ being the cam angle from vertical and ϕ being the angle from full lift. All angles are converted to radians prior to use.

Repeating this gives 360 values of OB for 360 degrees of cam rotation. Finding the maximum value for each column and then finding the first value of OB that equals the base radius and the tappet clearance is the contact point. OB then continues to rise until full lift is reached and then decreases. The point at which the base radius and tappet clearance is reached again is the closing point.

Figure 3.3.3. shows the measured and calculated valve lifts. To reduce noise the lift has been averaged i.e. $L_1 + L_2 + L_3/3$.

Now that the valve lift is known the contact force can be calculated through:

$$F_c = \text{Moving valve mass} \times \text{valve acceleration}$$

The moving valve mass includes the valve, retainer, collectors, tappet and half the spring mass.

Figure 3.3.2.

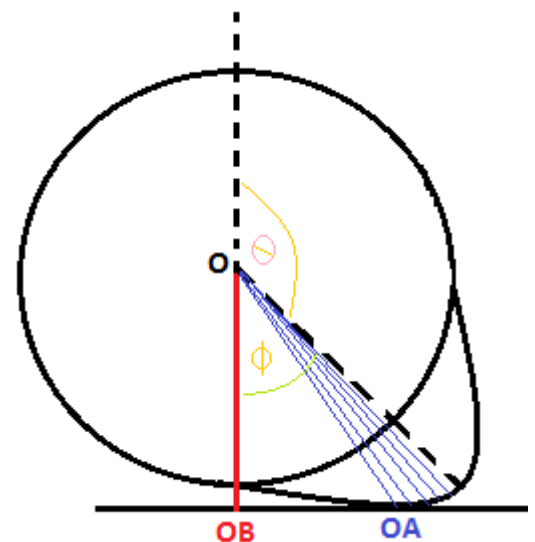
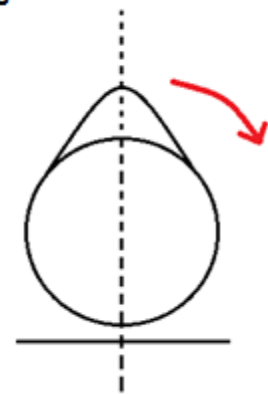
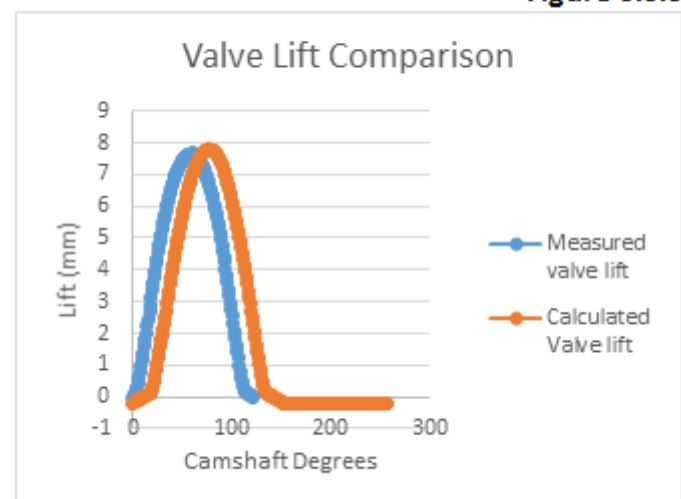


Figure 3.3.3.

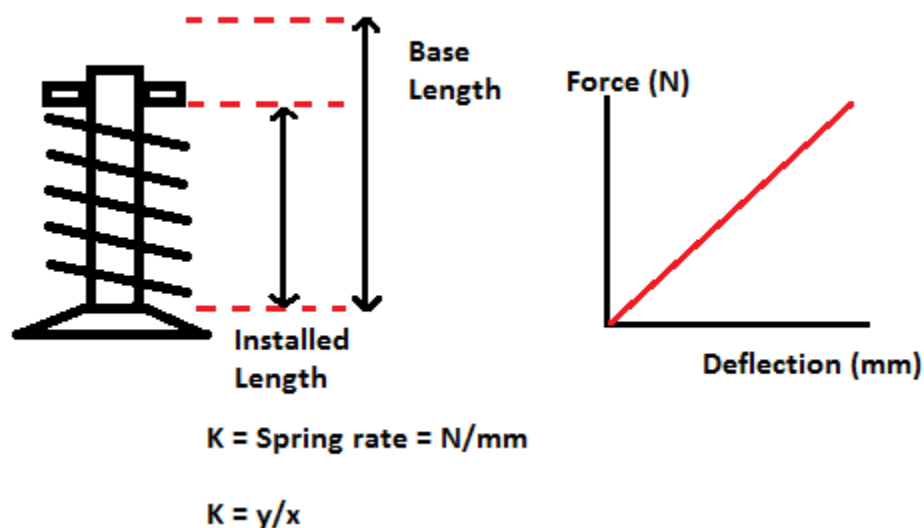


Masses						
	inlet valve	Tappet	Retainer	collets	Half spring	Total
(g)	32	25	7.73	0.6	10.16	75.49
(Kg)	0.032	0.025	0.00773	0.0006	0.01016	0.07549

To calculate the valve acceleration the velocity in mm per degree for each degree of cam rotation is required. This is found by subtracting L_1 from L_2 and L_2 from L_3 and so on. Subtracting V_1 from V_2 etc then gives the acceleration in mm per degree per degree. Dividing by 1000 and then Multiplying this by $(6.Rpm^2)$ converts it to metres per second squared.

Contact force is then found using the equation stated earlier.

The contact force from the cam lobe to the tappet can only push and give positive acceleration. When a spring is installed in the valve train there is an initial compression named 'Yo'



The spring force is opposing the contact force so the equation becomes:

$$F_c - \text{spring force} = m \times a$$

The spring force depends on the spring rate, initial compression and the amount of valve lift.

$$F_c - K(y_o + y) = m \times a$$

Rearranging gives:

$$F_c = m \times a + K(y_o + y)$$

So at full lift the acceleration of the valve train is negative. At full lift the spring rate will be a positive number. At high rpm the acceleration at full lift is negative and can be a big enough value that it is equal and opposite to the spring force ($F_c = 0$). This point is the float speed of the valve which dictates the maximum rpm possible for the current drive train.

From the contact force the friction force can be calculated by multiplying by the coefficient of friction. The work done per degree is then the average friction force multiplied by the distance moved.

The distance moved 'x' is calculated by first finding the instantaneous velocity fig 3.3.4. 'S' is already known and omega can be calculated for each rpm interval (remembering the cam turns at half engine speed). Looking at figure 3.3.5 it can then be said that vertical velocity = instantaneous velocity x $\sin \alpha$

and

$$x = s \times \sin \alpha$$

$$\text{but } v = s \times \omega$$

$$\text{so vertical velocity} = S \cdot \omega \cdot \sin \alpha, S \cdot \sin \alpha = x$$

$$\text{so vertical velocity} = \omega \cdot x \quad \text{thus}$$

$$x = \frac{\text{Vertical velocity (mm/sec)}}{\omega} =$$

$$\frac{\text{Vertical velocity mm per second} \times (60 \text{Rpm})}{\frac{\text{Rpm}}{60} \times 2\pi} = \text{Vertical velocity (mm per degree)} \times \frac{360}{2\pi}$$

The first contact point between the cam and the follower is when V and x = 0 (i.e. in middle). Another method of calculating 'x' is included in the appendices.

Converting 'x' to metres then allows the work done per stroke and friction mean effective pressure to be calculated in the same way as it was for the piston. The 1.6 Ecoboost is a 16-valve engine so the value for FMEP must be multiplied by this number. Figure 3.3.6. shows the Inlet, Exhaust and Total FMEP.

(Andrew Harrison, Valve Timing, UWTSD, 2015)

Figure 3.3.4.

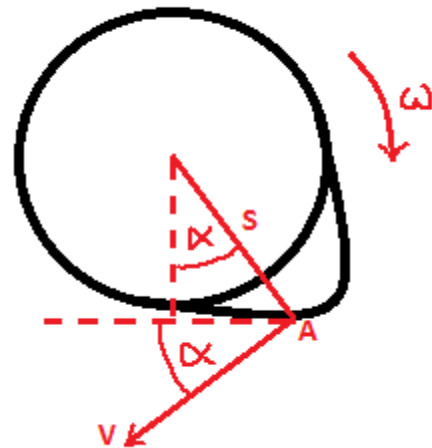
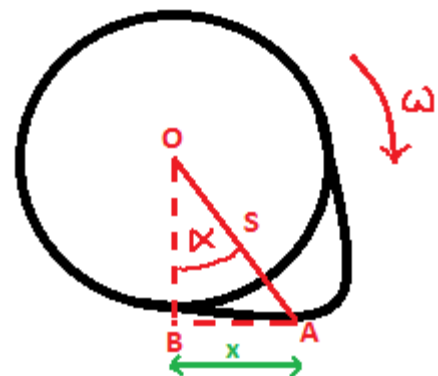
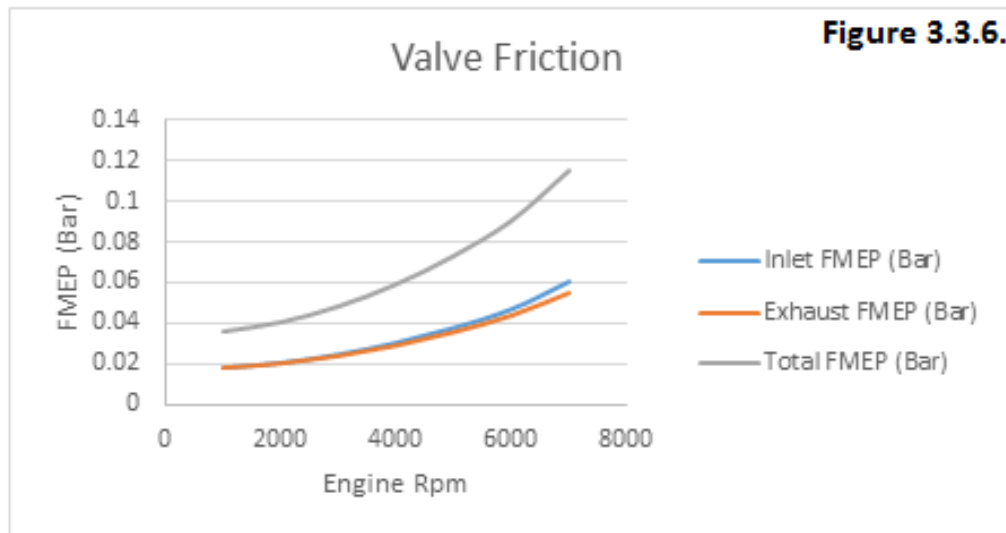


Figure 3.3.5.





3.4. Crankshaft Bearing friction

Crankshaft friction contributions in this case come from journal bearings, connecting rods and the associated seals. Large loads are carried by journal bearings with low energy losses under normal operating conditions due to the separation of the two surfaces by the hydrodynamic lubricant film. The loads on the bearings vary with the gas forces and inertia forces. To calculate a bearings friction the minimum oil thickness must be determined. This is the minimum separation distance between the journal and bearing surfaces. If the film thickness is too low asperities will break through the film thus increasing friction and wear. Dr. Dmitri Kopeliovich (Research and Development manager at King Engine Bearings Inc) states that typical values of oil clearance for passenger cars ranges from $0.0005 \times \text{Diameter}$ to $0.001 \times \text{Diameter}$.

Oil clearance of Journal bearings = $(0.0005 \times 0.047 + 0.001 \times 0.047) / 2 = 0.00003525 \text{ m}$

Oil clearance of Conrod bearings = $(0.0005 \times 0.044 + 0.001 \times 0.044) / 2 = 0.000033 \text{ m}$

Oil clearance of Camshaft bearings = $(0.0005 \times 0.025 + 0.001 \times 0.025) / 2 = 0.0000187 \text{ m}$

To calculate the bearing friction mean effective pressure the work done in 2 revolutions due to friction must first be calculated and is equal to:

$$2 \times \frac{\mu \times \pi^2 \times D^3 \times \omega \times L}{d}$$

'M' being the dynamic viscosity of the oil, 'D' the diameter of the bearing, ' ω ' the angular velocity, 'L' the length of bearing and 'd' the oil clearance. The oil viscosity of the recommended oil for the 1.6 Ecoboost at different temperatures is shown in table 3.4.1. Difficulty arises from the fact that the oil creating the film is likely to be hotter than the oil in the sump. Figure 3.4.2. shows how as oil temperature increases the friction mean effective pressure decreases. As an exact value of the oil temperature at the bearing is not known the value for dynamic viscosity at 100°C will be used. This requires conversion to Pa.s.

SAE 5W-40

3.4.1.

Temp. [°C]	Dyn. Viscosity [mPa.s]	Kin. Viscosity [mm²/s]	Density [g/cm³]
0	753.52	868.78	0.8674
10	378.65	439.85	0.8609
20	206.89	242.10	0.8545
30	121.90	143.70	0.8483
40	76.551	90.903	0.8421
50	50.861	60.849	0.8358
60	35.409	42.685	0.8295
70	25.631	31.135	0.8232
80	19.181	23.478	0.8170
90	14.742	18.185	0.8106
100	11.619	14.443	0.8045

Figure 3.4.2.

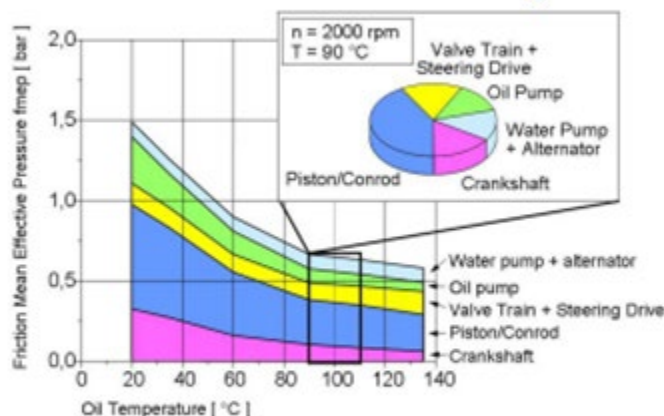


Fig. 1: Friction Source quantification for a 2.0l gasoline engine [7]

(Viscopedia.com, Viscosity tables, Substances, Engine oil.)

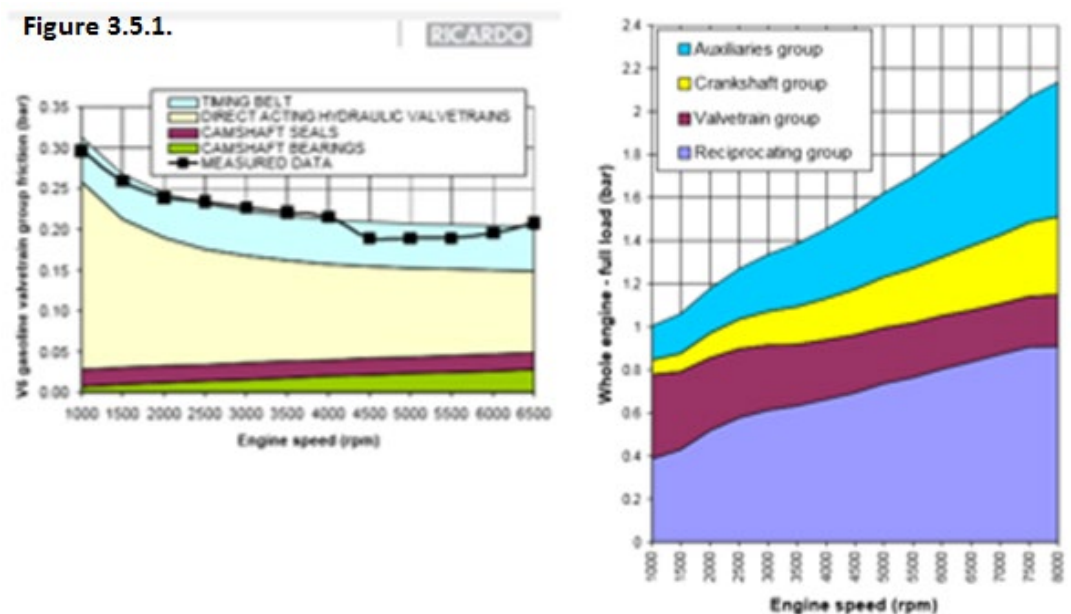
(KEW Engineering, Automotive Fluids, Lubricating oils and greases, fuels, coolants and Brake fluids.)

Dividing the work done by the engine displacement and then converting to Bar yields the bearings FMEP. This is then multiplied by the number of bearings - in this case 5 for the journal bearings. The FMEP of the conrod and camshaft bearings is calculated in the same way with the bearings diameter, oil clearance and number of bearings being the only difference.

3.5 Auxiliary Friction

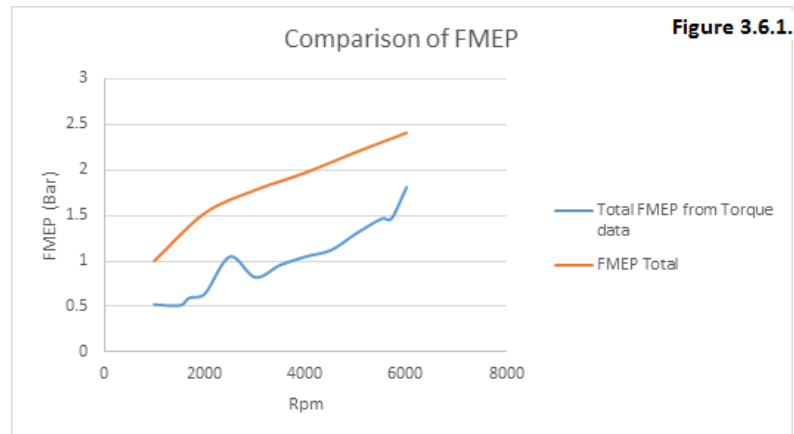
There are many additional components of the engine that will also cause friction. These include the water pump, oil pump, alternator, timing belt and the camshaft and crankshaft seals. Unfortunately, the friction produced by these components could not be calculated due to lack of data i.e. the Ecoboost engines use variable oil pumps which means a single value for flow rate cannot be obtained. Happily, however, the friction can be approximated through the use of Ricardo's paper on calculating engine friction in high performance engines. (Figure 3.5.1.) Reading from the graphs the timing belt seems to produce a relatively constant FMEP over the engine speed range of around 0.05 bar with the auxiliary friction increasing from around 0.175 Bar at 1000 rpm to around 0.5 bar at 6000 rpm.

Figure 3.5.1.



3.6. Comparison

As can be seen from figure 3.6.1., the sum of the different components' friction is much higher than that calculated from the engine's torque data. As the models are mathematically correct the only factor that could be the cause of such inaccuracy is the coefficient of friction used in the piston skirt, Top ring and valve friction models. As stated in chapter 2 the coefficient of friction changes with the side force and velocity of the piston. In this instance it seems the coefficient of friction used is much higher than it would be under real world operation. In order to determine how far off the models are the coefficient of friction will now be varied until the sum of the FMEP's matches that calculated from the torque. As a guide for how much friction should be generated from each component a data set from Mahle Powertrain for a 2.0 litre TFSI engine will be used. (Figure 3.6.2.)



Derived Teardown Results

Figure 3.6.2.

Engine speed (RPM)	Manifold losses FMEP (kPa)	Vacuum pump FMEP (kPa)	Water pump FMEP (kPa)	Balance shaft & Oil Pump FMEP (kPa)	Crank and seals FMEP (kPa)	Piston and rod friction FMEP (kPa)	Cylinder head with active valve train FMEP (kPa)	Pumping losses FMEP (kPa)	Complete engine measured result FMEP (kPa)
1000	1.8	4.0	-0.9	9.2	7.6	22.1	14.6	35.4	88.9
2000	6.1	2.8	0.6	16.9	13.4	34.2	13.7	37.7	115.8
3000	3.4	2.9	3.7	22.9	19.7	42.8	13.7	42.2	141.2
4000	8.2	2.6	7.9	23.7	26.4	55.2	13.6	64.0	183.0
5000	4.2	2.2	12.4	27.4	34.4	68.3	14.8	79.2	224.0
6000	6.2	2.4	18.1	34.3	41.9	80.7	17.8	108.6	283.1
6800	18.5	1.1	23.9	44.5	50.1	90.3	19.6	134.1	338.7

(Stewart Mackey, Mahle Powertrain, *Friction reduction programme*, 2009)

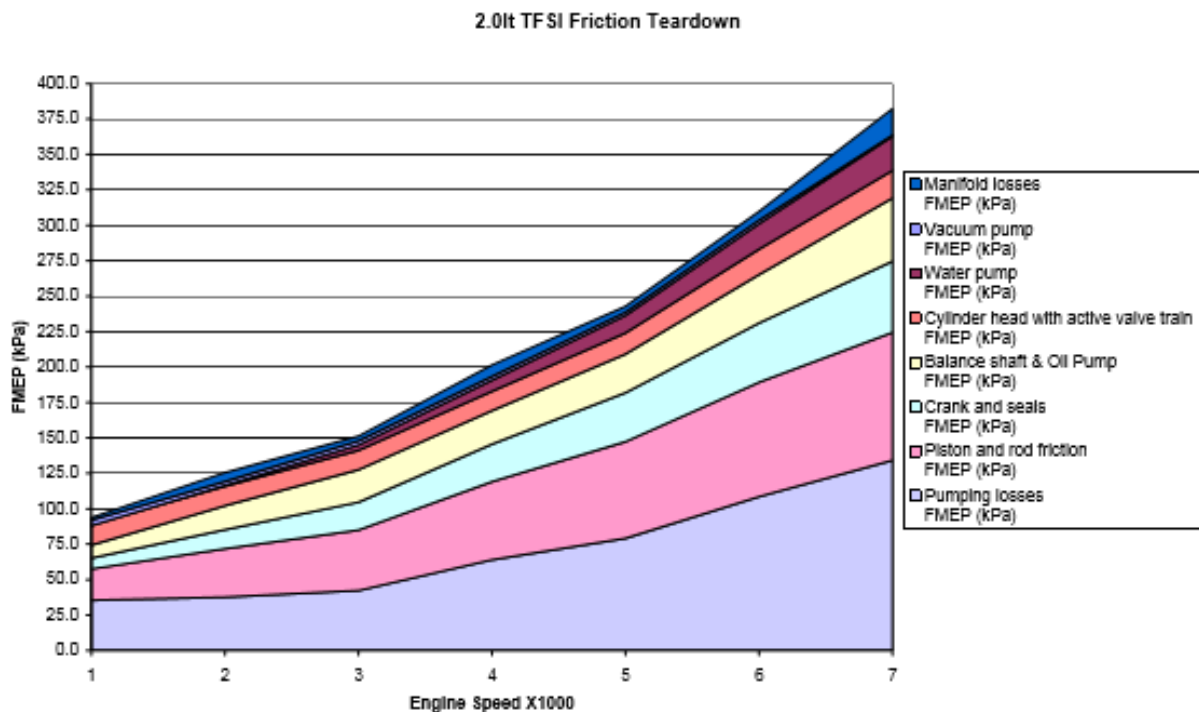
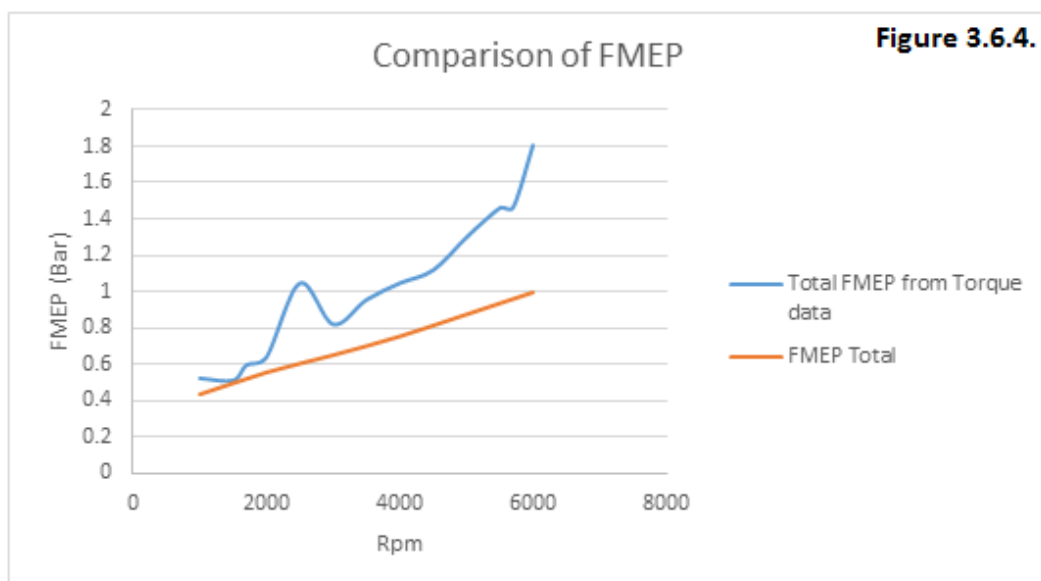
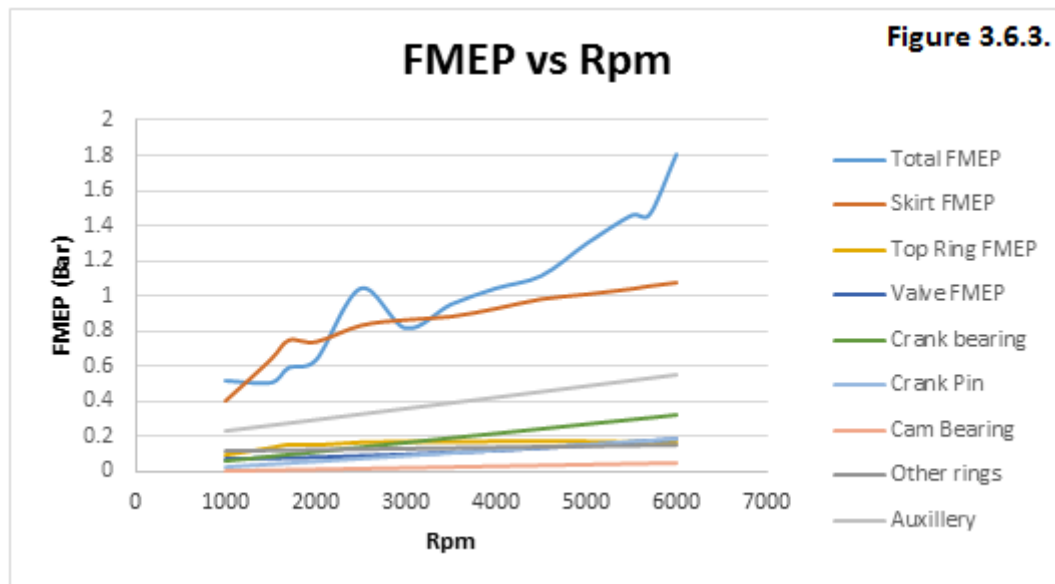
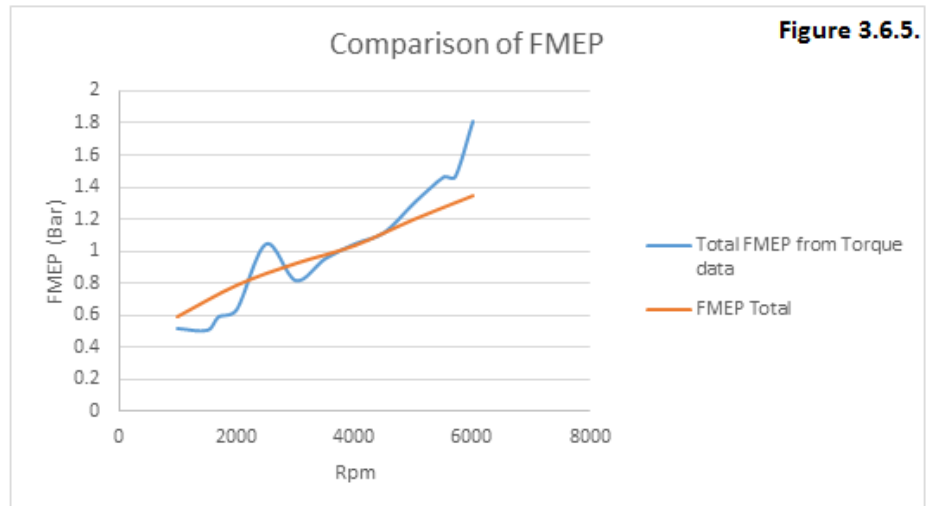


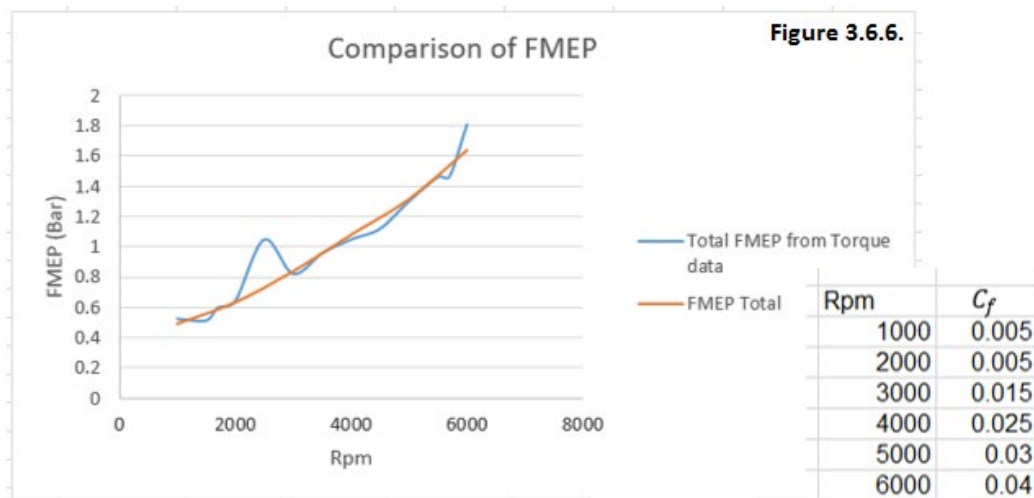
Figure 3.6.3. shows the FMEP for each component at a coefficient of friction of 0.1. Evidently the piston skirt friction is too high meaning the coefficient of friction must be reduced. Figure 3.6.4. shows the total friction with these components removed and thus shows the amount of friction that these components can occupy once the coefficient of friction is corrected. To make the values correspond the coefficient of friction was then altered.



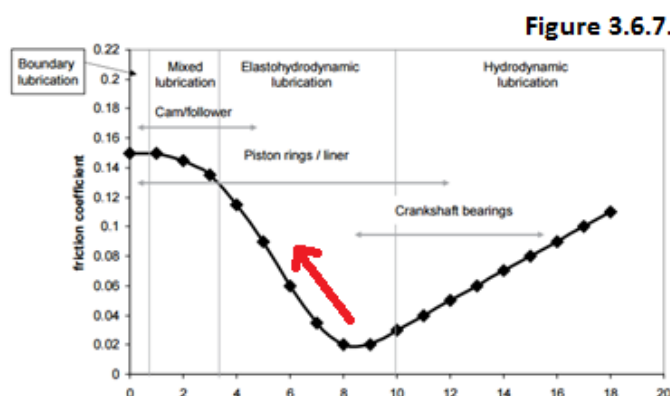
Using a coefficient of 0.025 alters the FMEP to that seen in figure 3.6.5. At this value, the FMEP values match at the mid rpm range. However, the FMEP at lower rpms remains too high and at higher rpms is too low meaning the coefficient of friction at high rpm is higher than that at low rpm. This is



almost certainly because of the increasing side forces and velocities. Higher side force means a lower duty parameter and thus the coefficient of friction is increased according to the Stribeck curve (figure 3.6.7). Higher velocities mean a higher duty parameter and so could be moving back up towards the right of the Stribeck curve. The coefficient of friction of each rpm interval is then altered (figure 3.6.6.) with the noise of the total FMEP from torque data trace ignored due to the likelihood of it being created by irregularities in combustion. The table in figure 3.6.6. shows the closest matching coefficient of friction values for the rpm intervals.

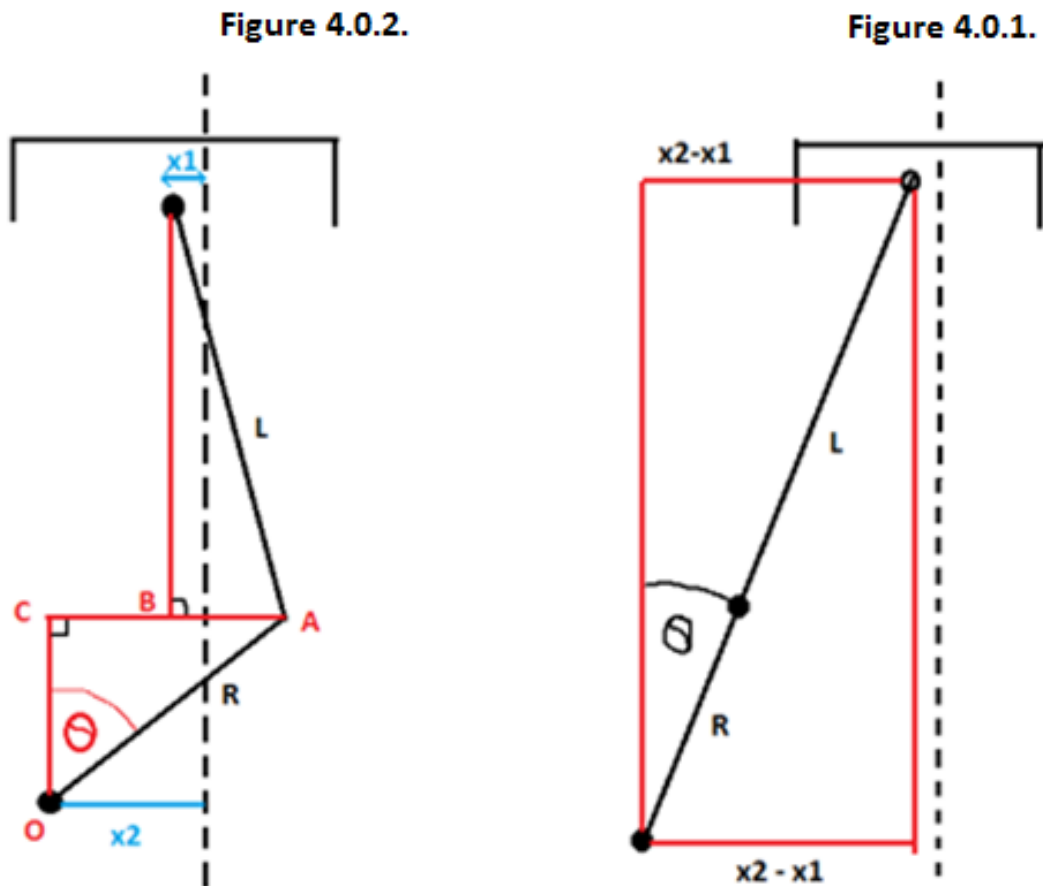


(Phil Carden, *Calculation of Friction in High Performance Engines*, 2014)



Chapter 4: Crank Offset Friction Model

Due to the 1.0 Ecoboost engine incorporating a crankshaft and piston offset, the method used previously to calculate friction could not be applied. In this case when the piston is at TDC the crank and rod angle is not straight i.e. they are still at an angle. Figure 4.0.1.



To calculate the actual crank angle a diagram is drawn such as the one in figure 4.0.2. From this it can be seen that $\sin\theta$ is equal to the crank offset minus the piston offset divided by the crank throw plus the rod length:

$$\sin\theta = \frac{(x_2 - x_1)}{(R + L)}$$

So the crank angle at TDC:

$$\theta_{TDC} = \sin^{-1} \frac{(x_2 - x_1)}{(R + L)}$$

In Excel this gives a value in radians. Adding this value to the original crank angle in radians gives the corrected crank angle.

The height at TDC can then be calculated by adding crank throw and rod length and multiplying by the cosine of the corrected crank angle.

$$HTDC = (R + L) \times \cos\theta$$

Likewise the conrod angle must be corrected as it is not straight at TDC. This is done by using the triangles again.

$$AC = AB + BC$$

$$R \cdot \sin\theta = L \cdot \sin\phi + (x_2 - x_1)$$

$$L \cdot \sin\phi = R \cdot \sin\theta - (x_2 - x_1)$$

$$\sin\phi = \frac{R}{L} \sin\theta - \frac{(x_2 - x_1)}{L}$$

$$\text{Thus the corrected conrod angle is } \phi = \sin^{-1} \left(\frac{R}{L} \sin\theta - \frac{(x_2 - x_1)}{L} \right)$$

The 'S' value or piston distance moved is also found from the triangles.

$$S = (R+L) \times \cos(\text{crank angle at TDC})$$

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

Using the corrected values for crank angle and conrod angle the velocity in metres per degree can then be calculated by subtracting S_1 from S_2 . Repeating gives the velocity at each degree over 720 degrees. The acceleration in metres per degree per degree is then found by subtracting V_1 from V_2 . Repeating for every crank angle to find acceleration in metres per second squared.

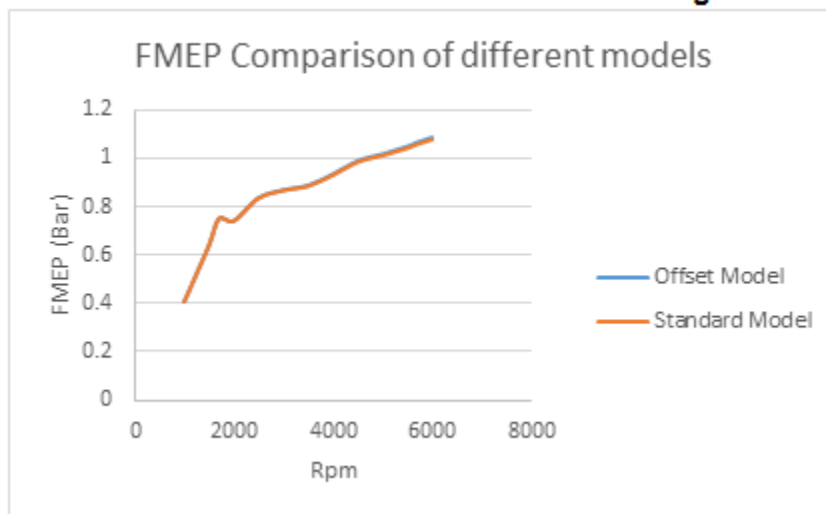
$$\text{Number of degrees per second} = 360 \times \frac{Rpm}{60}$$

This cancels to become 6 x Rpm.

Acceleration in metres per second squared is then equal to the acceleration per degree per degree multiplied by 6. Rpm^2 .

Knowing the acceleration means the inertia force can now be calculated from $F=ma$ with the mass being the reciprocating mass i.e. piston and pin mass plus a third of the rod mass.

Knowing the inertia force means the net downforce can be calculated and thus the rest of the spreadsheet can be completed. To check the new method works the offsets were set to zero and the FMEP compared to the FMEP obtained from the previous method. Figure 4.0.3. shows the comparison for the 1.6 Ecoboost.

Figure 4.0.3.

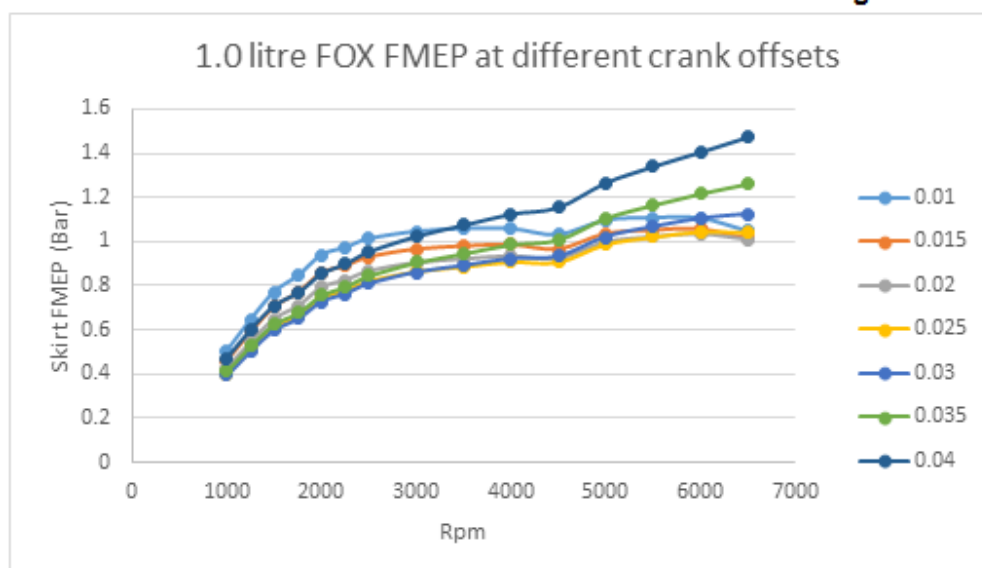
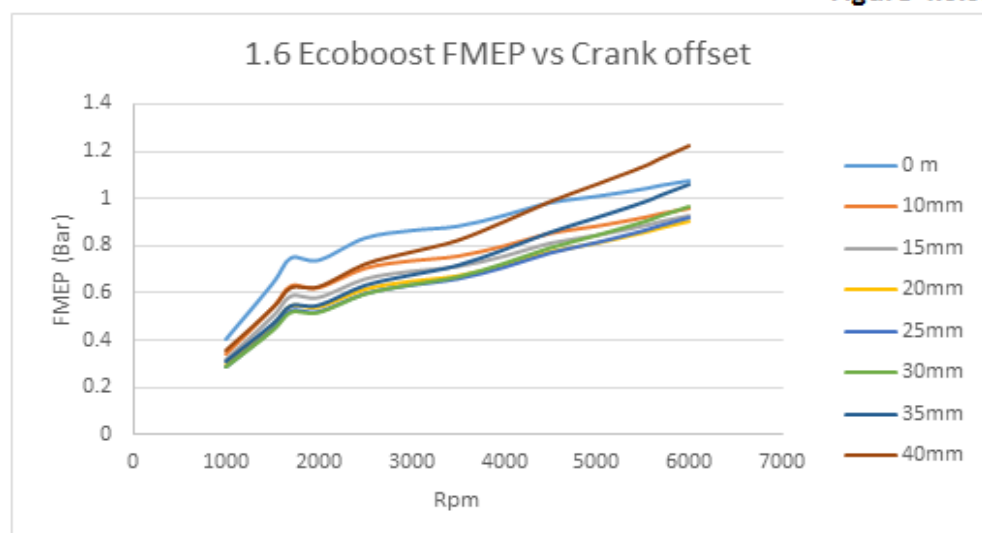
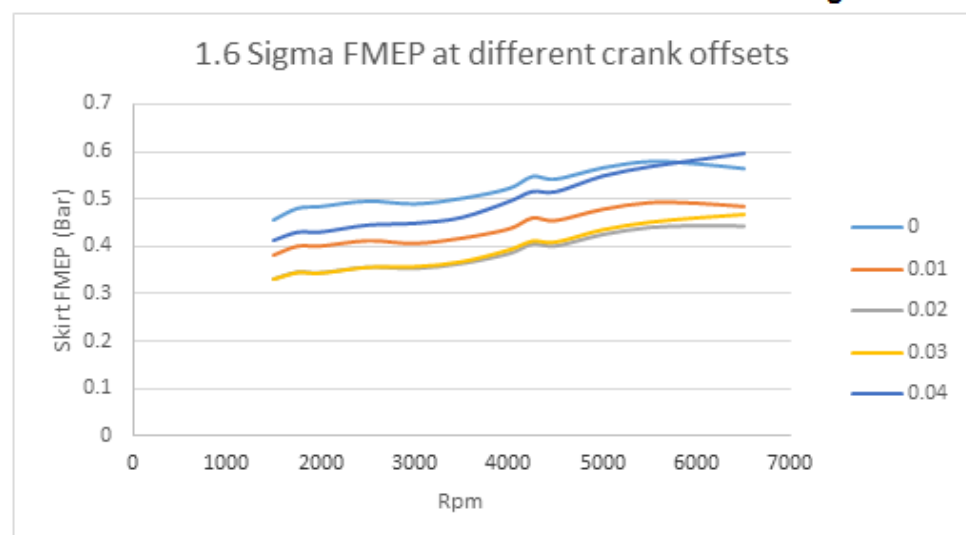
So $F_{side} = F_{net} \cdot \tan \phi$ and $F_{rod} = \frac{F_{net}}{\cos \phi}$

The friction force is then at right angles to the side force so $F_{friction} = \mu \times F_{side}$

The work done per degree due to friction can then be calculated seeing as work done is equal to a force multiplied by a distance. The friction mean effective pressure is then calculated using the same method as before.

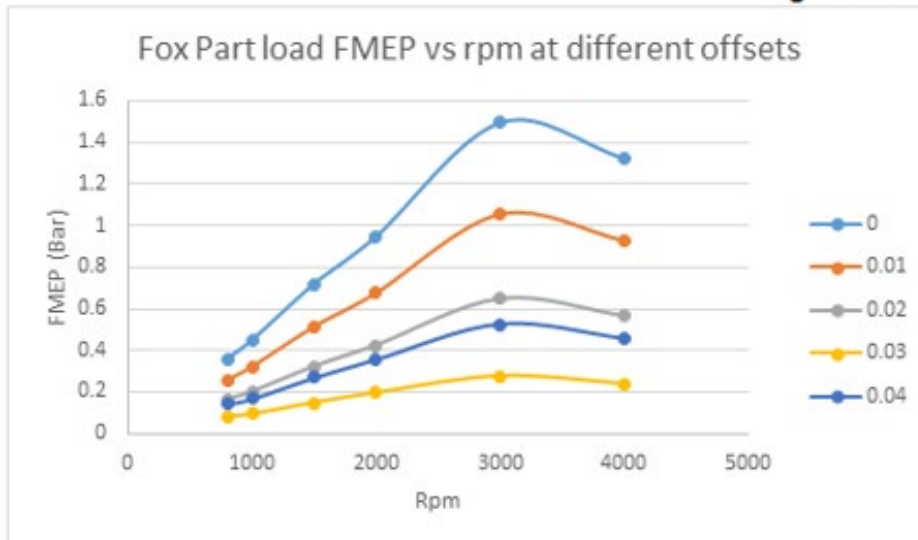
Looking at figures 4.0.4. to 4.0.6. it becomes evident how effective the use of a crank offset is. The offset model will now be applied to the 1.6 Ecoboost engine to see how much of a reduction in friction can be achieved. The optimum offset will then be applied to a Solidworks model of the engine. This will show how feasible it is and to what extent the engine would need to be modified.

Taking the values for the skirt FMEP at each rpm for a range of different crank offsets allows the following graphs to be produced:

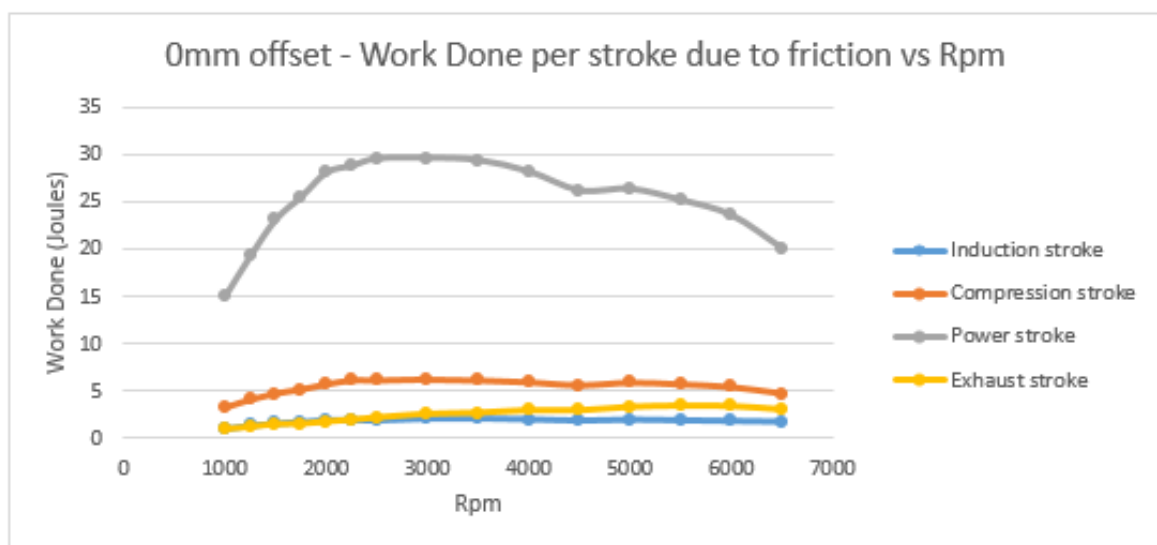
Figure 4.0.4.**Figure 4.0.5.****Figure 4.0.6.**

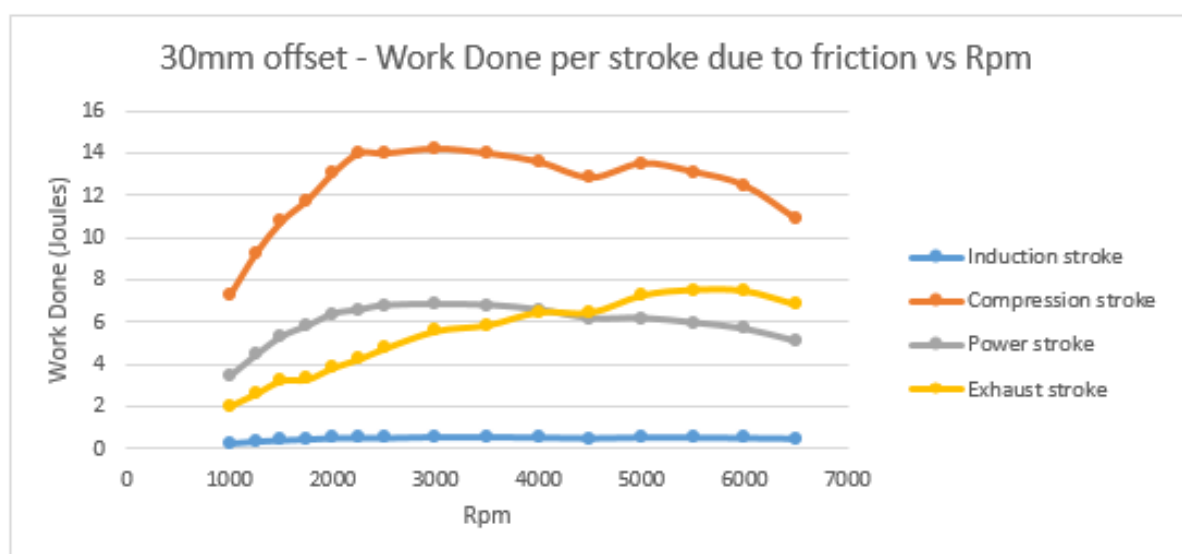
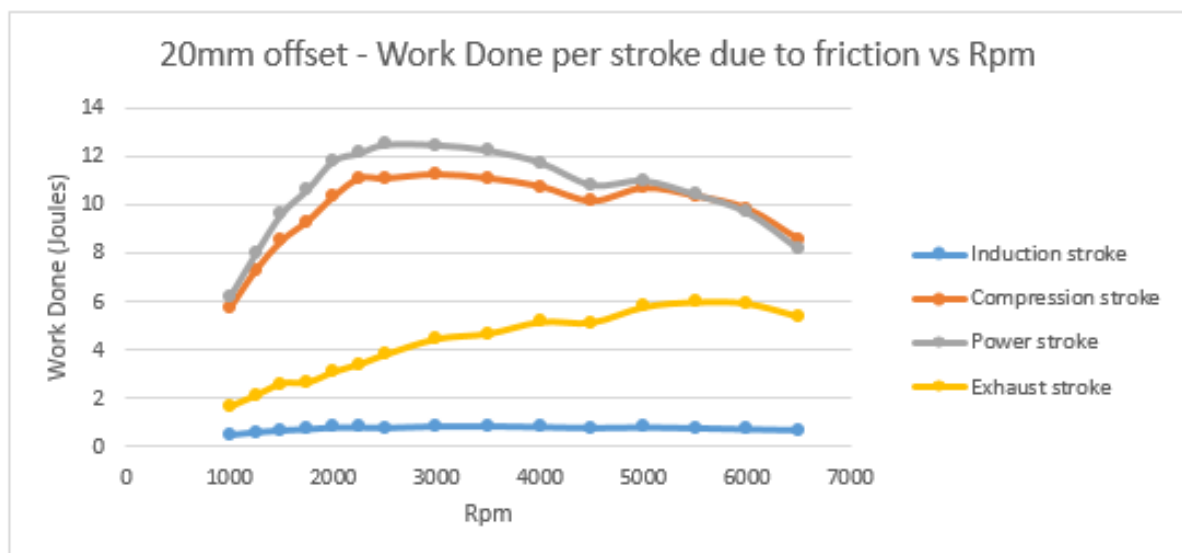
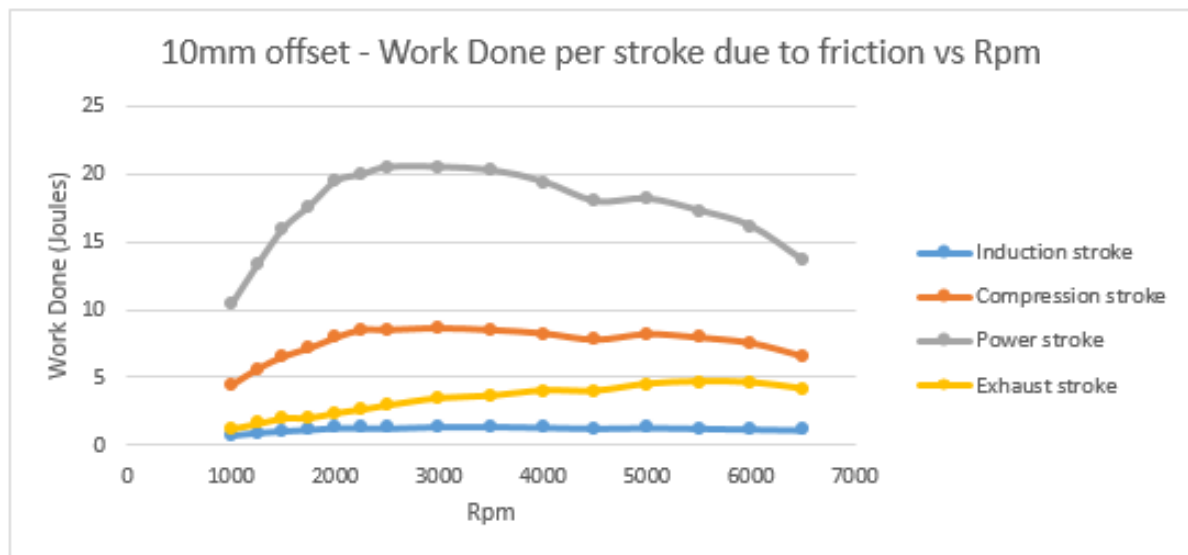
The graphs show that as the crank offset is increased from 0 to around 25mm the skirt FMEP starts to decrease with a minimum value at around 25mm. However, as the offset is increased further skirt FMEP starts to increase again until at 40mm the skirt FMEP is greater at high rpm than the 0 offset. This gives an indication that something is happening which makes the friction worse after a crank offset of 25mm is reached. Figure 4.0.7. shows the same graph for the 1.0 FOX engine at part load. This graph corresponds with the full load graphs showing the lowest FMEP occurring at around 25mm offset.

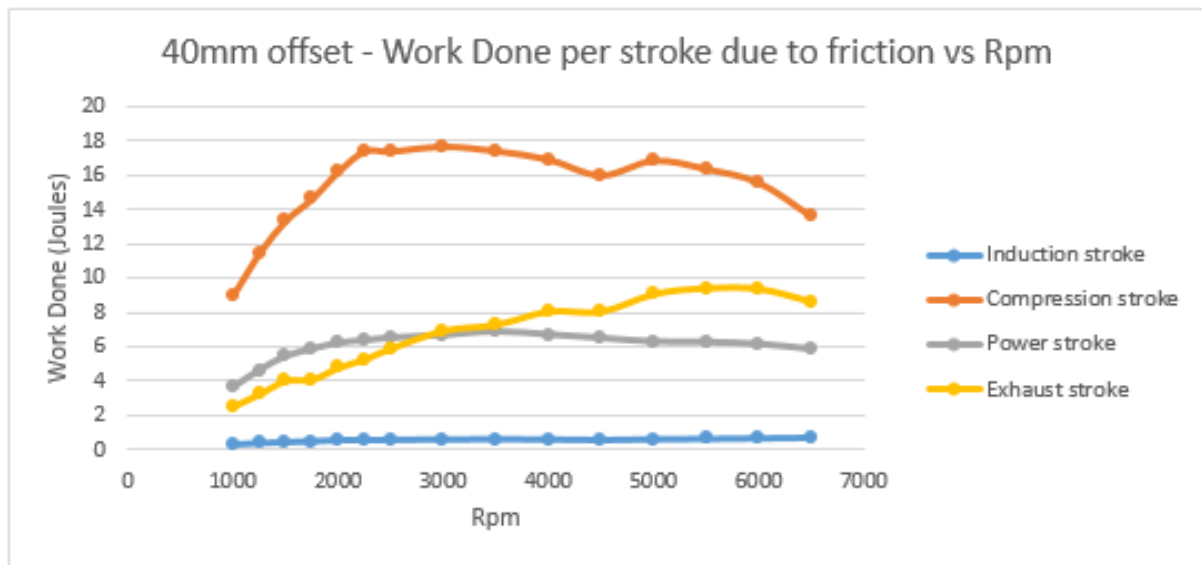
Figure 4.0.7.



The next graphs show the work done due to the friction of each stroke at different rpms with the crank offset left variable. This will allow understanding of which stroke is causing the most friction and how it changes with offset. The graphs go from zero offset to 40mm offset in 10mm intervals.







These graphs show a similar flipping characteristic with the compression stroke having a higher work done value after around a 22mm crank offset than the power stroke. The rod angles will now be examined to show the reason for this flipping.

From the stroke graphs it is evident that the total work done as a sum of the Individual total work done is lowest at 25mm, which corresponds with the previous offsets graphs (figures 4.0.4. to 4.0.6.). 25mm has the lowest overall total work done due to friction meaning lower maximum friction forces are generated and thus less wear generated. This corresponds with Musashi Institute of Technology in Japan who pursued a more detailed study using a floating liner single-cylinder engine which allowed them to measure piston friction force. Their measurements showed that on a crank angle basis, the friction reduction during the expansion stroke was greater than the friction increase during the compression stroke, resulting in a net reduction in friction.

(Steve Shin, Annette Cusenza and Fanghui Shi, *Offset Crankshaft Effects on SI Engine Combustion and Friction Performance*, 2004 SAE)

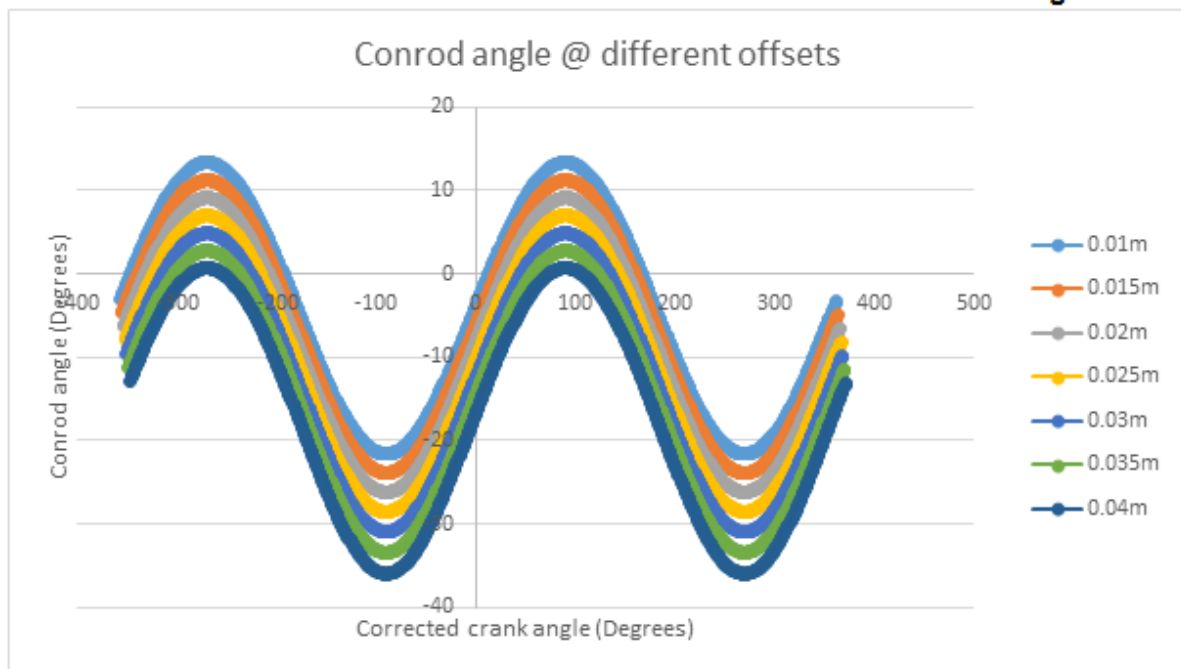
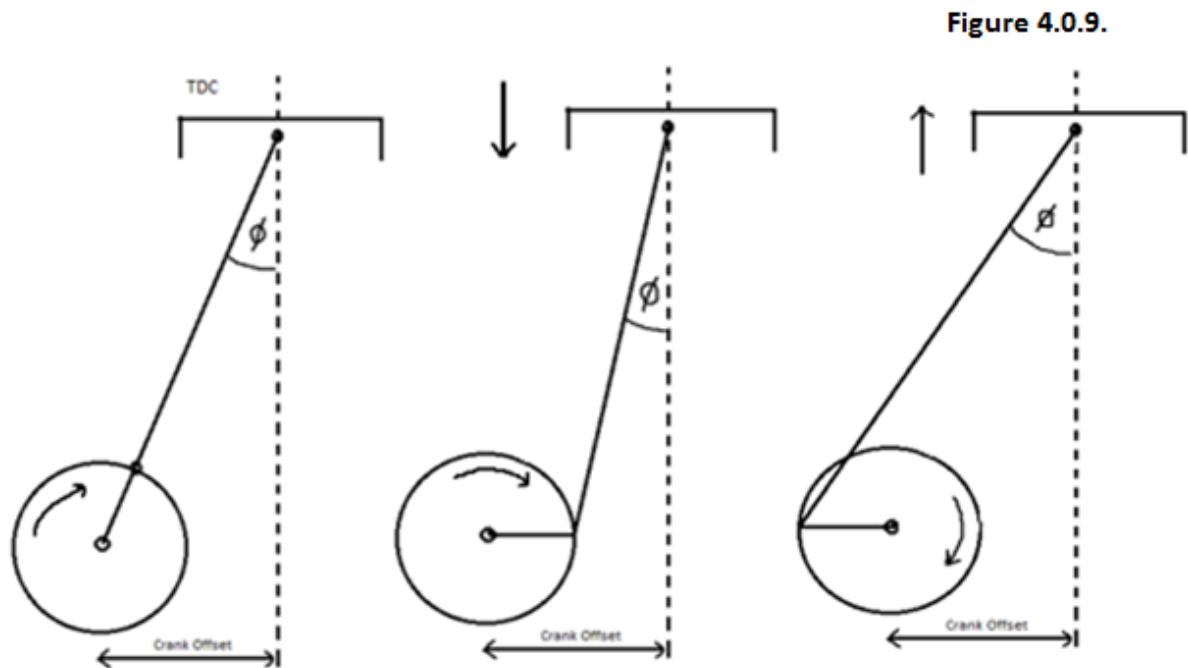
Figure 4.0.8.

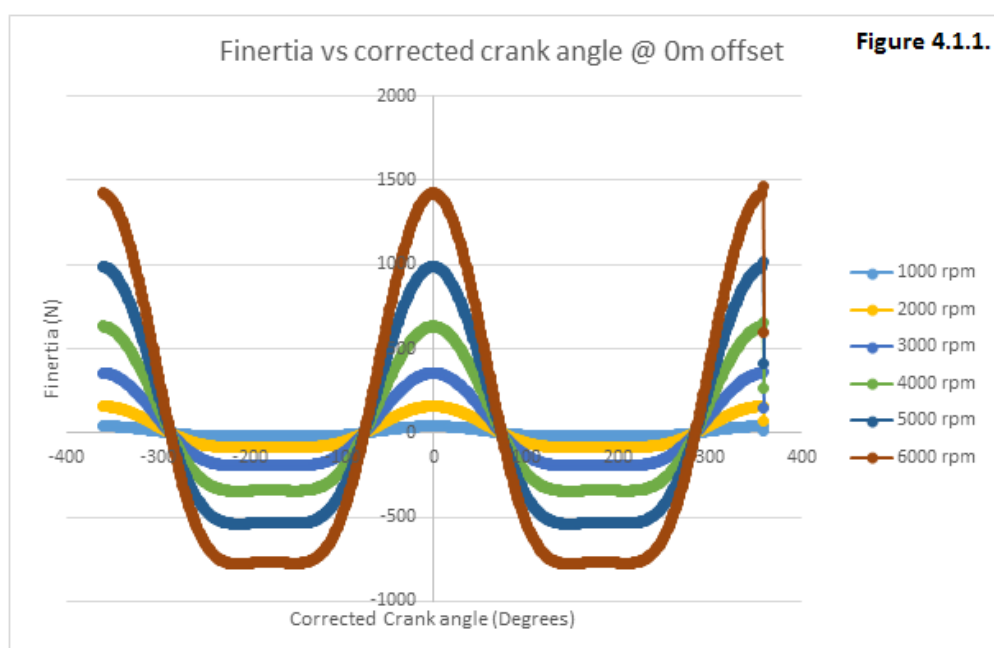
Figure 4.0.8. shows the conrod angle against crank angle at different offsets. As the crank offset increases, the conrod spends more time at a negative angle. This means as the offset increases, the rod angle on the down stroke gets smaller and smaller and so a straighter rod angle is achieved during the early parts of the expansion stroke, leading to reduced piston side forces when cylinder pressures are highest - thus reducing the friction. However, as the rod angle on the expansion stroke decreases the rod angle on the compression stroke is increasing (figure 4.0.9.). This direction of offset is termed positive crank offset or negative bore offset whereby the crankshaft is offset towards the major thrust side.

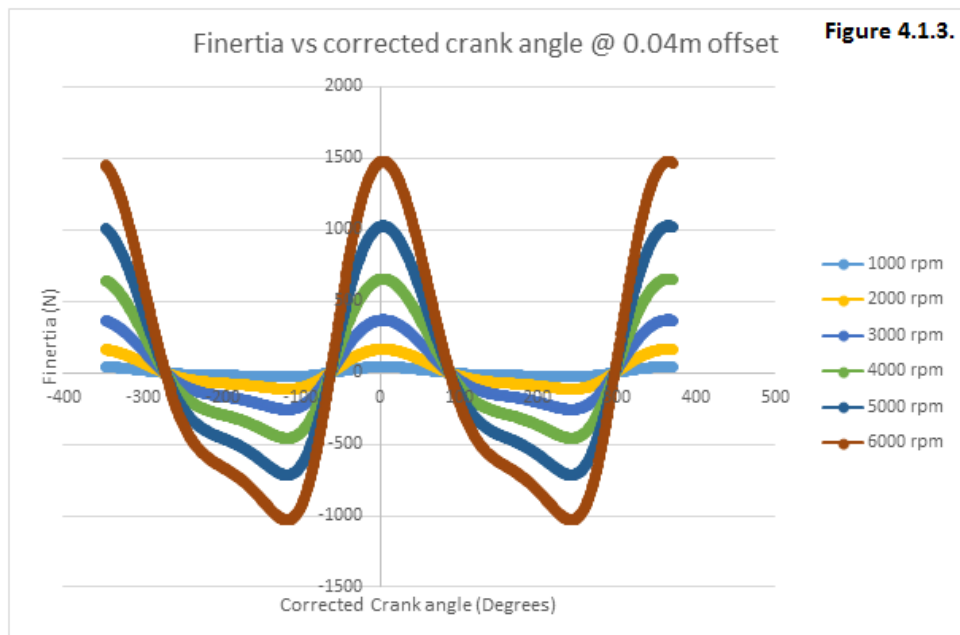
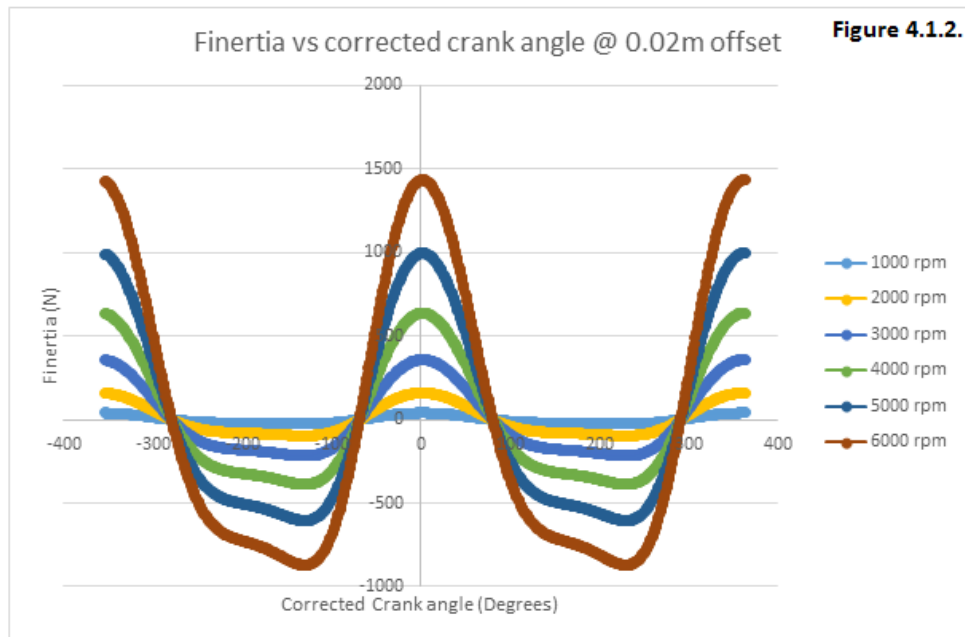
(Steve Shin, Annette Cusenza and Fanghui Shi, Offset Crankshaft Effects on SI Engine Combustion and Friction Performance, 2004 SAE)



4.1. The Effect of Offset on Inertia

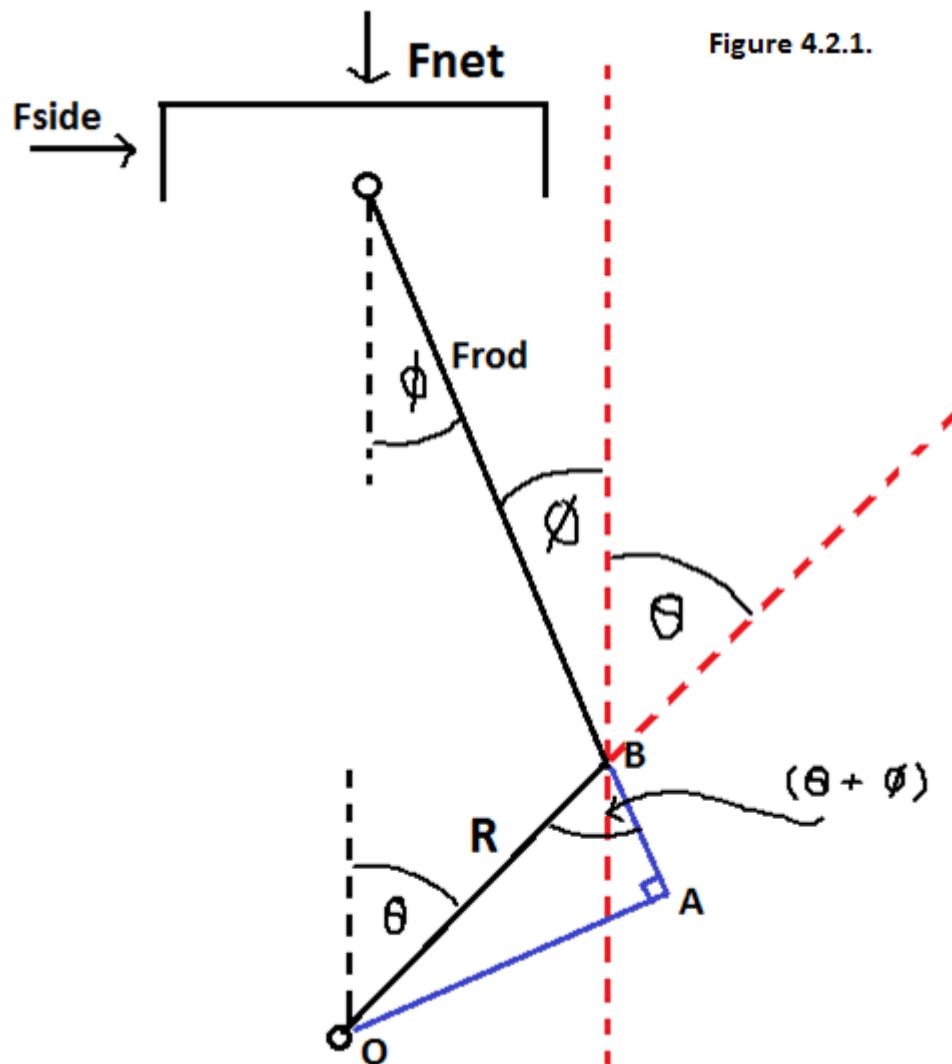
Figures 4.1.1. to 4.1.3. show the inertia as crankshaft offset is increased in 20mm intervals. As can be seen, the positive inertia doesn't seem to be affected, whereas negative inertia max is increasing as crank offset increases. The angle at which maximum negative inertia occurs is also seen to be moving further and further away from 180 degrees or BDC. This is because as the offset increases the BDC point in degrees is also increasing, with the rod spending more time at a negative angle and thus increasing negative inertia.





4.2. Effect of Offset on Torque

To see if the torque changes with offset the following graphs show the torque calculated through two different methods. One uses the rod force and the perpendicular distance. The other uses the IMEP.

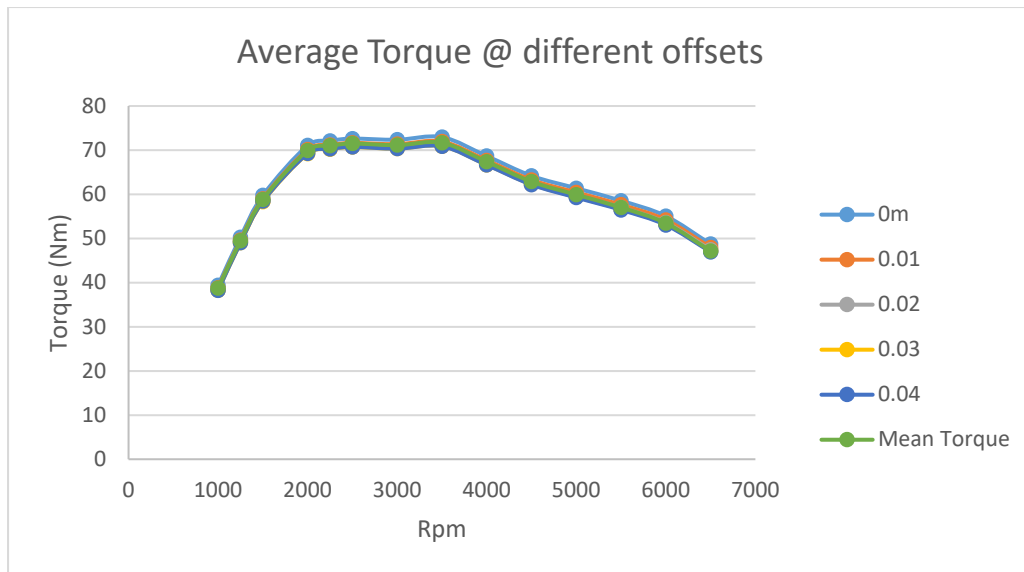


The torque at a particular angle is equal to $F_{rod} \times OA$.

We can see that $OA = OB \times \sin(\theta + \phi)$ so,

$$\text{Torque} = F_{rod} \times OB \times \sin(\theta + \phi)$$

F_{rod} is known to be $= \frac{F_{net}}{\cos \phi}$, OB is known as it is the Crank throw and the angles have been previously calculated



The above graph shows the torque at each rpm taken as an average of the total. As can be seen, increasing offset reduces torque ever so slightly with a reduction of around 2 Nm from 0 offset to 40mm.

The mean torque is calculated from:

$$\text{W.D. in 2 revs} = (\text{IMEP} - \text{FMEP}) \times \text{Eng. Disp}$$

$$\text{Indicated work done in 2 revs} = \text{IMEP} \times \text{Eng. Disp} = \text{Mean Torque} \times 4\pi$$

$$\text{Mean torque} = \frac{\text{IMEP} \times \text{Eng. Disp}}{4\pi}$$

It is shown as the green line and sits roughly in the middle of the values calculated using the rod force. This shows that the calculations are correct and that the effect of crank offset on torque can be said to be very small and insignificant.

4.3. The effect of rpm and load on the optimum offset.

Figure 4.3.1. and 4.3.2. show the work done due to friction against offset for each rpm interval. The 1.6 being at full load and the FOX at part load. As can be seen the lowest work done due to friction at both full load and part load occurs at around 25mm of crankshaft offset. This is also the case for the rpm intervals.

Figure 4.3.1

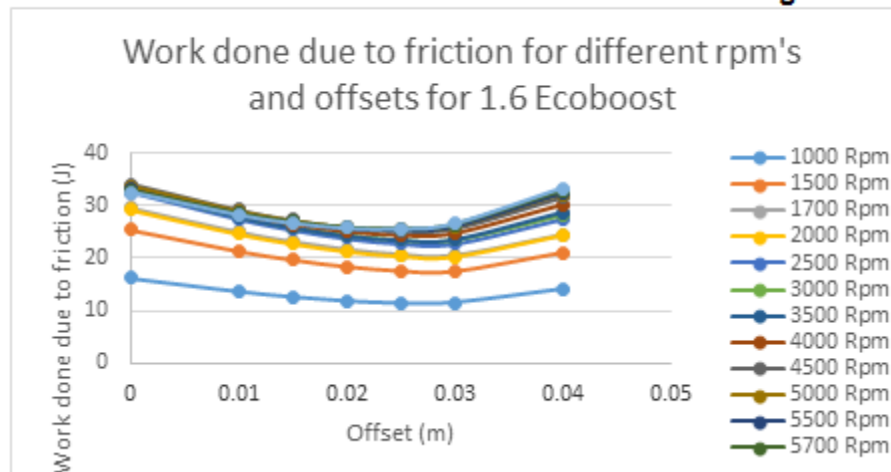
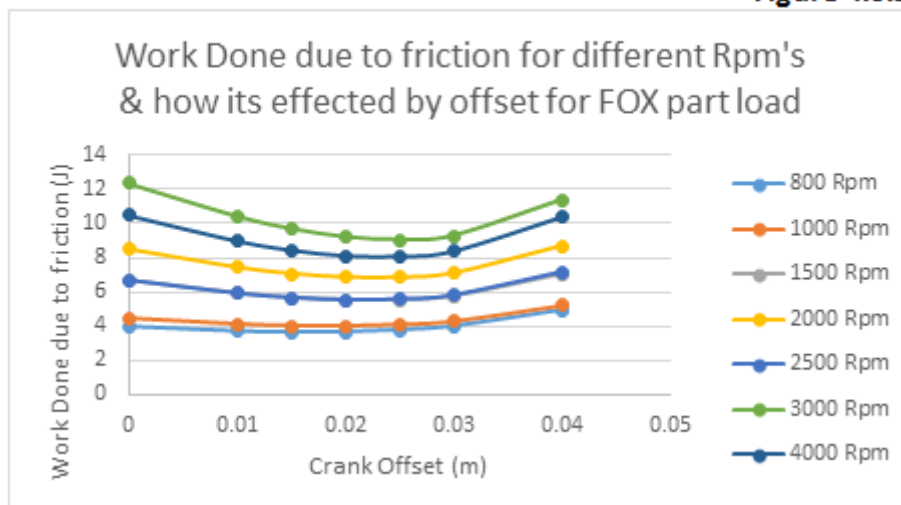


Figure 4.3.2.

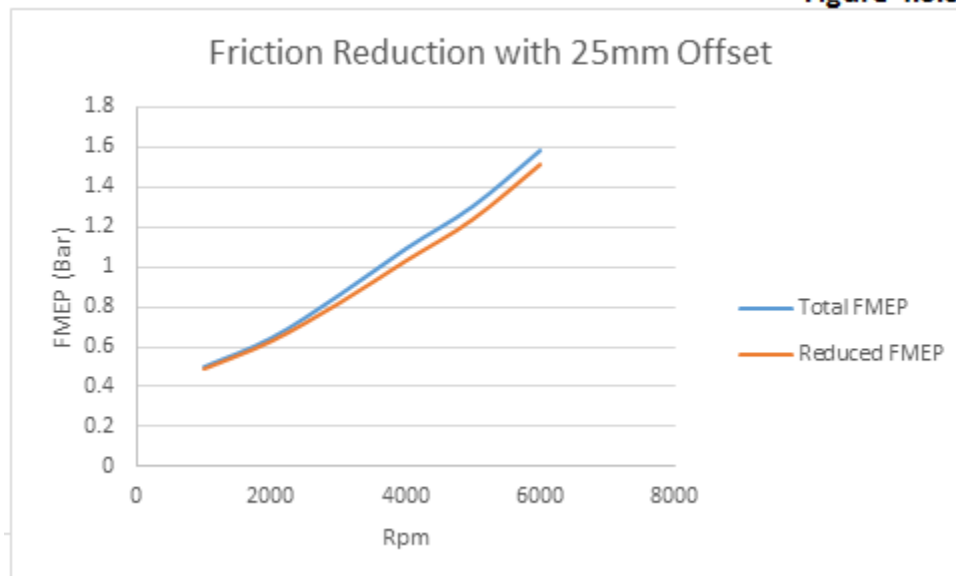


It seems then that a crank offset of 25mm would yield the lowest friction for the piston skirt with no foreseeable side effects. Now that the coefficient of friction has been corrected the crank offset can be reapplied to the model to see how much of a reduction in friction can be generated. Figure 4.3.3. shows the comparison of skirt FMEP for 0 offset and 25mm offset. Using this offset means a reduction in skirt friction of around 5% can be achieved.

0mm	25mm	Friction Reduction %
0.020308426	0.014328561	29.44523898
0.031970808	0.022177434	30.63223834
0.187252097	0.13078823	30.15393026
0.036995862	0.026024731	29.6550209
0.208270277	0.148816212	28.54659143
0.129742589	0.094731768	26.98483289
0.220788858	0.164409092	25.53560276
0.232212143	0.176654079	23.92556343
0.24560567	0.191810786	21.90294876
0.302824339	0.243276802	19.66405233
0.2599635	0.215068076	17.26989512
0.263970524	0.221011488	16.27417898
0.429894762	0.366319787	14.78849733

Total FMEP	Total FMEP with 25mm offset	% reduction
0.497820114	0.491804835	1.208323826
0.641855688	0.630817703	1.71969884
0.854593361	0.819363934	4.122361429
1.088406711	1.03247018	5.139304081
1.301068008	1.241080834	4.610610164
1.579047779	1.514944862	4.059593242

Figure 4.3.3.



Using the 1.6 Ecoboost engine a model was then made within Solidworks. Safety factors and life cycles were then found for individual components using the simulation software. An attempt has been made to modify the engine to accommodate 25mm offset while producing the same safety factors and life cycles as the original.

Chapter 5: CAD Models

To see what the maximum crank offset that would be possible within the Ford engine, a detailed model of the engine's crankshaft, conrod, piston and block were created. Creating an assembly out of these parts and then rotating it allows the clearance between piston, conrod, crankshaft and block to be analysed.

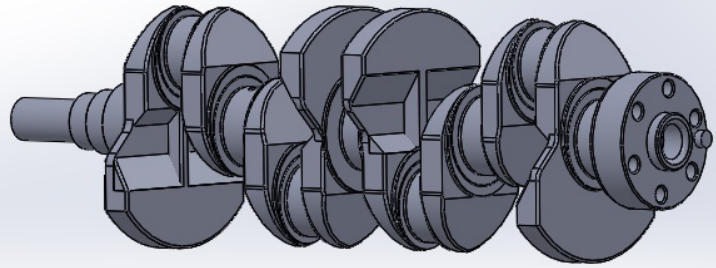


Figure 5.0.1.

Modelling of the block and its components lead to the understanding that a crank offset cannot be applied without drastic changes to the block's design. Because of this the block needed to be redesigned to accommodate

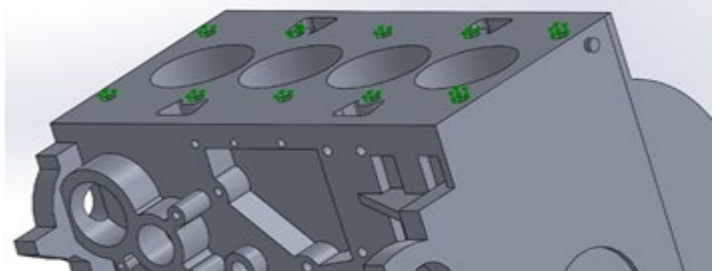
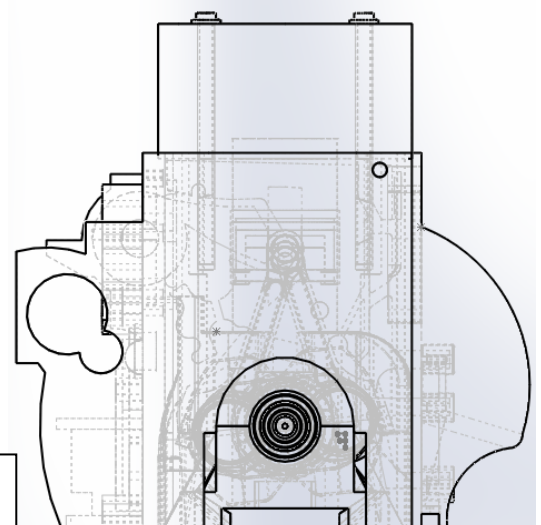
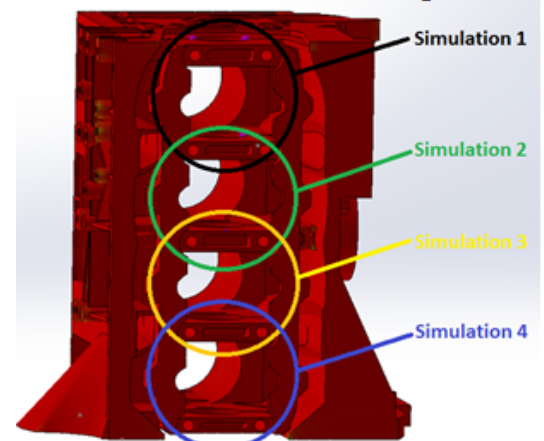


Figure 5.0.2.

the optimum crankshaft offset. It was then subject to simulation and relevant safety factors obtained. The redesigned block and engine components were designed to have the same factors of safety. To begin the block and crank ladder were mated together. Fixtures were applied to each head bolt location to represent the block being pushed away from the cylinder head during combustion in the cylinders (figure 5.0.1.). The forces were then applied to the crank ladder main journal locations at either side of each cylinder. (figure 5.0.2.) Using the maximum net downforce value calculated in the friction model of 26847 N means 13423.5 N is reacted by each side of the crank ladder (figure 5.0.3). Simulating the block for each cylinder individually means the force is only reacted by the crank ladder on either side of the cylinder instead of being spread out across the whole crank ladder which would not be a true representation of real life.



To the right: Image showing assembly of 1.6 Ecoboost engine.

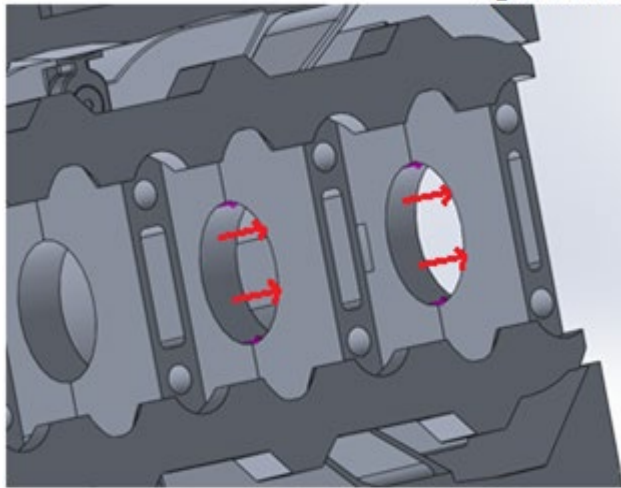


Figure 5.0.3.

A standard mesh size of 20mm is used with a refined mesh of 5mm on the crank ladder and around its fixings on the block. throughout the simulations (Figure 5.0.4.) with aluminium used as the material.

Hyundai motor company (2009) state approximate values for the factor of safety of an engine block in figure 5.0.6. Simulation of the standard 1.6 Ecoboost engine block gave a minimum factor of safety of 1.6 which corresponds nicely.

Screenshots of the simulation results are available in the appendices.

To integrate the 25mm crankshaft offset into the engine block, modifications had to be made to the journal locations, effectively moving them 25mm to the side. This led to the bottom of the oil ways also needing to be moved to the side. Thus, to maintain the original cylinder head mounting points, the oil ways need to be angled. Due to the crankshaft and thus flywheel moving the engine blocks accessories and relevant sensors would also require repositioning. However, by simply altering the accessory mounting points in accordance to the crankshaft movement means the accessories do not need modifying. Such items that will require mounting point repositioning include the crankshaft sensor, starter motor, timing belt, Oil pump and water pump. Items other than the block that will require alteration include the sump, timing belt, oil pump chain and alternator belt.

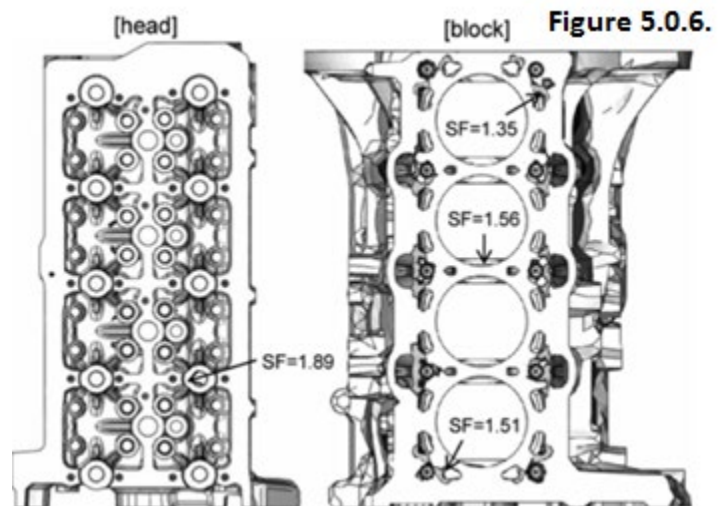
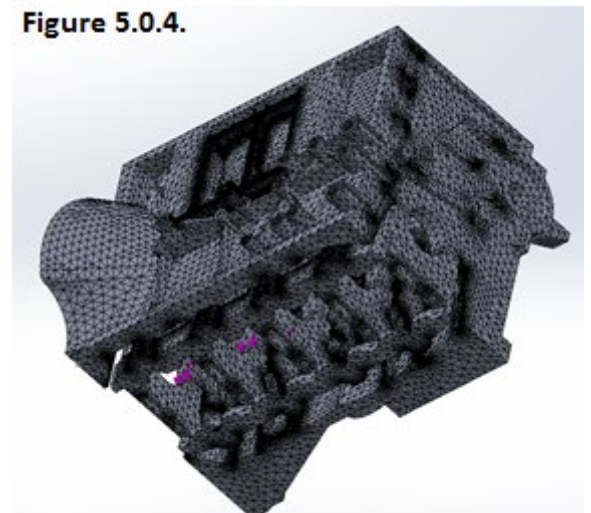
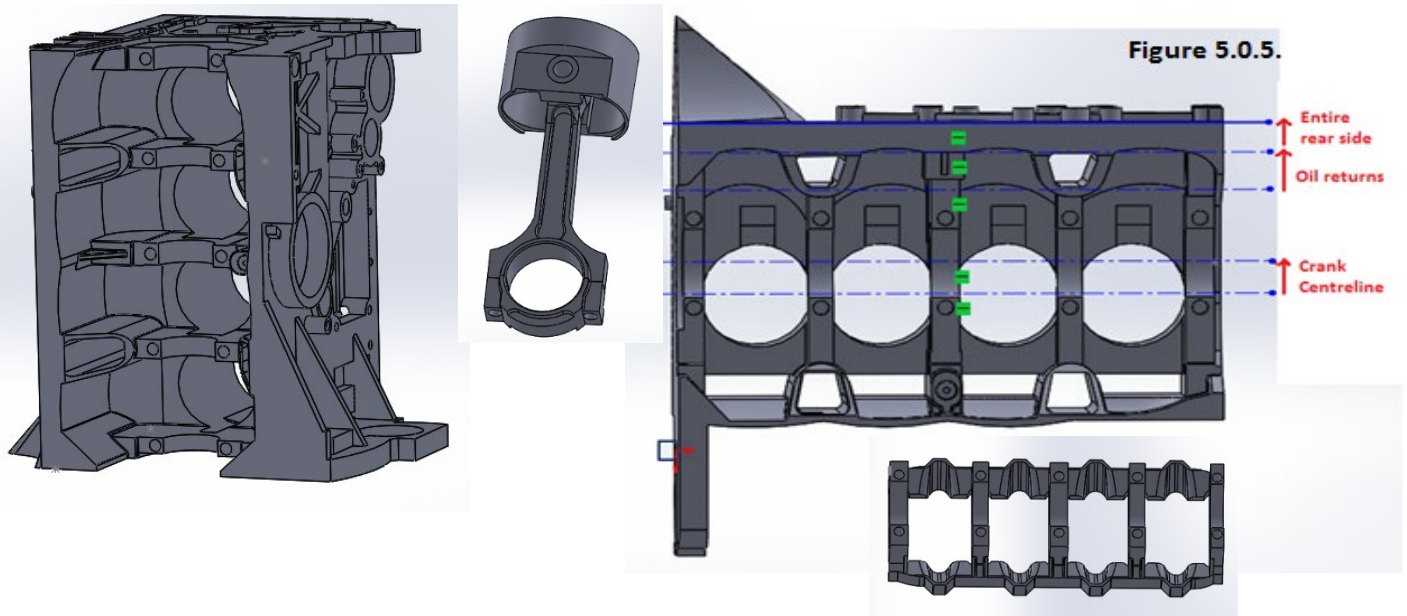


Figure 5.0.6.

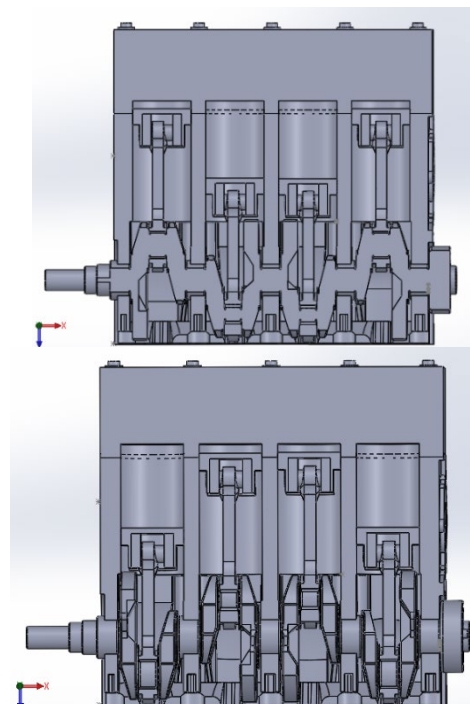
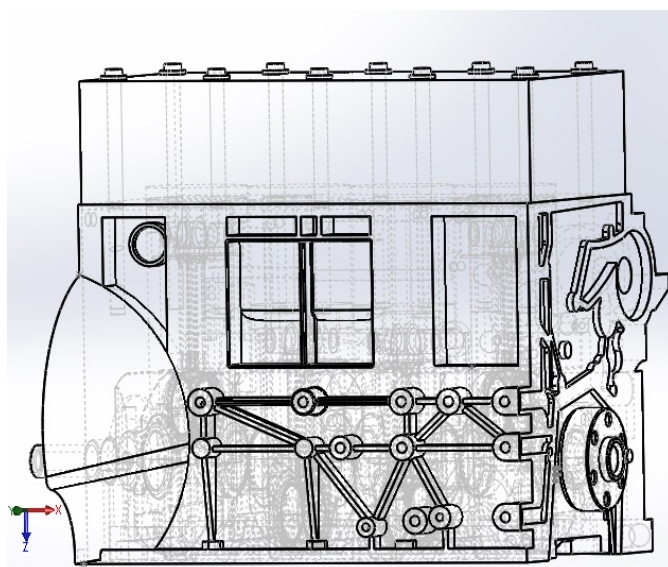
Figure 5.0.4.





Aside from alterations to the block the only other component that would require modification is the sump. This item is of relatively low complexity and so would require minimum effort to alter accordingly. The crankshaft, crank ladder, conrods etc. do not require modification in terms of fitment but may require attention once component balance is reviewed. Changes to the chassis of the vehicle may also be needed due to the thickening of the bottom of the engine. Figure 5.0.5. illustrates the changes that have been made to the block to accommodate the crankshaft offset. Technical drawings of each component are available in the appendices.

Using the same force value and position the modified block was altered in such a way as to retain the original factor of safety. The individual simulations are also included in the appendices.



Chapter 6: Conclusion

In the first part of this report a detailed friction model of the piston skirt is created. This not only shows how the friction of the piston skirt is calculated but also what contributes to it. Analysis of the skirt friction showed how the friction mean effective pressure is higher for turbo engines with the 1.0 Fox generating the most friction. It also showed the skirt FMEP as a proportion of the IMEP with the 1.0 Fox having the highest IMEP. Knowing the 1.0 Fox engine uses a crankshaft offset of 8mm, the friction generated once this offset was applied showed a significant reduction in friction.

Chapter 2: demonstrates how a single value used for the coefficient of friction does not represent real world conditions and thus requires a method of relating it to the piston velocity and side force. The literature review carried out into the matter concluded that currently there is no mathematical way of doing so and that the common idea of calculating the duty parameter and relating it to the coefficient of friction does not correspond when data from a fired engine is used. Clearly this is a subject that requires further research as a mathematical method of predicting the coefficient of friction for a reciprocating component would not only reduce practical testing costs but also allow a more extensive research into ways of reducing the friction.

In order to calculate the coefficient of friction change with engine speed the total friction of an engine was then calculated using an engine's torque data. This allowed comparison with the friction calculated for each individual component. As expected the value of 0.1 used for the coefficient of friction was too high and had to be drastically reduced in order to match that of the total engine friction. Obviously, the values stated at each engine speed are not completely accurate as they change with each crankshaft degree but they do give an insight into the approximate range of the coefficient of friction throughout the engine speed range. The calculated values correspond with (*Phil Carden: Calculation of Friction in High Performance Engines, 2010*) who states estimated values for the different regimes of lubrication with boundary occurring around 0.1 and hydrodynamic around 0.005. (*J. Halling, Principles of Tribology, 1987 Pg. 6*) then states that both mixed lubrication and hydrodynamic lubrication (where increasing speeds cause increased friction) have been observed in the mid stroke region, depending on engine speed and load.

Once the Component friction was made to match that of the total engine friction the friction models were adjusted to incorporate a crankshaft offset. This allowed the potential friction reduction of the crankshaft offset to be calculated and an optimum value found. This value was then incorporated into a 1.6 Ecoboost engine to show the extent of modifications required.

Comparing the Total friction of the standard engine to that of the modified engine incorporating the corrected coefficient of friction correction yields a maximum friction reduction of around 5% at 25mm of crank offset. This value is far higher than that currently being used by manufacturers and suggests there is still room for improvement in reducing the piston skirt friction, with the engine design conducted within Solidworks suggesting that only modifications to the block would be necessary. The reason for this reduction in friction comes from the side force. The positive crankshaft offset results in an increase in force during the compression stroke and a decrease during the expansion stroke. The decrease during the expansion stroke, however, is larger in magnitude, longer in duration, and occurs at a higher piston speed: all resulting in a net reduction in piston side force due to the offset.

Other benefits of the crank offset that are not evident in the friction model, according to Toyota, include a possible 3% improvement in fuel economy due to changes in combustion performance and friction levels. It is thought that combustion is improved due to reduction of time loss. Increased crank offset means the volume to crank angle profile is altered in such a way that the piston dwells nearer top dead centre for longer thus giving it a nearer constant volume combustion event, which in theory helps to increase cylinder pressures during early expansion and improves heat release. Another possible advantage of offsetting the crankshaft was found by Musashi Institute of Technology, who measured piston motion and found that the piston skirt tilt angle during compression was reduced due to the offset, allowing improved oil film thickness development and lowering piston friction. However, there are several factors that have been overlooked in this report which may lead to friction increasing in other areas of the engine. In the case of a well-lubricated piston skirt and gudgeon pin, analysis suggests that friction could increase if the crankshaft were offset, due to an increased moment about the pin that reduces oil film development at the skirt. Also in regards to the piston pin two other factors need to be considered: One being the moment generated by the combustion force acting on the piston surface and pin offset and the other being the friction of the piston pin itself. Both these factors influence the magnitude of the moment about the pin in which a greater moment is detrimental as it increases piston tilt, thus reducing hydrodynamic film development. The reduction in friction from offsetting the crankshaft then is determined by the net result of both the reduction in side force and the moment about the piston pin. Thus in order to clearly understand the friction reduction the piston pin would need to be examined. This could be the subject of further work.

Chapter 7: Recommended further work

For the friction models to be developed any further it would be necessary to accurately measure the coefficient of friction at each crankshaft degree of rotation. As there is no mathematical method of relating them it would be necessary to develop a reciprocating test rig which operates under the same conditions as the real engine. Unfortunately it seems currently there is no way of measuring the data at such high speeds and loads. For this reason the general trend is to use a motored test rig, details of which will now be discussed.

A direct motoring of the engine is conducted under conditions as close as possible to firing. Engine temperatures are maintained as close to normal operating temperatures as possible. This can be done either by heating the water and oil flows or by conducting a “grab” motoring test where the engine is switched rapidly from firing to motored operation. The power required to motor the engine includes the pumping work. Motoring tests on a progressively disassembled engine can be used to identify the contribution that each major component of the engine makes to the total frictional losses. However, these tests are still fairly inaccurate with motoring losses being different from the firing losses for the following reasons. Only the compression pressure and not the firing pressure acts on the piston, piston rings and bearings. The lower gas loads during motoring mean lower rubbing friction. Piston and cylinder bore temperatures are lower in motored operation, meaning a greater viscosity of the lubricant and therefore increased friction. Piston cylinder clearances are greater during motoring operation which tends to make friction lower. However, in firing operation the lubrication of the top ring near the top of the stroke is inadequate to maintain normal hydrodynamic lubrication with the higher gas pressures behind the ring. The resulting boundary friction in this region makes friction in the firing engine higher. In motored operation, the exhaust blowdown phase is missing and the gases discharged later in the exhaust stroke have a higher density than under firing conditions. These effects can result in different pumping work. When motoring, net-work is done during the compression and expansion strokes because of heat loss from the gas to the walls, and because of gas loss through blow by. This work is not part of the true total friction work in a firing engine and should not be deducted from the indicated work of the firing engine to obtain the brake work; heat losses and blow by are additional energy transfers to the indicated work, friction work and brake work. (*John B. Haywood book, Internal Combustion Engine Fundamentals, 1998, Pg. 719*)

In addition to the reciprocating test rig it would also be beneficial to research the piston pin friction and how the moment about the pin influences the friction. Along with additional research into accessory friction and continued development of their models.

Chapter 8: Appendix

8.1. Derivation of method to convert acceleration in m/degree/degree to metres per second squared.

If rpm = rpm

Number of revs/second = rpm/60

Number of degrees/second = $360 \times \text{rpm}/60 = 6.\text{rpm}$

If $v = v$ mm/degree

Number of degrees per second = $6.\text{rpm}$

Therefore V in mm/s = $6.\text{rpm} \times V$ (mm/degree)

If $a = a$ (mm/degree/degree)

There is $6.\text{rpm}$ in a second

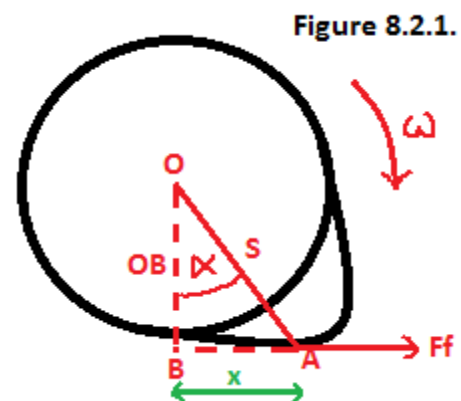
Therefore in $\text{mm/second}^2 = 6.\text{rpm} \times 6.\text{rpm} \times a$ (mm/degree/degree)

$= (6.\text{rpm})^2 \times a$ (mm/degree/degree)

8.2. Additional method to calculate 'x' (distance lobe has moved)

From figure 8.2.1. and using trigonometry it is evident that $x = OB \cdot \tan \alpha$ and $\alpha = \{180 - (\theta + \phi)\}$

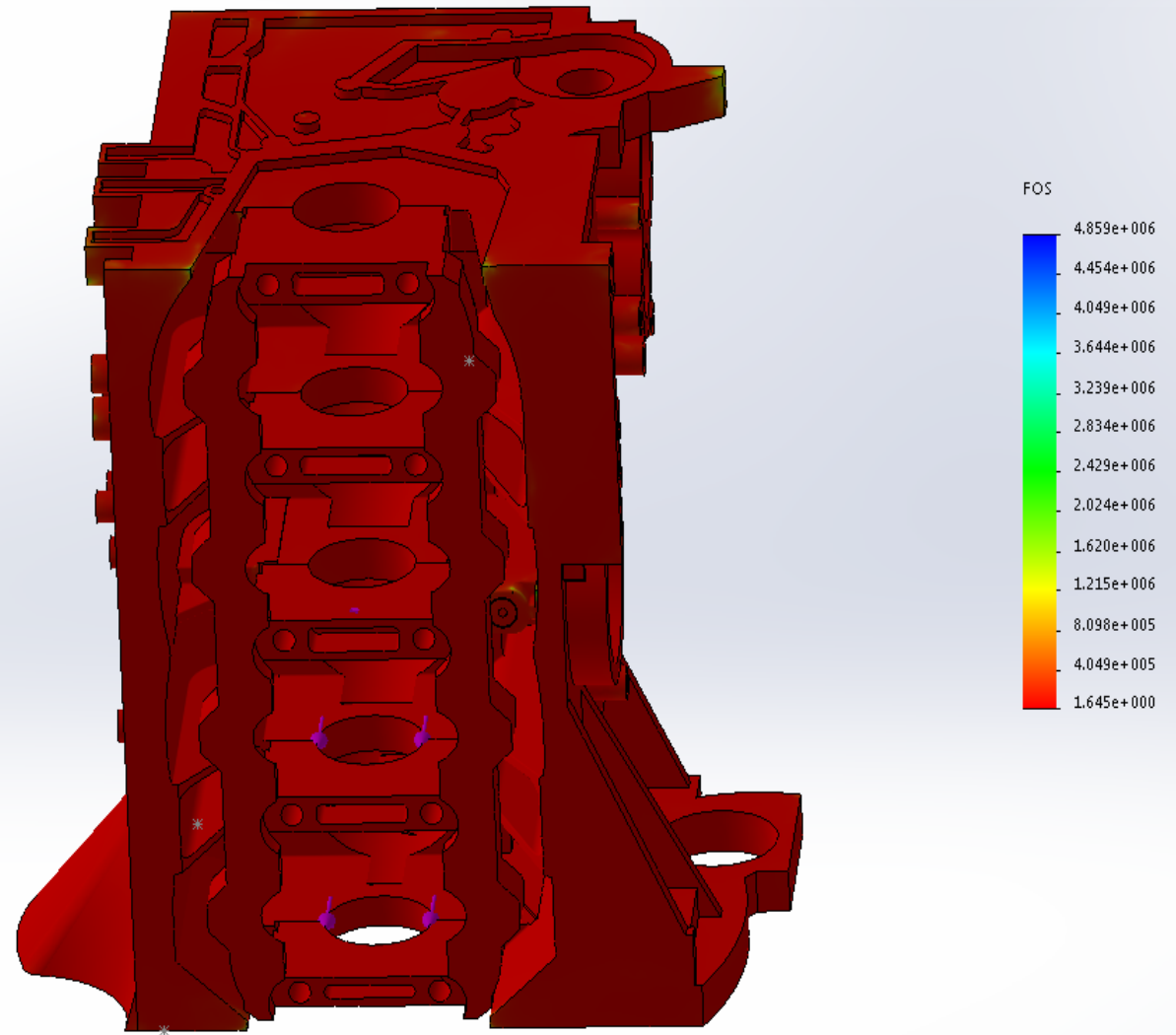
Finding the cam angle (ϕ) and angle from the nose (θ) for each corresponding OB value starting from the contact point thus allows calculation of 'x'. The reason for not choosing this method is due to the layout of the spreadsheet which would mean finding the corresponding values by scrolling through the matrix each time which would lead to large amounts of time being wasted.



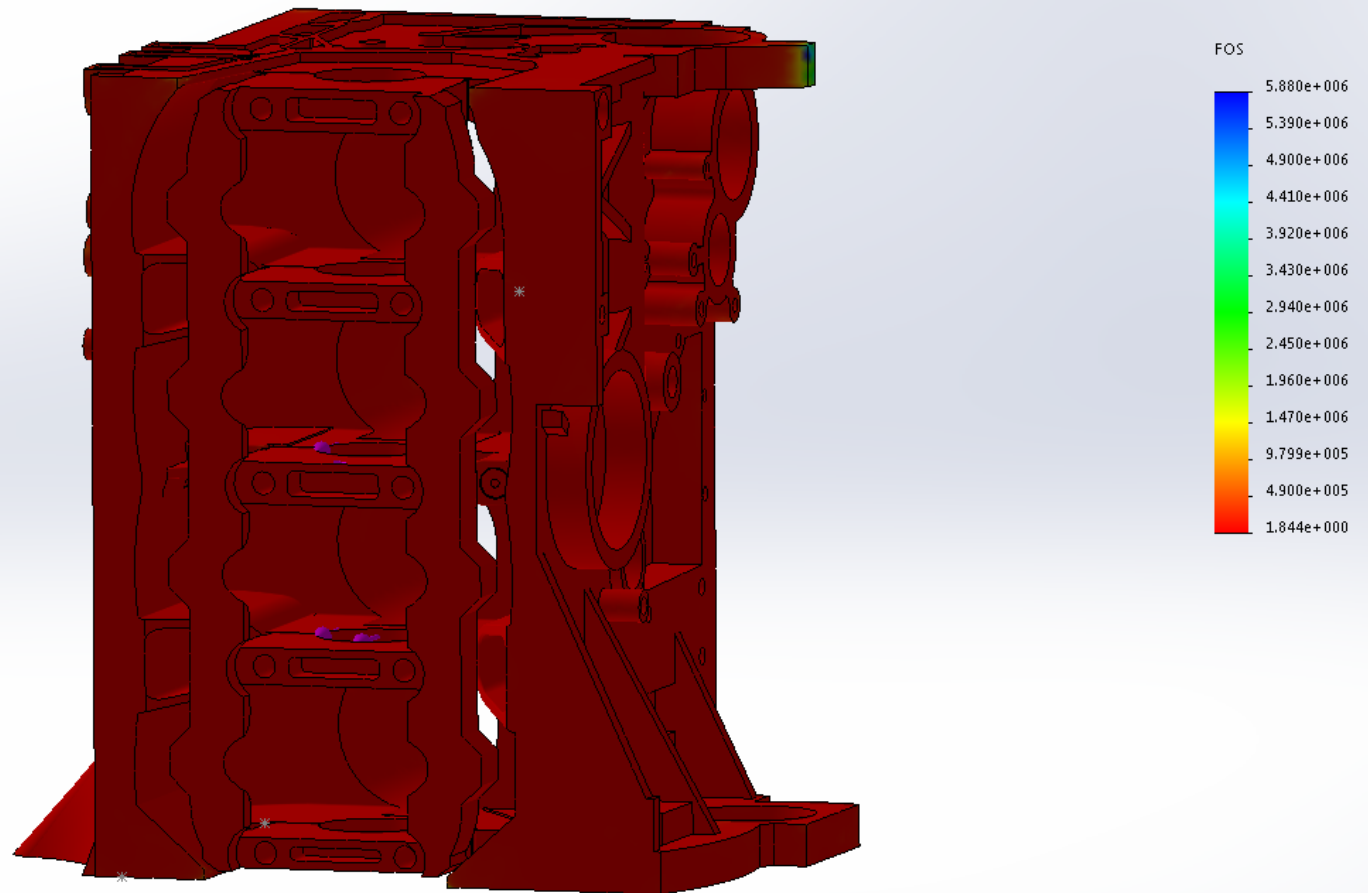
The bearings are in predominantly hydrodynamic lubrication, in contrast to the valvetrain oscillating components, which are characterized to be mostly in the mixed/boundary lubrication regimes.

8.3. Simulation of Block

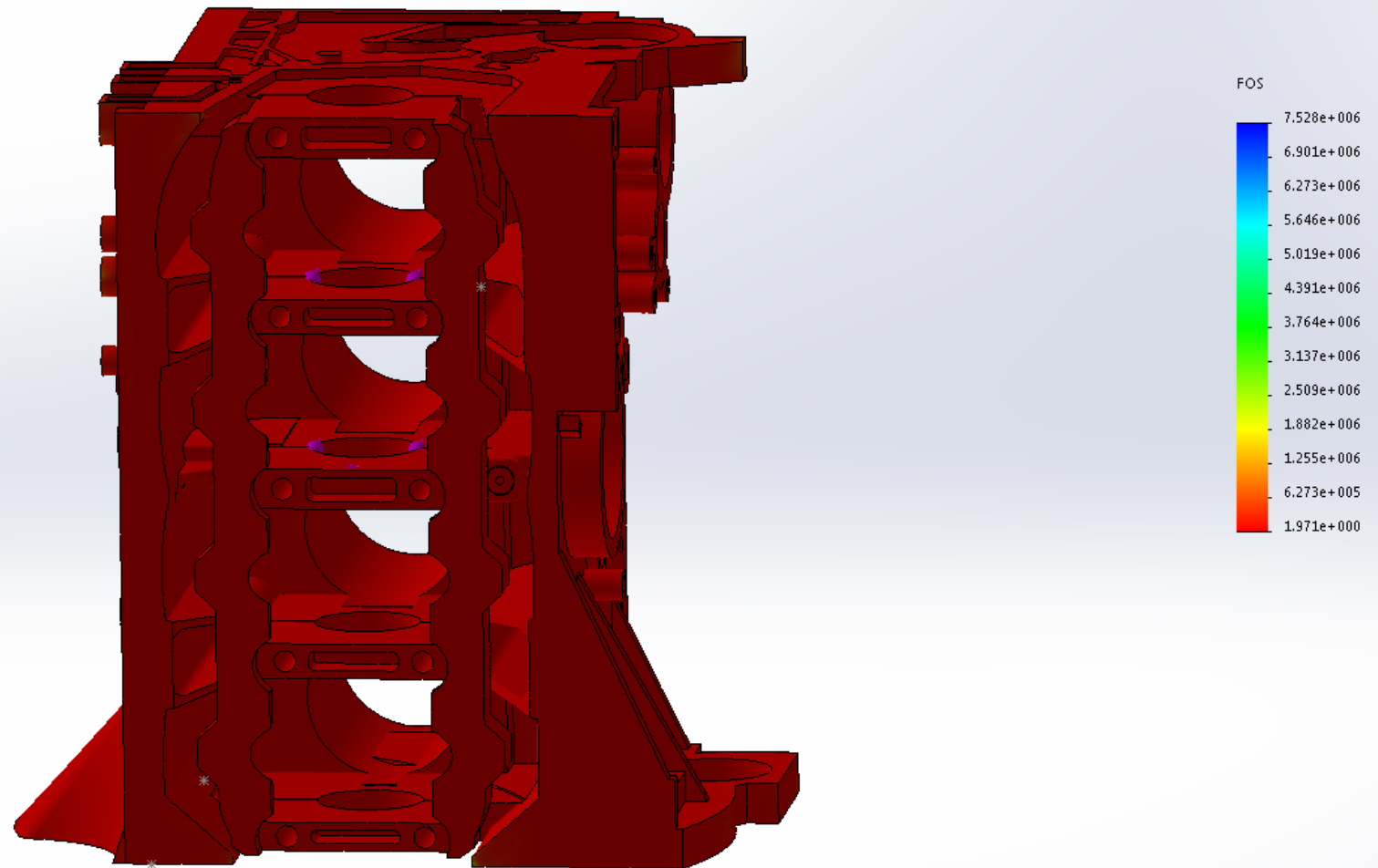
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Criterion : Automatic
Factor of safety distribution: Min FOS = 1.6



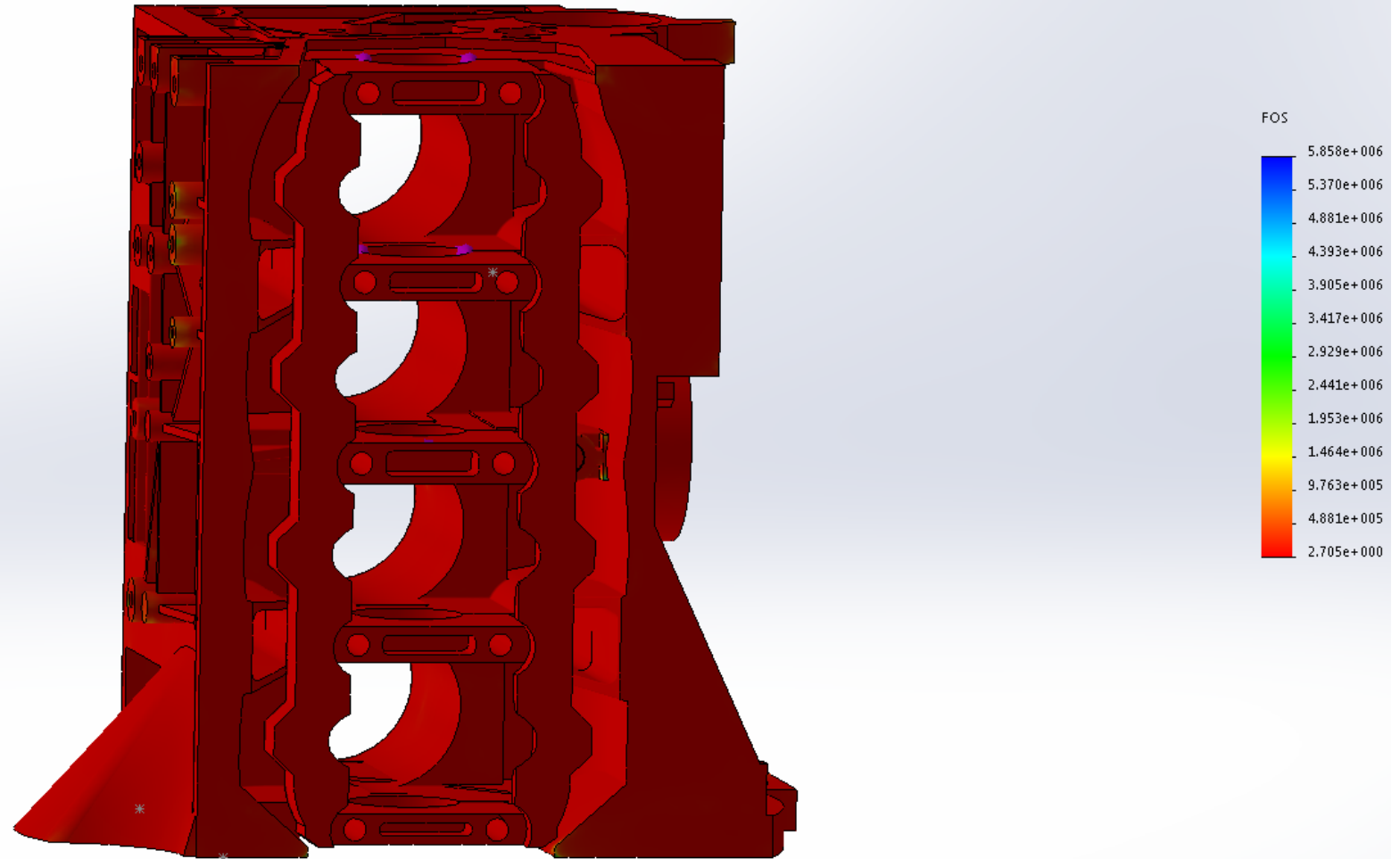
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Factor of safety distribution: Min FOS = 1.8



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Criterion : Automatic
Factor of safety distribution: Min FOS = 2



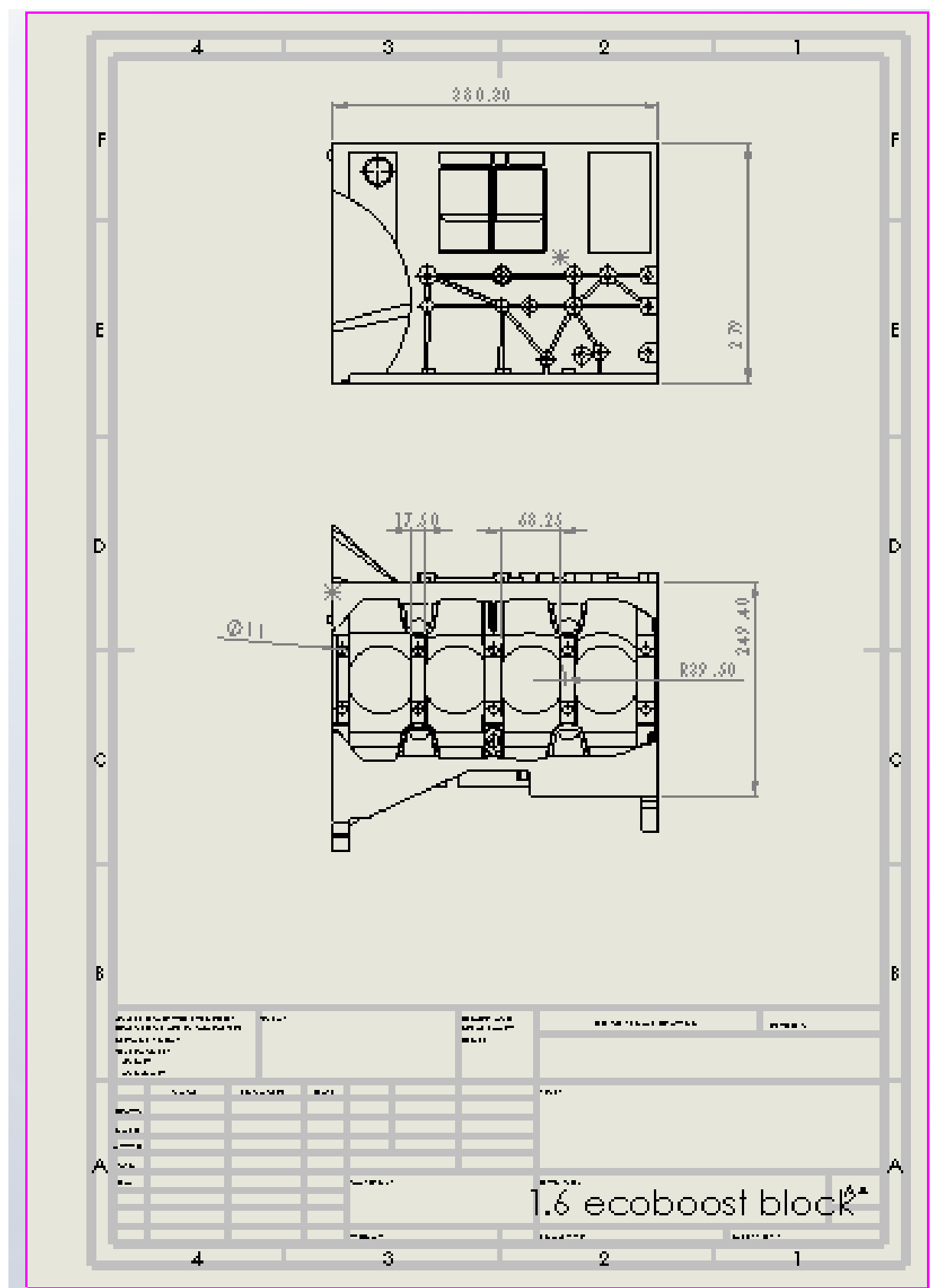
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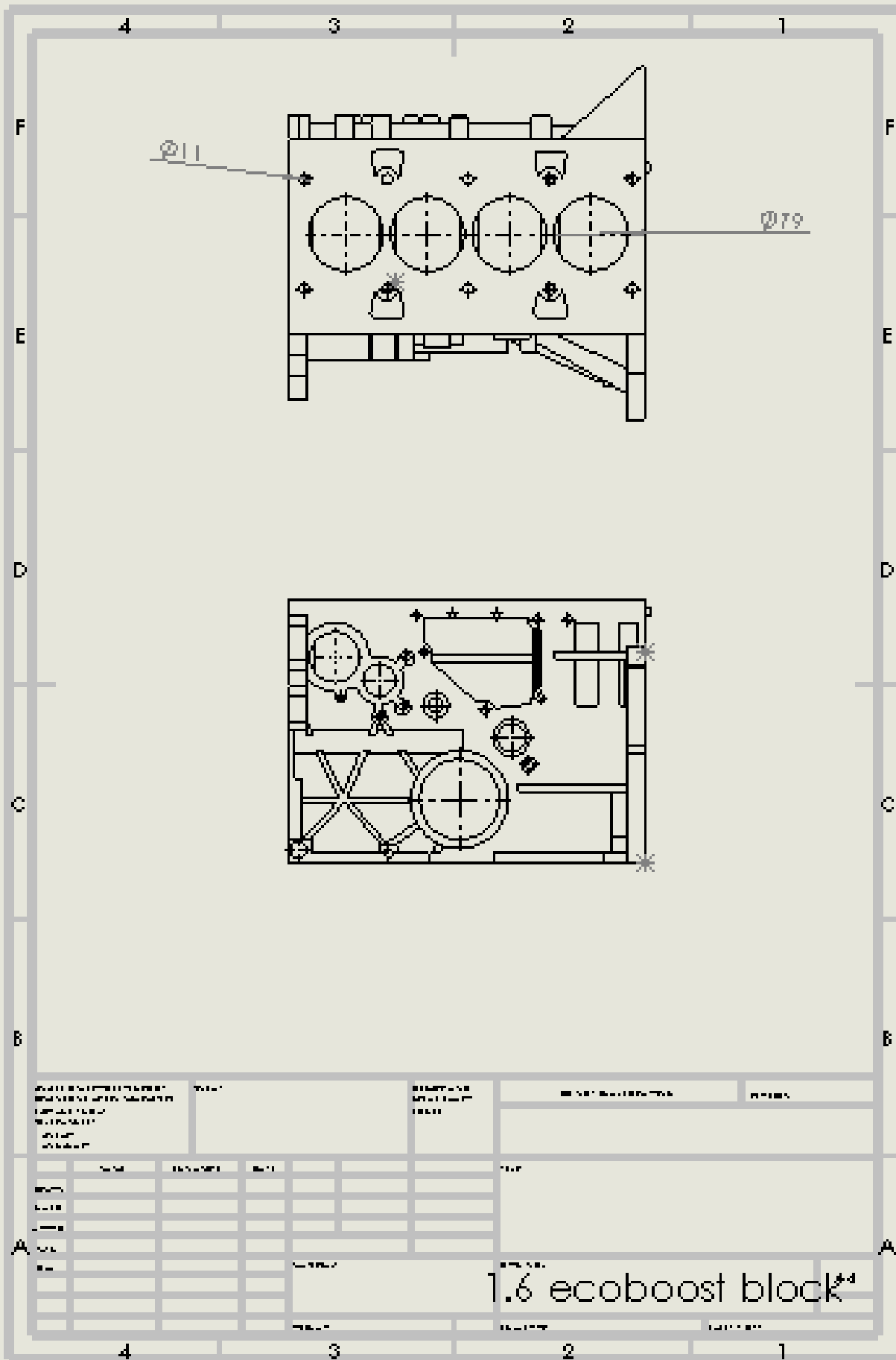


8.4. Simulation of Modified Block

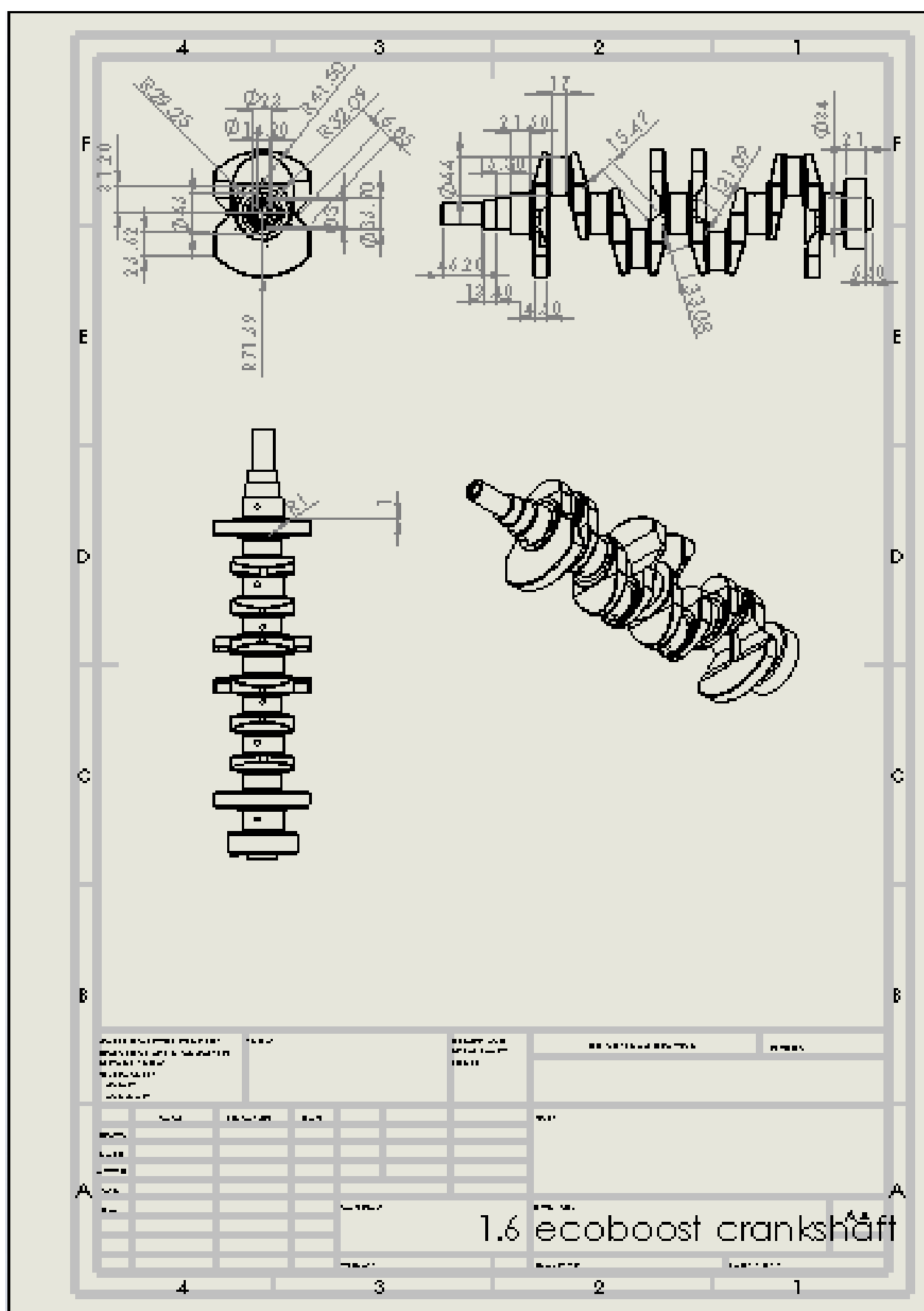
8.5. Technical drawings

8.5.1. – Standard Engine Block

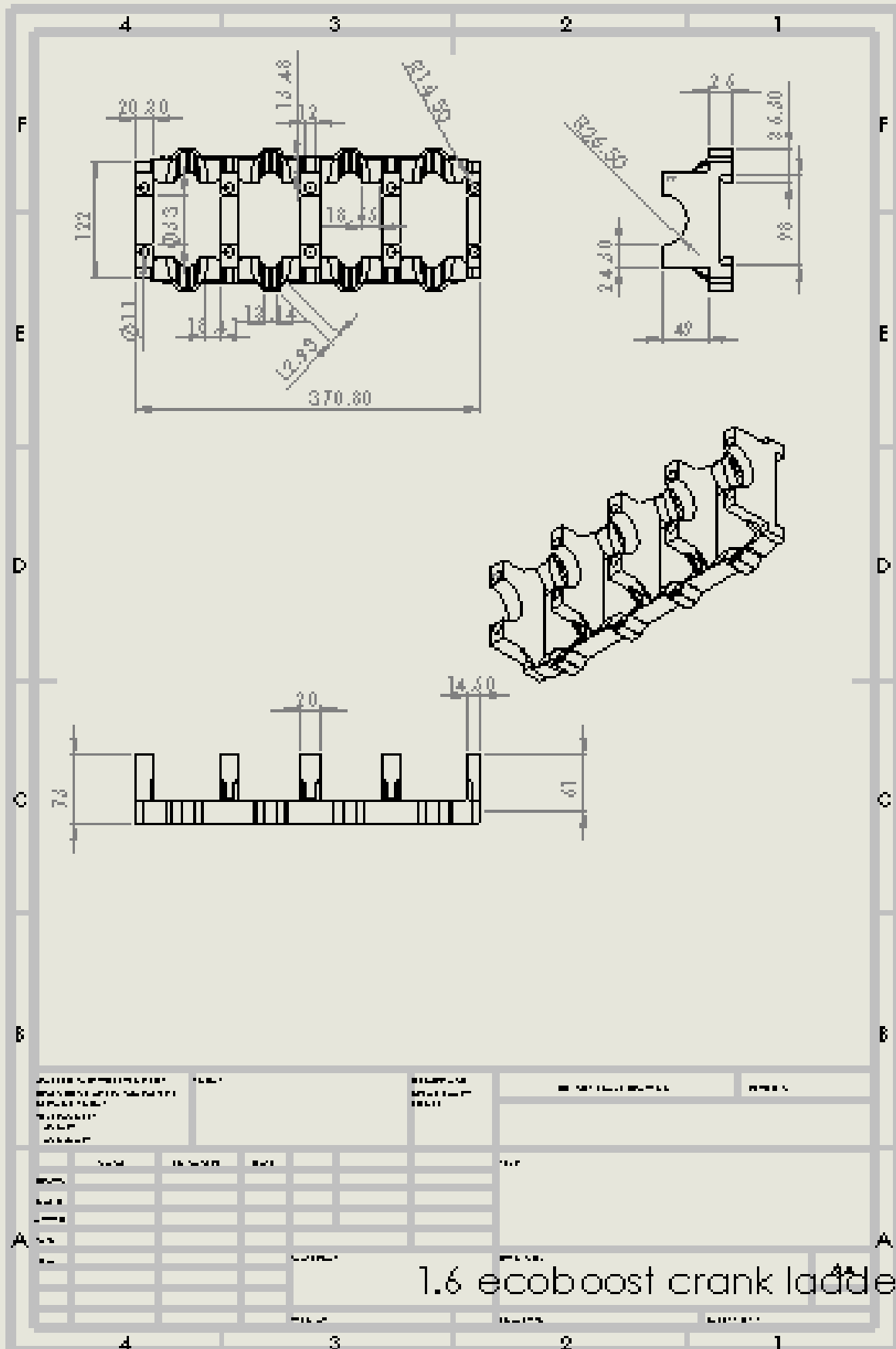




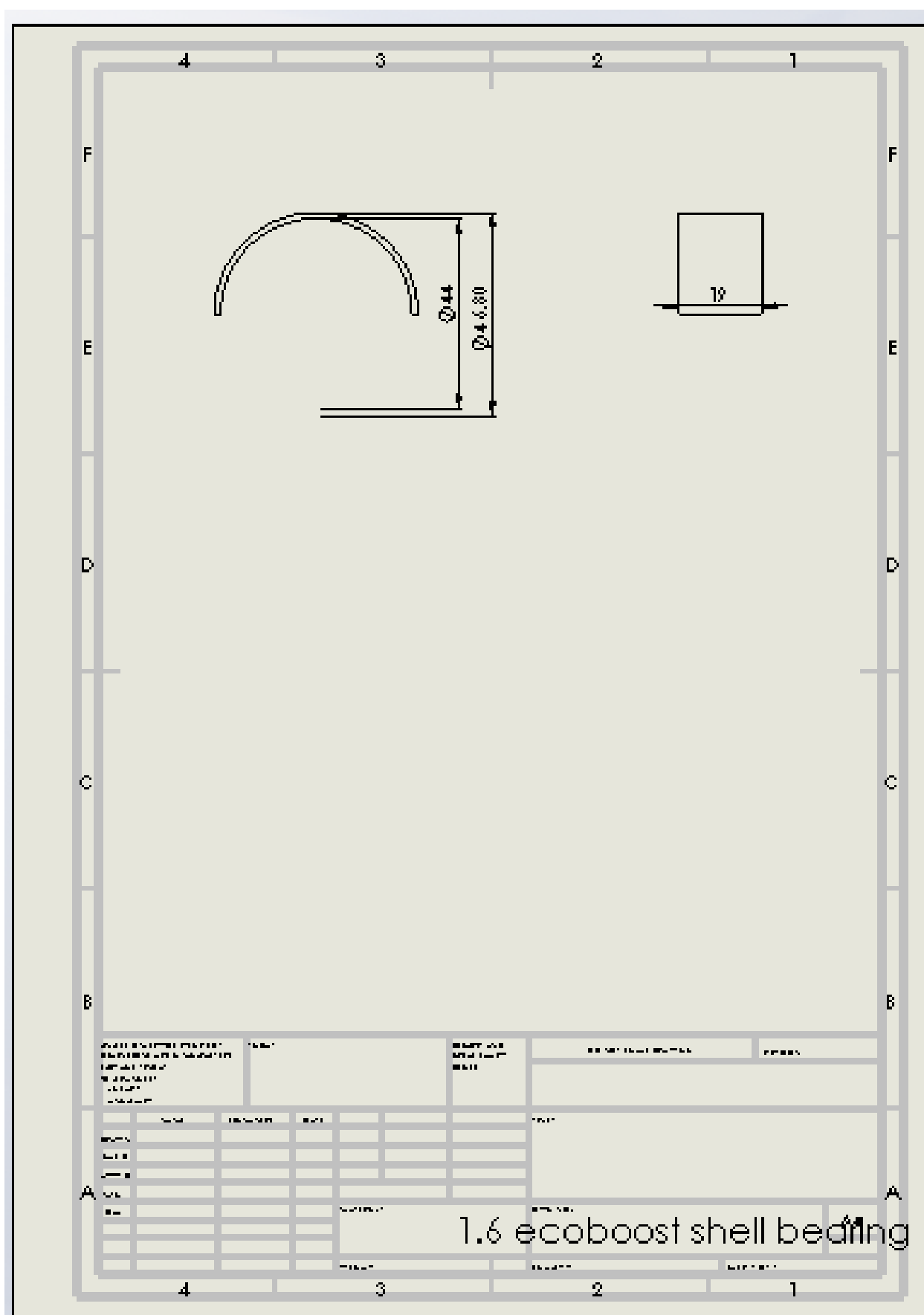
8.5.3. Crankshaft



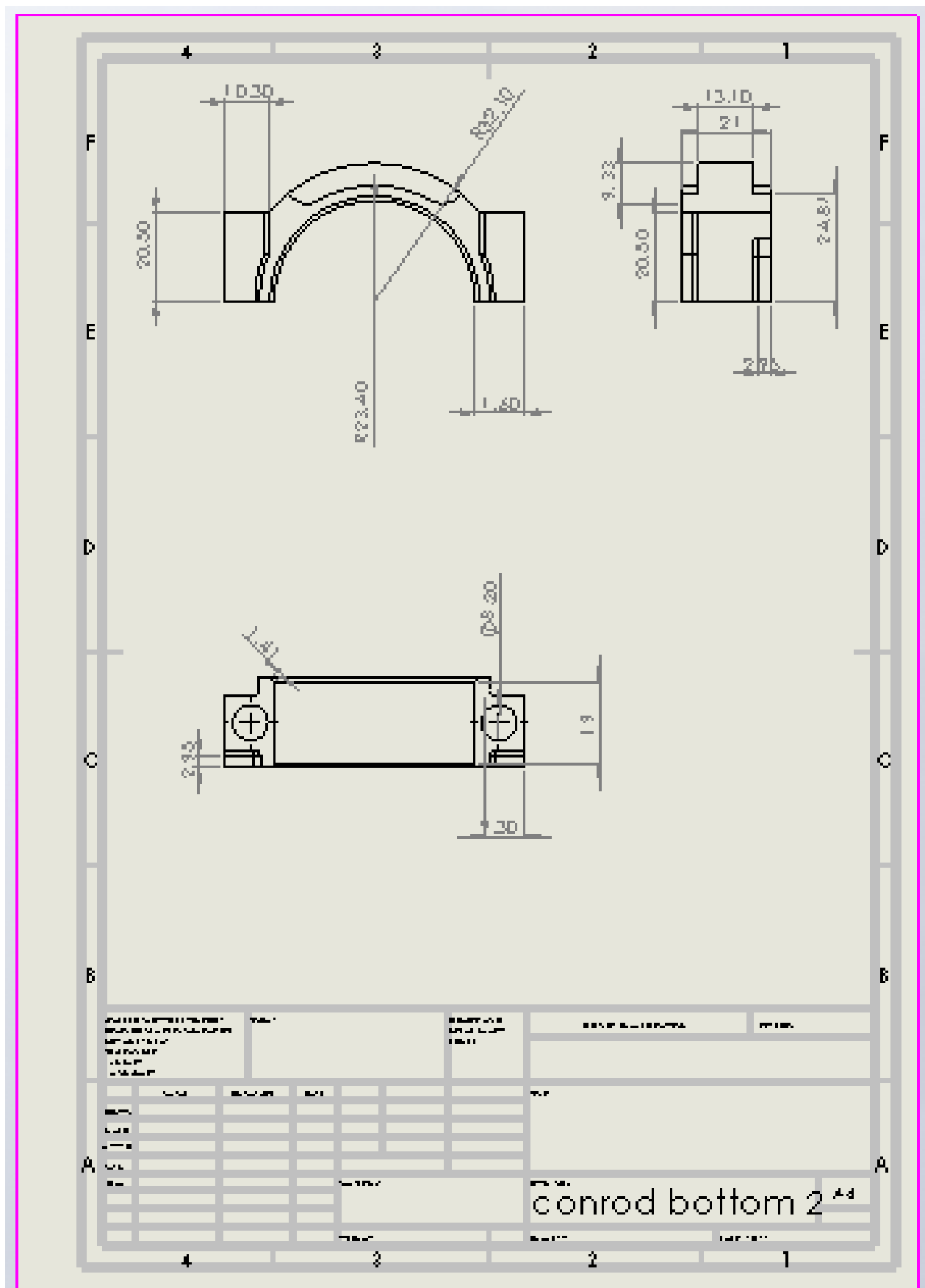
8.5.4. Crank Ladder



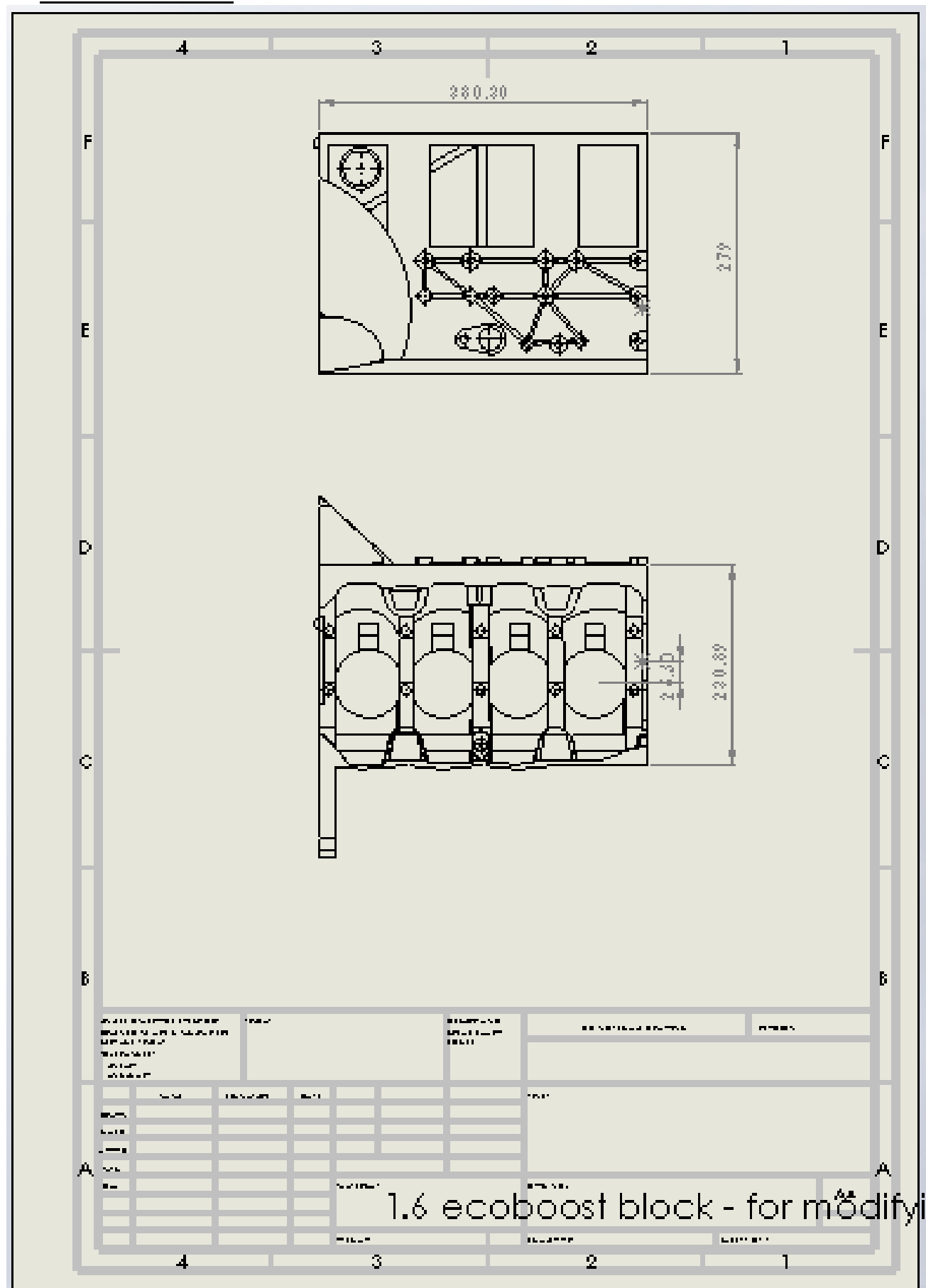
8.5.5. Shell Bearing



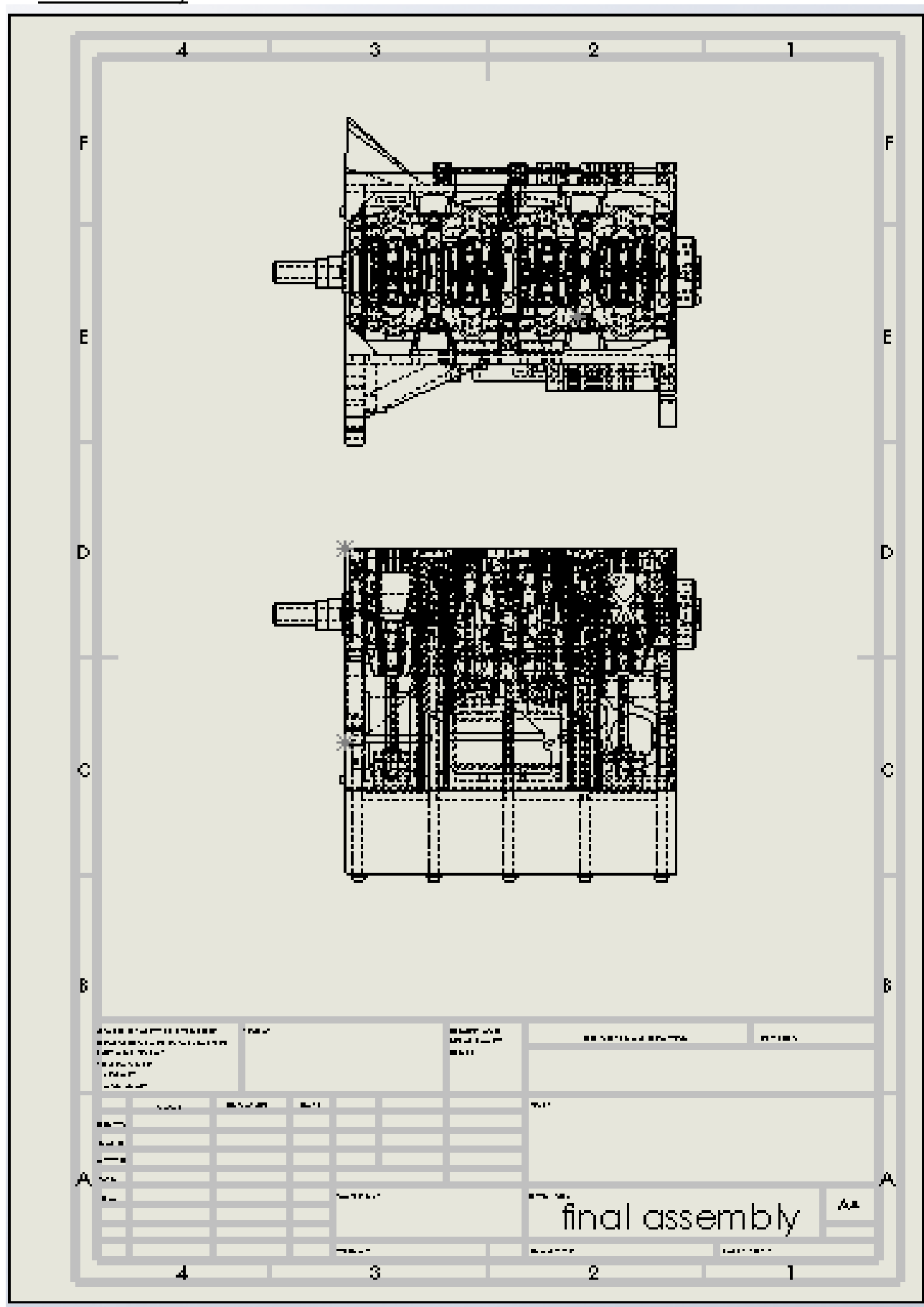
8.5.6. Conrod-Cap

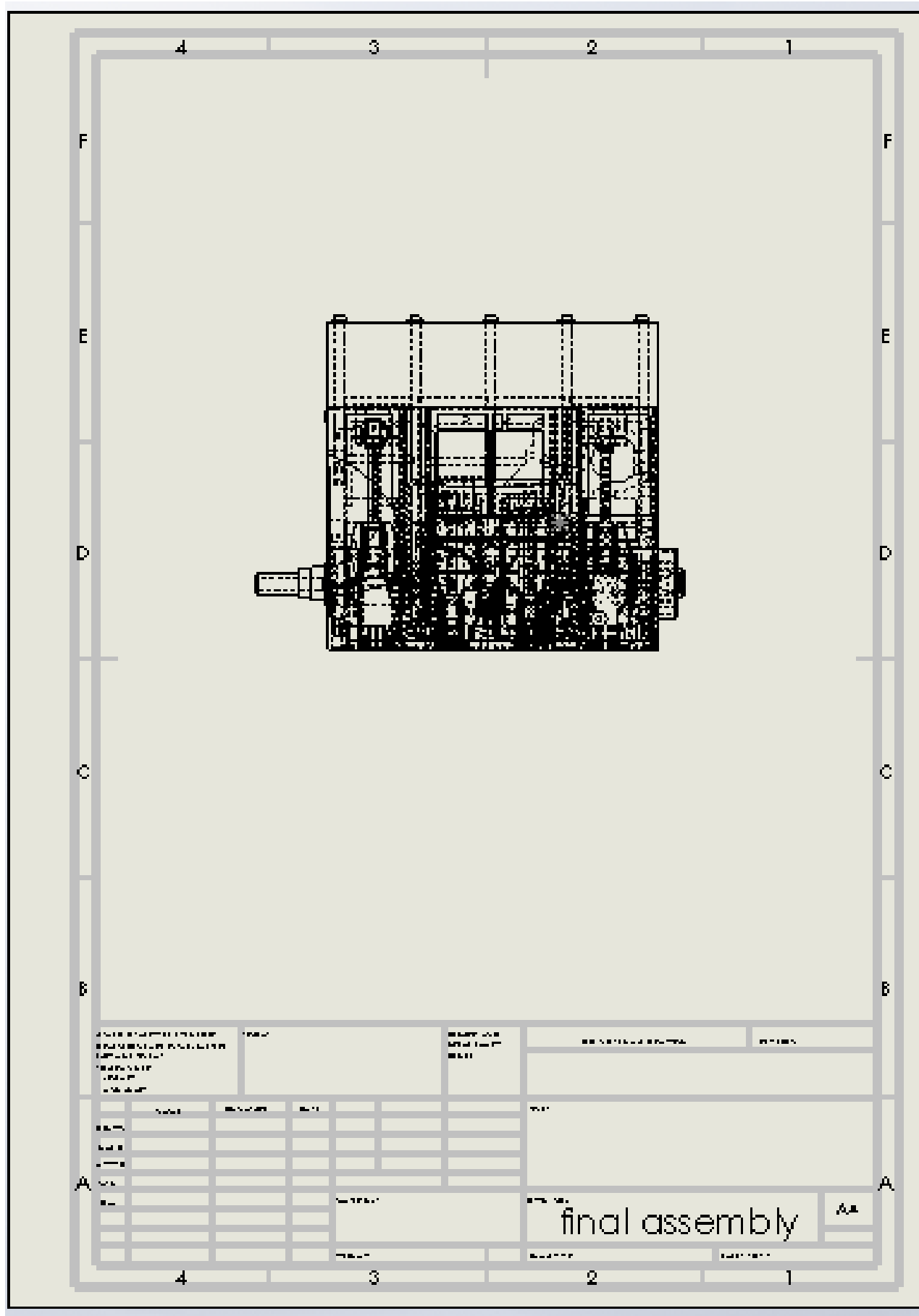


8.5.7. Modified Block



8.5.8. Assembly





8.6. Spreadsheets

Piston skirt Friction Model

	A	B	C	D	E	F	G	H	I	J	K	L	M	N	O	P	Q	
1	Engine Speed:991.9					44.4725												
2						11.4369												
3	Crank Angle	CYLPR1	CYLPR2	CYLPR3	CYLPR4	AVRG P	AVRG P	Piston Offset	Crank Offset	Crank angle @ TDC	Corrected crank angle	Corrected crank angle	HTDC	S new	Conrod angle new	Corrected conrod angle	Fside New	R.C
4	deg	bar	bar	bar	bar	bar	Pa	x1 = m	x2 = m	Rad	Radians	Degrees	m	m	Radians	Degrees	N	
5	-360	1.132	1.234	0.992	1.084	1.1105	111050	0	0.04	0.230610011	-6.053389989	-346.8336981	0.1704	3.25494E-05	-0.230859314	-13.22726436	-50.32509222	
6	-359	1.132	1.234	0.992	1.084	1.1105	111050	0	0.04	0.230610011	-6.035934433	-345.8335684	0.1704	4.01372E-05	-0.225531591	-12.92200832	-49.13553635	
7	-358	1.12	1.234	0.992	1.084	1.1075	110750	0	0.04	0.230610011	-6.018478878	-344.8334388	0.1704	6.44192E-05	-0.220233753	-12.61846458	-47.70958527	
8	-357	1.132	1.234	0.992	1.097	1.11375	111375	0	0.04	0.230610011	-6.001023322	-343.8333091	0.1704	0.000105321	-0.214967156	-12.3167108	-47.15148098	
9	-356	1.132	1.234	0.992	1.097	1.11375	111375	0	0.04	0.230610011	-5.983567766	-342.8331795	0.1704	0.000162762	-0.209733147	-12.01682416	-46.05794723	
10	-355	1.145	1.26	1.005	1.11	1.13	113000	0	0.04	0.230610011	-5.966112211	-341.8330498	0.1704	0.000236653	-0.204533065	-11.71888137	-46.36376046	
11	-354	1.145	1.26	1.017	1.122	1.136	113600	0	0.04	0.230610011	-5.948656655	-340.8329201	0.1704	0.000326896	-0.199368239	-11.42295866	-45.78672694	
12	-353	1.158	1.273	1.03	1.122	1.14575	114575	0	0.04	0.230610011	-5.9312011	-339.8327905	0.1704	0.000433389	-0.194239992	-11.12913178	-45.51077275	
13	-352	1.171	1.273	1.03	1.135	1.15225	115225	0	0.04	0.230610011	-5.913745544	-338.8326608	0.1704	0.00055602	-0.189149638	-10.83747597	-44.96633089	
14	-351	1.171	1.286	1.043	1.148	1.162	116200	0	0.04	0.230610011	-5.896289989	-337.8325311	0.1704	0.000694673	-0.184098482	-10.54806601	-44.66440938	
15	-350	1.184	1.286	1.043	1.148	1.16525	116525	0	0.04	0.230610011	-5.878834433	-336.8324015	0.1704	0.000849222	-0.179087818	-10.26097616	-43.86665365	
16	-349	1.184	1.299	1.043	1.161	1.17175	117175	0	0.04	0.230610011	-5.861378878	-335.8322718	0.1704	0.001019536	-0.174118936	-9.976280178	-43.30895799	
17	-348	1.197	1.299	1.043	1.174	1.17825	117825	0	0.04	0.230610011	-5.843923322	-334.8321422	0.1704	0.001205477	-0.169193113	-9.694051319	-42.74446862	
18	-347	1.184	1.299	1.043	1.161	1.17175	117175	0	0.04	0.230610011	-5.826467766	-333.8320125	0.1704	0.001406901	-0.164311619	-9.414362312	-41.29696012	
19	-346	1.184	1.286	1.043	1.161	1.1685	116850	0	0.04	0.230610011	-5.809012211	-332.8318828	0.1704	0.001623657	-0.159475714	-9.137285365	-40.10515821	
20	-345	1.171	1.273	1.03	1.148	1.1555	115550	0	0.04	0.230610011	-5.791556655	-331.8317532	0.1704	0.001855587	-0.154686649	-8.862892147	-38.32475668	
21	-344	1.171	1.26	1.017	1.122	1.1425	114250	0	0.04	0.230610011	-5.7741011	-330.8316235	0.1704	0.00210253	-0.149945665	-8.591253783	-36.60984048	
22	-343	1.158	1.247	1.017	1.11	1.133	113300	0	0.04	0.230610011	-5.756645544	-329.8314938	0.1704	0.002364314	-0.145253994	-8.322440841	-35.16605982	
23	-342	1.145	1.234	1.005	1.097	1.12025	112025	0	0.04	0.230610011	-5.739189989	-328.8313642	0.1704	0.002640767	-0.140612858	-8.056523326	-33.58313321	
24	-341	1.132	1.222	1.005	1.084	1.11075	111075	0	0.04	0.230610011	-5.721734433	-327.8312345	0.1704	0.002931706	-0.136023468	-7.793570659	-32.23889536	
25	-340	1.12	1.209	1.005	1.084	1.1045	110450	0	0.04	0.230610011	-5.704278878	-326.8311049	0.1704	0.003236946	-0.131487026	-7.533651675	-31.11299791	
26	-339	1.12	1.209	1.005	1.071	1.10125	110125	0	0.04	0.230610011	-5.686823322	-325.8309752	0.1704	0.003556296	-0.127004723	-7.276834601	-30.17266056	
27	-338	1.12	1.222	1.005	1.071	1.1045	110450	0	0.04	0.230610011	-5.669367766	-324.8308455	0.1704	0.003889558	-0.122577738	-7.023187048	-29.57503497	
28	-337	1.12	1.222	1.017	1.071	1.1075	110750	0	0.04	0.230610011	-5.651912211	-323.8307159	0.1704	0.00423653	-0.118207241	-6.772775993	-28.95835965	
29	-336	1.12	1.234	1.017	1.084	1.11375	111375	0	0.04	0.230610011	-5.634456655	-322.8305862	0.1704	0.004597007	-0.113894388	-6.525667764	-28.48588928	
30	-335	1.132	1.234	1.03	1.097	1.12325	112325	0	0.04	0.230610011	-5.6170011	-321.8304565	0.1704	0.004970776	-0.109640327	-6.281928025	-28.14000566	
31	-334	1.145	1.247	1.03	1.11	1.133	113300	0	0.04	0.230610011	-5.599545544	-320.8303269	0.1704	0.005357622	-0.105446192	-6.04162176	-27.77476842	
32	-333	1.145	1.247	1.03	1.122	1.136	113600	0	0.04	0.230610011	-5.582089989	-319.8301972	0.1704	0.005757324	-0.101313104	-5.804813252	-27.10009393	
33	-332	1.158	1.247	1.03	1.122	1.13925	113925	0	0.04	0.230610011	-5.564634433	-318.8300676	0.1704	0.006169658	-0.097242472	-5.571568069	-26.42733806	
	SGDI	Sheet1	1000rpm	1500rpm	1700rpm	2000rpm	2500rpm	3000rpm	3500rpm	4000rpm	4500rpm	5000rpm	5500rpm	5500rpm	5500rpm	5500rpm	5500rpm	5500rpm

	R	S	T	U	V	W	X	Y	Z	AA	AB	AC	AD	AE	AF	AG
1					rpm	358.5639651										
2					1000	41.56212603								0.000106413		
3	R.Cosθ corrected	L.Cosφ Corrected	Velocity per degree	Acceleration per degree per degree	Acceleration	Finertia Corrected	Frod	Torque	Mean Torque	Velocity	Dynamic Viscosity	Duty parameter	Load	Corrected Duty Parameter		
4			metres per degree	metres per degree per degree	ms ²	N	N	Nm	Nm	m/sec	Pas.s		N/m		Bore Area	C.A.(RAD)
5	0.039922235	0.130444998	7.58774E-06	1.66943E-05	600.995639	244.0042295	219.939	-0.00959	39.8061457	0.04553	0.011619	-4.20444E-07	-1258.13	4.20444E-07	0.00406	-6.284
6	0.039753145	0.130606499	2.42821E-05	1.66202E-05	598.326823	242.92069	219.723	0.19565		0.14569	0.011619	-1.37807E-06	-1228.39	1.37807E-06	0.00406	-6.26654444
7	0.039571943	0.130763419	4.09023E-05	1.65385E-05	595.387222	241.7272119	218.393	0.39808		0.24541	0.011619	-2.39068E-06	-1192.74	2.39068E-06	0.00406	-6.24908889
8	0.039378684	0.130915776	5.74408E-05	1.64495E-05	592.180408	240.4252456	221.041	0.60851	<Max torque	0.34464	0.011619	-3.39707E-06	-1178.79	3.39707E-06	0.00406	-6.23163333
9	0.039173427	0.131063592	7.38902E-05	1.63531E-05	588.710086	239.0162948	221.221	0.81416	444.137784	0.44334	0.011619	-4.47366E-06	-1151.45	4.47366E-06	0.00406	-6.21417778
10	0.038956234	0.131206895	9.02433E-05	1.62494E-05	584.980088	237.5019157	228.269	1.05104		0.54146	0.011619	-5.42771E-06	-1159.09	5.42771E-06	0.00406	-6.19672222
11	0.038727172	0.131345714	0.000106493	1.61387E-05	580.994376	235.8837166	231.188	1.27725		0.63896	0.011619	-6.48575E-06	-1144.67	6.48575E-06	0.00406	-6.17926667
12	0.03848631	0.131480083	0.000122631	1.6021E-05	576.757037	234.1633568	235.782	1.5186		0.73579	0.011619	-7.51394E-06	-1137.77	7.51394E-06	0.00406	-6.16181111
13	0.038233721	0.13161004	0.000138653	1.58965E-05	572.272282	232.3425465	239.152	1.75823		0.83192	0.011619	-8.59845E-06	-1124.16	8.59845E-06	0.00406	-6.14435556
14	0.037969484	0.131735625	0.000154549	1.57651E-05	567.544447	230.4230454	243.987	2.0148		0.92729	0.011619	-9.64905E-06	-1116.61	9.64905E-06	0.00406	-6.1269
15	0.037693677	0.131856883	0.000170314	1.56272E-05	562.577986	228.4066625	246.259	2.25521		1.02188	0.011619	-1.08267E-05	-1096.67	1.08267E-05	0.00406	-6.10944444
16	0.037406385	0.131973861	0.000185941	1.54827E-05	557.377476	226.2952552	249.993	2.51285		1.11565	0.011619	-1.19723E-05	-1082.72	1.19723E-05	0.00406	-6.09198889
17	0.037107696	0.132086608	0.000201424	1.53319E-05	551.947607	224.0907282	253.847	2.77675		1.20854	0.011619	-1.31405E-05	-1068.61	1.31405E-05	0.00406	-6.07453333
18	0.036797701	0.132195179	0.000216756	1.51748E-05	546.293185	221.7950332	252.468	2.9838		1.30054	0.011619	-1.46363E-05	-1032.42	1.46363E-05	0.00406	-6.05707778
19	0.036476494	0.132299631	0.000231931	1.50116E-05	540.419132	219.4101675	252.55	3.20504		1.39158	0.011619	-1.61264E-05	-1002.63	1.61264E-05	0.00406	-6.03962222
20	0.036144173	0.132400021	0.000246942	1.48425E-05	534.330476	216.9381732	248.748	3.37171		1.48165	0.011619	-1.79678E-05	-958.119	1.79678E-05	0.00406	-6.02216667
21	0.03580084	0.132496412	0.000261785	1.46676E-05	528.032356	214.3811364	245.071	3.53152		1.57071	0.011619	-1.99401E-05	-915.246	1.99401E-05	0.00406	-6.00471111
22	0.035446598	0.132588869	0.000276452	1.44869E-05	521.530015	211.7411863	242.954	3.70667		1.65871	0.011619	-2.19218E-05	-879.151	2.19218E-05	0.00406	-5.98725556
23	0.035081556	0.132677459	0.000290939	1.43008E-05	514.828802	209.0204935	239.623	3.85644		1.74564	0.011619	-2.4158E-05	-839.578	2.4158E-05	0.00406	-5.9698
24	0.034705825	0.13276225	0.00030524	1.41093E-05	507.934162	206.2212696	237.742	4.02286		1.83144	0.011619	-2.64023E-05	-805.972	2.64023E-05	0.00406	-5.95234444
25	0.03431952	0.132843316	0.000319349	1.39125E-05	500.851639	203.3457656	237.307	4.20942		1.9161	0.011619	-2.86223E-05	-777.825	2.86223E-05	0.00406	-5.93488889
26	0.033922757	0.132920729	0.000333262	1.37107E-05	493.586873	200.3962706	238.211	4.41759		1.99957	0.011619	-3.08001E-05	-754.317	3.08001E-05	0.00406	-5.91743333
27	0.033515659	0.132994565	0.000346973	1.3504E-05	486.145593	197.3751109	241.881	4.6781		2.08184	0.011619	-3.27152E-05	-739.376	3.27152E-05	0.00406	-5.89997778
28	0.033098349	0.133064902	0.000360477	1.32926E-05	478.533616	194.2846481	245.551	4.94168		2.16286	0.011619	-3.47123E-05	-723.959	3.47123E-05	0.00406	-5.88252222
29	0.032670955	0.13313182	0.000373769	1.30766E-05	470.756843	191.1272783	250.649	5.23798		2.24262	0.011619	-3.65893E-05	-712.147	3.65893E-05	0.00406	-5.86506667
30	0.032233606	0.133195399	0.000386846	1.28561E-05	462.821256	187.9054298	257.172	5.56995		2.32108	0.011619	-3.83349E-05	-703.5	3.83349E-05	0.00406	-5.84761111
31	0.031786436	0.133255724	0.000399702	1.26315E-05	454.732912	184.6215624	263.891	5.913		2.39821	0.011619	-4.01297E-05	-694.369	4.01297E-05	0.00406	-5.83015556
32	0.03132958	0.133312877	0.000412334	1.24027E-05	446.497943	181.278165	267.947	6.20108		2.474	0.011619	-4.24285E-05	-677.502	4.24285E-05	0.00406	-5.8127
33	0.030863179	0.133366944	0.000427736	1.21701E-05	438.122548	177.8777545	272.198	6.46636		2.54842	0.011619	-4.48175E-05	-660.681	4.48175E-05	0.00406	-5.79524444
SGDI Sheet1 1000rpm 1500rpm 1700rpm 2000rpm 2500rpm 3000rpm 3500rpm 4000rpm 4500rpm 5000rpm 5500rpm + ← →																

	AH	AI	AJ	AK	AL	AM	AN	AO	AP	AQ	AR	AS	AT	AU	AV	AW	AX	AY	AZ	BA	BB	BC
1																						
2	18055.8								17863.7													
3	4643.37		L						4600.01	R/L	Sin C.A.	Conrod ar	Fside	Cf Oil	Ffriction	Rcos	Lcos	S	Work done per degree due to friction	Corrected	FMEP	VOL(CC) S
4	Fgas (N)	Piston mass(kg)	M	R	W*2	F inertia 1st	2nd order	Total	Fnet (N)	m		Radians	N		N				J	J	Bar	C*2
5	450.863	0.406	0.134	0.041	10966.2	182.543756	54.2154	236.759	214.104	0.30597	-0.00081	-0.00025	-50.3251	0.005	-0.25163	0.041	0.134	1.8E-08	-1.8867E-06	1.9E-06	0.01783	
6	450.863	0.406	0.134	0.041	10966.2	182.518542	54.1855	236.704	214.159	0.30597	0.01664	0.00509	-49.1355	0.005	-0.24568	0.04099	0.134	7.4E-06	-5.879E-06	5.9E-06		
7	449.645	0.406	0.134	0.041	10966.2	182.437717	54.0895	236.527	213.118	0.30597	0.03409	0.01043	-47.7096	0.005	-0.23855	0.04098	0.13399	3.1E-05	-9.70008E-06	9.7E-06	sum of the	
8	452.183	0.406	0.134	0.041	10966.2	182.301305	53.9276	236.229	215.954	0.30597	0.05153	0.01577	-47.1515	0.005	-0.23576	0.04095	0.13398	7.1E-05	-1.33851E-05	1.3E-05	J	
9	452.183	0.406	0.134	0.041	10966.2	182.109349	53.7	235.809	216.373	0.30597	0.06895	0.0211	-46.0579	0.005	-0.23029	0.0409	0.13397	0.00013	-1.70727E-05	1.7E-05	0.71303	
10	458.78	0.406	0.134	0.041	10966.2	181.861905	53.4069	235.269	223.511	0.30597	0.08636	0.02643	-46.3638	0.005	-0.23182	0.04085	0.13395	0.0002	-2.07899E-05	2.1E-05		
11	461.216	0.406	0.134	0.041	10966.2	181.55905	53.0488	234.608	226.608	0.30597	0.10373	0.03174	-45.7867	0.005	-0.22893	0.04078	0.13393	0.00029	-2.43063E-05	2.4E-05	WD 180	
12	465.175	0.406	0.134	0.041	10966.2	181.200876	52.626	233.827	231.348	0.30597	0.12108	0.03705	-45.5108	0.005	-0.22755	0.0407	0.13391	0.00039	-2.77384E-05	2.8E-05	0.01572	
13	467.814	0.406	0.134	0.041	10966.2	180.787493	52.139	232.927	234.887	0.30597	0.13838	0.04235	-44.9663	0.005	-0.22483	0.04061	0.13388	0.00051	-3.10688E-05	3.1E-05		
14	471.772	0.406	0.134	0.041	10966.2	180.319025	51.5886	231.908	239.864	0.30597	0.15565	0.04764	-44.6644	0.005	-0.22332	0.0405	0.13385	0.00065	-3.4206E-05	3.4E-05	WD 360	
15	473.092	0.406	0.134	0.041	10966.2	179.795616	50.9752	230.771	242.321	0.30597	0.17287	0.05292	-43.8667	0.005	-0.21933	0.04038	0.13381	0.0008	-3.71181E-05	3.7E-05	0.40395	
16	475.731	0.406	0.134	0.041	10966.2	179.217426	50.2998	229.517	246.213	0.30597	0.19003	0.05818	-43.309	0.005	-0.21654	0.04025	0.13377	0.00097	-4.00022E-05	4E-05		
17	478.37	0.406	0.134	0.041	10966.2	178.58463	49.563	228.148	250.222	0.30597	0.20714	0.06342	-42.7445	0.005	-0.21372	0.04011	0.13373	0.00116	-4.23199E-05	4.2E-05	WD 540	
18	475.731	0.406	0.134	0.041	10966.2	177.897421	48.7658	226.663	249.067	0.30597	0.22419	0.06865	-41.297	0.005	-0.20648	0.03996	0.13368	0.00136	-4.4111E-05	4.4E-05	0.15893	
19	474.411	0.406	0.134	0.041	10966.2	177.15601	47.9093	225.065	249.346	0.30597	0.24116	0.07386	-40.1052	0.005	-0.20053	0.03979	0.13363	0.00158	-4.54758E-05	4.5E-05		
20	469.133	0.406	0.134	0.041	10966.2	176.36062	46.9943	223.355	245.778	0.30597	0.25806	0.07904	-38.3248	0.005	-0.19162	0.03961	0.13358	0.00181	-4.62613E-05	4.6E-05	WD 720	
21	463.855	0.406	0.134	0.041	10966.2	175.511496	46.022	221.534	242.321	0.30597	0.27489	0.08421	-36.6098	0.005	-0.18305	0.03942	0.13353	0.00205	-4.69746E-05	4.7E-05	0.13443	
22	459.998	0.406	0.134	0.041	10966.2	174.608895	44.9937	219.603	240.395	0.30597	0.29163	0.08935	-35.1661	0.005	-0.17583	0.03922	0.13347	0.00232	-4.75147E-05	4.8E-05		
23	454.822	0.406	0.134	0.041	10966.2	173.653093	43.9105	217.564	237.258	0.30597	0.30828	0.09447	-33.5831	0.005	-0.16792	0.039	0.1334	0.00259	-4.78755E-05	4.8E-05		
24	450.965	0.406	0.134	0.041	10966.2	172.64438	42.7739	215.418	235.546	0.30597	0.32484	0.09956	-32.2389	0.005	-0.16119	0.03878	0.13334	0.00289	-4.83438E-05	4.8E-05		
25	448.427	0.406	0.134	0.041	10966.2	171.583065	41.5851	213.168	235.259	0.30597	0.3413	0.10462	-31.113	0.005	-0.15556	0.03854	0.13327	0.00319	-4.89288E-05	4.9E-05		
26	447.108	0.406	0.134	0.041	10966.2	170.46947	40.3456	210.815	236.292	0.30597	0.35765	0.10965	-30.1727	0.005	-0.15086	0.03829	0.1332	0.00352	-4.97791E-05	5E-05		
27	448.427	0.406	0.134	0.041	10966.2	169.303936	39.0569	208.361	240.066	0.30597	0.3739	0.11465	-29.575	0.005	-0.14788	0.03803	0.13312	0.00385	-5.07737E-05	5.1E-05		
28	449.645	0.406	0.134	0.041	10966.2	168.086816	37.7207	205.808	243.837	0.30597	0.39003	0.11962	-28.9584	0.005	-0.14479	0.03775	0.13304	0.0042	-5.17683E-05	5.2E-05		
29	452.183	0.406	0.134	0.041	10966.2	166.818482	36.3385	203.157	249.026	0.30597	0.40604	0.12456	-28.4859	0.005	-0.14243	0.03747	0.13296	0.00457	-5.29126E-05	5.3E-05		
30	456.04	0.406	0.134	0.041	10966.2	165.499321	34.912	200.411	255.628	0.30597	0.42193	0.12946	-28.14	0.005	-0.1407	0.03717	0.13288	0.00495	-5.4076E-05	5.4E-05		
31	459.998	0.406	0.134	0.041	10966.2	164.129733	33.4429	197.573	262.425	0.30597	0.43769	0.13432	-27.7748	0.005	-0.13887	0.03686	0.13279	0.00534	-5.4834E-05	5.5E-05		
32	461.216	0.406	0.134	0.041	10966.2	162.710138	31.9331	194.643	266.573	0.30597	0.45332	0.13915	-27.1001	0.005	-0.1355	0.03655	0.1327	0.00575	-5.51778E-05	5.5E-05		
33	462.536	0.406	0.134	0.041	10966.2	161.240966	30.3844	191.625	270.91	0.30597	0.46881	0.14394	-26.4272	0.005	-0.13214	0.03622	0.13261	0.00617	-5.53989E-05	5.5E-05		
		SGDI	Sheet1	1000rpm	1500rpm	1700rpm	2000rpm	2500rpm	3000rpm	3500rpm	4000rpm	4500rpm	5000rpm	5500rpm								

Indicated Mean Effective Pressure

BC	BD	BE	BF	BG	BH	BI					
IMEP Calculation											
VOL(CC) Sigma	VOL (CC) 1.6 Ecoboost	VOL 1.6 ecoboost	Work done on piston per degree	Corrected	Sum of the work done per degree on the piston	IMEP					
C^2	C^2	M^2	J	J	J	Bar					
39.27	44.59753335	4.45975E-05	0.004078123	0.004078123	416.4321335	10.4108					
39.31	44.63425666	4.46343E-05	0.01303306	0.01303306							
39.43	44.75177748	4.47518E-05	0.021985882	0.021985882		IMEP					
39.63	44.94973703	4.49497E-05	0.030962592	0.030962592		Pa					
39.9	45.22774011	4.52277E-05	0.040119986	0.040119986		1250547					
40.26	45.58535558	4.55854E-05	0.049485059	0.049485059							
40.69	46.02211691	4.60221E-05	0.058801351	0.058801351							
41.2	46.53752264	4.65375E-05	0.068194791	0.068194791							
41.79	47.13103692	4.7131E-05	0.077649241	0.077649241							
42.45	47.80209009	4.78021E-05	0.087037886	0.087037886							
43.2	48.55007924	4.85501E-05	0.096318228	0.096318228							
44.02	49.37436873	4.93744E-05	0.105740855	0.105740855							
44.91	50.2742909	5.02743E-05	0.114545543	0.114545543							
45.89	51.24914658	5.12491E-05	0.122753041	0.122753041							
46.93	52.2982058	5.22982E-05	0.130434797	0.130434797							
48.06	53.42070836	5.34207E-05	0.137323448	0.137323448							
49.26	54.61586456	5.46159E-05	0.144151935	0.144151935							
50.53	55.88285586	5.58829E-05	0.150740135	0.150740135							
51.87	57.22083553	5.72208E-05	0.157072871	0.157072871							
53.29	58.6289294	5.86289E-05	0.163630235	0.163630235							
54.78	60.10623657	6.01062E-05	0.170459649	0.170459649							
56.34	61.65183012	6.16518E-05	0.177885771	0.177885771							
57.97	63.26475789	6.32648E-05	0.185728955	0.185728955							
59.67	64.94404319	6.4944E-05	0.193764352	0.193764352							
61.44	66.68868564	6.66887E-05	0.202333993	0.202333993							
63.28	68.49766189	6.84977E-05	0.211214842	0.211214842							
65.18	70.36992641	7.03699E-05	0.219467432	0.219467432							
67.15	72.30441237	7.23044E-05	0.22702672	0.22702672							
69.19	74.30003237	7.43E-05	0.234523619	0.234523619							
GDI	Sheet1	1000rpm	1500rpm	1700rpm	2000rpm	2500rpm	3000rpm	3500rpm	4000rpm	4500rpm	5000rpm

Ring Friction

BK	BL	BM	BN	BO	BP		
Ring Friction							
radial ring force	Top Ring Ffriction	Top ring work done per degree	Corrected	Sum of	FMEP		
N	N	J	J	J	Bar		
24.431	0.122155	9.26881E-07	9.26881E-07	0.176257726	0.00441		
24.431	0.122155	2.96217E-06	2.96217E-06				
24.365	0.121825	4.99698E-06	4.99698E-06				
24.5025	0.1225125	7.03721E-06	7.03721E-06				
24.5025	0.1225125	9.11852E-06	9.11852E-06				
24.86	0.1243	1.1247E-05	1.1247E-05				
24.992	0.12496	1.33644E-05	1.33644E-05				
25.2065	0.1260325	1.54994E-05	1.54994E-05				
25.3495	0.1267475	1.76482E-05	1.76482E-05				
25.564	0.12782	1.97821E-05	1.97821E-05				
25.6355	0.1281775	2.18913E-05	2.18913E-05				
25.7785	0.1288925	2.40329E-05	2.40329E-05				
25.9215	0.1296075	2.6034E-05	2.6034E-05				
25.7785	0.1288925	2.78995E-05	2.78995E-05				
25.707	0.128535	2.96454E-05	2.96454E-05				
25.421	0.127105	3.1211E-05	3.1211E-05				
25.135	0.125675	3.2763E-05	3.2763E-05				
24.926	0.12463	3.42604E-05	3.42604E-05				
24.6455	0.1232275	3.56997E-05	3.56997E-05				
24.4365	0.1221825	3.71901E-05	3.71901E-05				
24.299	0.121495	3.87423E-05	3.87423E-05				
24.2275	0.1211375	4.04301E-05	4.04301E-05				
24.299	0.121495	4.22127E-05	4.22127E-05				
24.365	0.121825	4.4039E-05	4.4039E-05				
24.5025	0.1225125	4.59867E-05	4.59867E-05				
24.7115	0.1235575	4.80052E-05	4.80052E-05				
24.926	0.12463	4.98808E-05	4.98808E-05				
24.992	0.12496	5.15989E-05	5.15989E-05				
25.0635	0.1253175	5.33028E-05	5.33028E-05				
100rpm	2000rpm	2500rpm	3000rpm	3500rpm	4000rpm	4500rpm	5000rpm

Crankshaft Bearing Friction

BQ	BR	BS	BT	BU	BV	BW	BX	BY	BZ	CA	CB
Crankshaft Bearing Friction											
L	R	ω	d	v	μ	Surface area	Shear Force	Shear stress in oil	Drag Force	Work done/rev due to friction	FMEP
m	m	Rad/s	m	m/s	Pas.s	m ²	N	N/m ²	N		Bar
0.0245	0.0235	104.72	0.00003525	2.46091	0.01162	0.003617544	2.934404133	811.1592232	5.868808265	0.866558032	0.01083
Force	σ	N	coefficient of friction			Duty paramater					x5
17863.74122	15513453.08	1000	0.009855956			7.48963E-07					0.05416

Conrod Bearing Friction

Conrod Bearing friction											
L	R	ω	d	v	μ	Surface area	Shear Force	Shear stress in oil	Drag Force	Work done/rev due to friction	FMEP
m	m	Rad/s	m	m/s	Pas.s	m ²	N	N/m			Bar
0.021	0.022	104.72	3.5E-05	2.30383	0.01162	0.002902832	2.204361276	759.3831025	4.40872255	0.609418057	0.00762
											x4
											0.03047

Images show friction models for 1.6 Ecoboost at 1000 Rpm. These models are repeated for each rpm interval and also for the 1.0 Ecoboost (Full and Part load) and the 1.6 Sigma giving a total of 46 spreadsheets.

Valve Friction

	C	D	E	F	G	H	I	J	K	L	M	N	O	P	Q	R	S
1	OB Max	valve lift	spring rate	Mass	Velocity	Acceleration	500	Inertia force	Contact Force	initial compression	Measured Valve lift	Measured Valve lift	Coefficient of friction	Friction force	X	W.D per degree	Correct
2	mm	mm	N/mm	kg	mm/s	mm/s ²	m/s ²	N	N	mm	0.01mm	mm		N	m	J	
3	15.05	-0.2	14	0.06849	0	0	0	0	81.2	6	0	0	0.05	4.06	0	0	
4	15.05	-0.2	14	0.06849	0	0	0	0	81.2	6	2	0.02	0.05	4.06	0	0	
5	15.05	-0.2	14	0.06849	0	0	0	0	81.2	6	8	0.08	0.05	4.06	0	0	
6	15.05	-0.2	14	0.06849	0	0	0	0	81.2	6	16	0.16	0.05	4.06	0	0	
7	15.05	-0.2	14	0.06849	0	0	0	0	81.2	6	30	0.3	0.05	4.06	0	0	
8	15.05	-0.2	14	0.06849	0	0	0	0	81.2	6	41	0.41	0.05	4.06	0	0	
9	15.05	-0.2	14	0.06849	0	0	0	0	81.2	6	59	0.59	0.05	4.06	0	0	
10	15.05	-0.2	14	0.06849	0	0	0	0	81.2	6	77	0.77	0.05	4.06	0	0	
11	15.05	-0.2	14	0.06849	0	0	0	0	81.2	6	97	0.97	0.05	4.06	0	0	
12	15.05	-0.2	14	0.06849	0	0	0	0	81.2	6	117	1.17	0.05	4.06	0	0	
13	15.05	-0.2	14	0.06849	0	0	0	0	81.2	6	138	1.38	0.05	4.06	0	0	
14	15.05	-0.2	14	0.06849	0	0	0	0	81.2	6	159	1.59	0.05	4.06	0	0	
15	15.05	-0.2	14	0.06849	0	0	0	0	81.2	6	184	1.84	0.05	4.06	0	0	
16	15.05	-0.2	14	0.06849	0	0	0	0	81.2	6	202	2.02	0.05	4.06	0	0	
17	15.05	-0.2	14	0.06849	0	0	0	0	81.2	6	225	2.25	0.05	4.06	0	0	
18	15.05	-0.2	14	0.06849	0	0	0	0	81.2	6	246	2.46	0.05	4.06	0	0	
19	15.05	-0.2	14	0.06849	0	0	0	0	81.2	6	270	2.7	0.05	4.06	0	0	
20	15.05	-0.2	14	0.06849	0	0	0	0	81.2	6	290	2.9	0.05	4.06	0	0	
21	15.05	-0.2	14	0.06849	0	0	0	0	81.2	6	312	3.12	0.05	4.06	0	0	
22	15.05	-0.2	14	0.06849	0	0	0	0	81.2	6	336	3.36	0.05	4.06	0	0	
23	15.05	-0.2	14	0.06849	0	0	0	0	81.2	6	356	3.56	0.05	4.06	0	0	
24	15.05	-0.2	14	0.06849	0	0	0	0	81.2	6	378	3.78	0.05	4.06	0	0	
25	15.05	-0.2	14	0.06849	0	0	0	0	81.2	6	397	3.97	0.05	4.06	0	0	
26	15.05	-0.2	14	0.06849	0	0	0	0	81.2	6	419	4.19	0.05	4.06	0	0	
27	15.05	-0.2	14	0.06849	0	0	0	0	81.2	6	436	4.36	0.05	4.06	0	0	
28	15.05	-0.2	14	0.06849	0	0	0	0	81.2	6	456	4.56	0.05	4.06	0	0	
29	15.05	-0.2	14	0.06849	0	0	0	0	81.2	6	475	4.75	0.05	4.06	0	0	
30	15.05	-0.2	14	0.06849	0	0	0	0	81.2	6	491	4.91	0.05	4.06	0	0	
31	15.05	-0.2	14	0.06849	0	0	0	0	81.2	6	508	5.08	0.05	4.06	0	0	
32	15.05	-0.2	14	0.06849	0	0	0	0	81.2	6	527	5.27	0.05	4.06	0	0	
33	15.05	-0.2	14	0.06849	0	0	0	0	81.2	6	546	5.46	0.05	4.06	0	0	
34	15.05	-0.2	14	0.06849	0	0	0	0	81.2	6	561	5.61	0.05	4.06	0	0	
35	15.05	-0.2	14	0.06849	0	0	0	0	81.2	6	575	5.75	0.05	4.06	0	0	
36	15.05	-0.2	14	0.06849	0	0	0	0	81.2	6	588	5.88	0.05	4.06	0	0	
37	15.05	-0.2	14	0.06849	0	0	0	0	81.2	6	603	6.03	0.05	4.06	0	0	
38	15.05	-0.2	14	0.06849	0	0	0	0	81.2	6	617	6.17	0.05	4.06	0	0	
39	15.05	-0.2	14	0.06849	0	0	0	0	81.2	6	630	6.3	0.05	4.06	0	0	

Sheet1
500 rpm
1000 Rpm
1500 Rpm
2000 Rpm
2500 Rpm
3000 Rpm
3500 Rpm
...

[illegible]

Camshaft Bearing Friction

AR	AS	AT	AU	AV	AW	AX	AY	AZ	BA	BB	BC
Camshaft Bearing Friction											
n	R m	 Rad/s	d m	v m/s	 Pa.s	Surface m ²	Shear Fo N	Shear st N/m	Drag Fo	Work do	FMEP Bar
0.0115	0.0125	52.36	0.00001875	0.6545	0.0116	0.0009	0.3663	405.58	0.7326	0.0575	0.0007
											x10
											0.0072

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