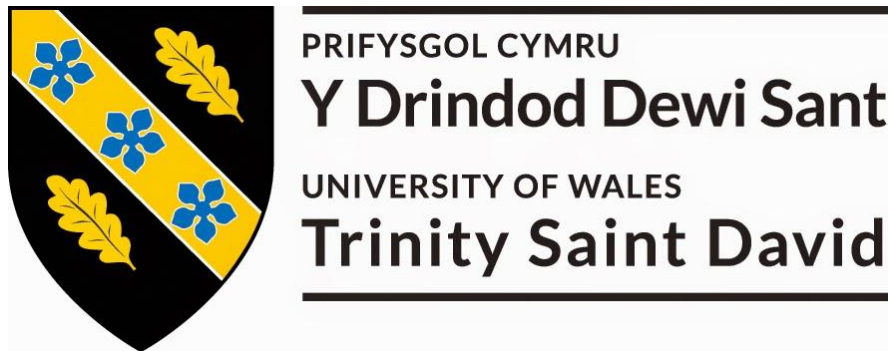




School of Engineering

A Holistic Approach to The Use of Ultra-low
Viscosity Lubrication in The Modern
Combustion Engine and Associated Engine
Architecture

by

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P146116

Declaration of Authenticity

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

Signed:**Date:****Abstract**

The role of modern engine lubrication has become a tortuous one, with the requirements made of it becoming at many times conflicting. This dissertation of this work sets out to explore the roles of a modern engine lubricant, the reasons for the paths being taken in the development of new lubrications and the engineering compromises being made to work with such styles of engine lubrication. The goal is to understand how the development of engine lubrication fits into the drive for a modern, more efficient internal combustion engine.

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List of symbols and abbreviations

TDC – Top dead centre

BDC – Bottom dead centre

ATDC – After top dead centre

ABDC – After bottom dead centre

GDI – Gasoline direct injection

FMEP – Friction mean effective pressure

RPM – Revolutions per minute

LHV – Lower heating value

SAE – Society of Automotive Engineers

Φ – Conrod angle

Θ – Crank angle

1. Introduction

All machinery that involves parts rotating or reciprocating in proximity to each other requires a degree of lubrication. Lubrication typically acts to prevent wear between the two surfaces of the parts in proximity and as a coolant to a certain extent. It was Richard Stribeck, who first formalized the study of entrained lubrication films in Berlin in the 1900s. With his work on journal bearing lubrication, he sought to characterize how lubrication performed between these two proximate components and determine the variables upon which the friction and wear could be derived. It is therefore, no surprise that all modern internal combustion engines feature a fluid, typically housed within a sump and pumped throughout the engine to act as lubrication between all parts that move in proximity to each other. However, far more scrutiny is being placed upon every aspect of the modern powertrain with the view to eke out every percentile of efficiency. As such the lubrication used in a modern engine is now expected to fulfil a tortuous role, whereby it must now provide lubrication, anti-wear protection and cooling to the engine without presenting significant parasitic losses. The aim of this body of work is to take a holistic view of the role of modern lubrication being in current use and in development, with a view to understanding the direction of future powertrain lubrication development and the engineering compromises that these pose. Can there ever be a perfect solution, one by which efficiencies can be gained without detriment to another fundamental requirement or property of a system?

This work addresses key areas of concern, such as - the nature and thermo-viscous properties of modern multi-grade oils used in modern internal combustion engines; the increasing propensity for manufacturers appearing to use decreasing viscosity oils; the necessity for the adaptation of engine geometry and combustion phasing to meet the hardware limits of these modern powertrain solutions when combined with reduced viscosity oils. This will take the form of modelling the heating and thermo-viscous properties of various engine lubricants, one of which is proposed for future use, within the context of a drive cycle simulation. Thus, demonstrating the benefits, if any, of such reduced viscosity lubrication over the term of an entire drive cycle.

2. Literature Review

The formative research for this work has been gathered from various research papers aimed at specifically addressing the performance of rheological lubrication within internal combustion engines. The approaches within each of the papers shall be compared and evaluated, in doing which it is intended to show where each of the hypotheses have similar ideas and assumptions, but also where they differ. The pros and cons of each of these ideas will be considered and how this has influenced the thought process and work carried out in the main body of this project.

2.1 Stribeck curves

At the core of each paper's main hypothesis is that the friction of any of these engine components be they in rational opposition like a journal bearing, or in reciprocating opposition, such as a piston to bore interface, is that the friction may be characterised in a standard form. This form takes the identity of the Stribeck curve, an example of which is shown below.

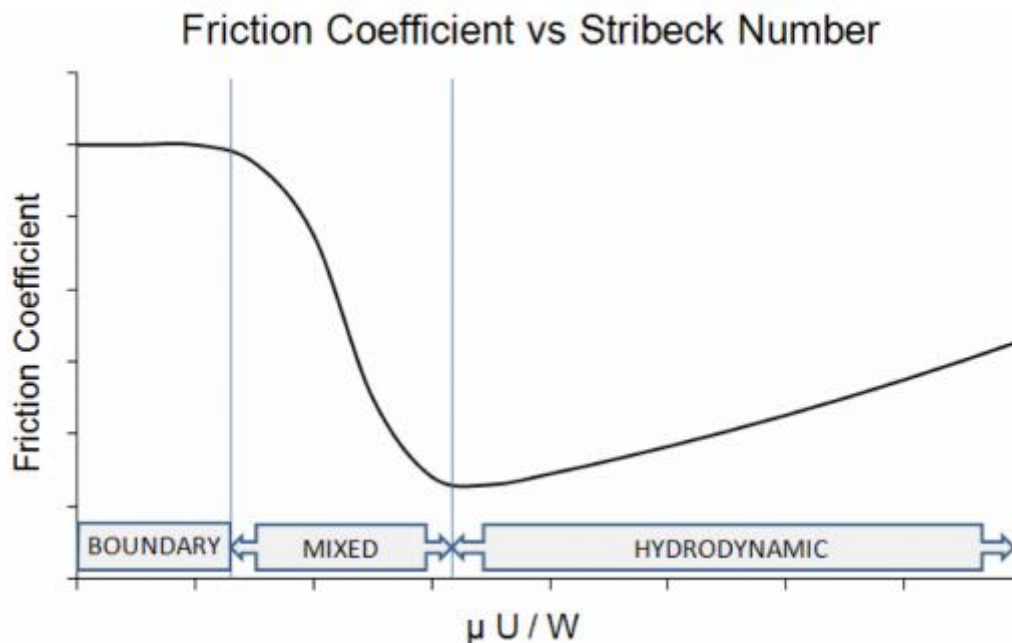


Figure 1 - Idealised Stribeck curve; (Plumley, 2014).

This concept is used by both the papers published by Carden et al of Ricardo Powertrain (Carden, 2014) and the group from the Massachusetts Institute of Technology (Plumley, 2014). The former is a study of lubrication performance in the piston bore region of spark ignition engines and the latter is an optimization of base oil temperature dependence for friction reduction. The idea is that this concept is based upon the work of Richard Stribeck in the early 1900's on journal bearings. The fundamentals are that coefficient of friction for two surfaces in opposed motion with a form of lubricity in play can be defined as conforming to one of three regimes. Firstly, boundary friction, whereby the lubrication does not offer any kind of separation between the two components in question. This means that direct sliding contact occurs between the two surfaces, as the figure shows from (Plumley, 2014), this is the region in which the friction coefficient is seen to be its highest. The second is mixed boundary and hydrodynamic regime, where the entrained lubrication film provides sufficient hydrodynamic pressure to allow there to be only partial asperity contact. This regime can be quite varied and general depending upon many variables, for now it shall be said that this regime represents a transitional region where friction is inversely proportional to the hydrodynamic pressure produced by the lubricant. The third regime is a fully hydrodynamic state between the two

surfaces. This describes a situation where there is sufficient hydrodynamic pressure created to ensure no asperity contact occurs.

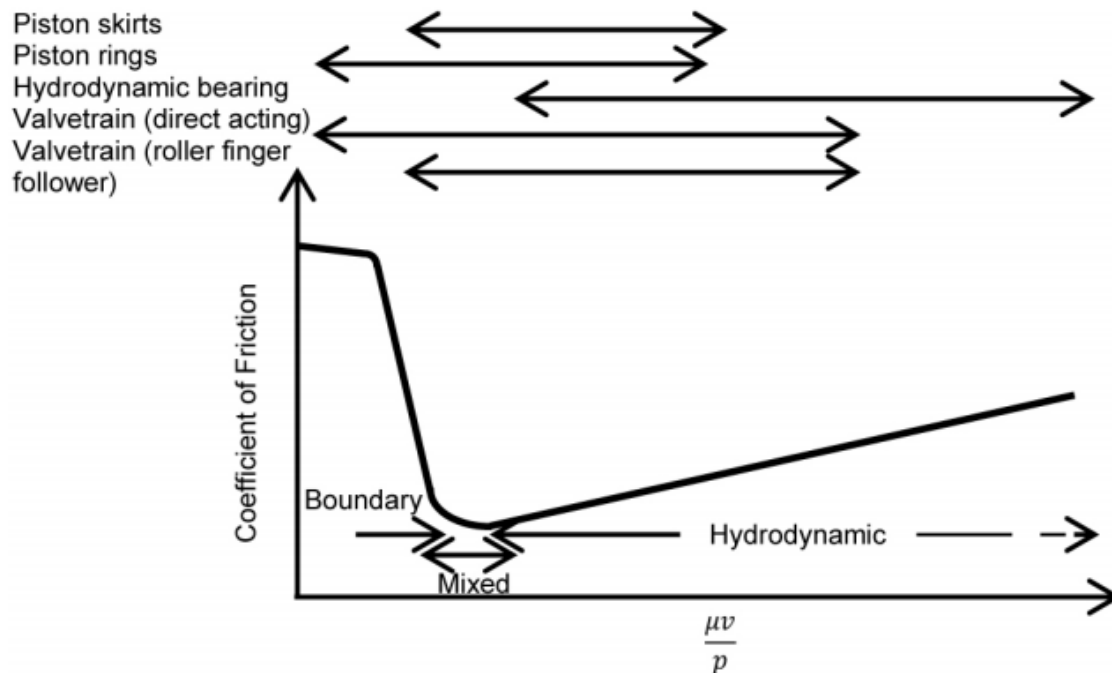


Figure 2 - Stylized Stribeck curve with components regime indication; (Carden, 2014)

As Figure 1 and 2 show, both of these papers agree on the Stribeck model of regime characterisations. However, what is added by (Carden, 2014), is the indication of where components are expected to fall within these regimes. It is expected that any reciprocating component will fall out of hydrodynamic lubrication at the apogee and perigee of its sinusoidal motion, whereas conditions being appropriate a rotational style bearing should maintain its mixed to hydrodynamic regime across a range of angular velocities.

These papers also have added to the basic Stribeck curve with new notation for the values on the X-axis of each of their graphs, with what is known as Hersey's number. Hersey's number is given by $\frac{\mu V}{P}$ (Carden, 2014), where μ is the absolute viscosity of the the lubricant, V is the speed of entrainment of the lubricant and P is the normal loading between the two surfaces in opposed motion (Plumley, 2014). What this suggests is that, the value of Hersey's is proportional to both absolute viscosity and speed of entrainment, whereas, it is inversely proportional to the normal load. It important to note that this loading only applies to the normal load and no other loading conditions the lubricant may be subject to. As per both of the diagrams presented in each of the papers, Hersey's number relates to the regime of the lubrication type to be expected and therefore, the coefficient of friction. The fact, however, is that neither paper makes the claim that these variables have a direct mathematical link to the coefficient of friction in the algebraic form it is presented. It can be said that this relationship gives only an indication of the lubrication regime, or moreover the shift in lubrication regime

with changes to these key variables. (Carden, 2014), mentions it is expected that fluctuations in loading speed or viscosity will then directly affect the lubrication regime. Further to which this paper explicitly states that there must be an optimum condition for every given component, where the aim is to reduce frictional losses. (Plumley, 2014), is primarily concerned with the fluctuations of the lubrication temperature and its direct effect upon the lubrication regime. What is also mentioned by (Carden, 2014), is that with the increasing trend of original equipment manufacturers to specify lower viscosity oil to reduce shear work done on the lubricant itself there is a risk of shifting the lubricant regime towards mixed and boundary friction, thereby actually increasing the overall coefficient of friction, the result being that these sub systems require re-engineering to cope with lower viscosity oils and still maintain efficiency and prevent wear. This is something further supported by the paper of (Gulf Science and Technology Co-T.W.Selby), a study upon engine lubricant viscosity and the performance of internal combustion engines. In this study, also subscribing to the generalised Stribeck model, where the intention was to explore how various viscosity lubrication was tested to compare performance and wear rate, it was stated that high viscosity lubricants had far greater load bearing capabilities, before loss of hydrodynamic regime.

These papers raise some key questions in their use of the Stribeck curve and even its use as a mathematical tool, or whether it simply provides for the generalised display of what is otherwise a complex system. What methods are being used by manufactures to mitigate the potential pitfalls of using reduced viscosity oil?

2.2 Viscosity temperature dependence

As was alluded to in the previous section, the lubricant temperature has an effect on its viscosity. Owing to the fact that (Carden, 2014), (Plumley, 2014) and (Gulf Science and Technology Co-T.W.Selby) show that the viscosity is a key component on determining the lubrication regime, then understanding lubricant temperature dependency behaviour is critical to performance and wear. Clearly this is of importance with combustion engines given that a significant proportion of the energy produced is rejected as heat, meaning that the lubricant can be expected to act in part as a coolant. The upshot of this means that the lubricant will be subject to significant temperature changes throughout the operation of the engine. (Carden, 2014) uses the “Walther equation”, which is a logarithmic function that takes the form

$$\log[\log(\mu + K)] = A + B \log T \quad (\text{Carden, 2014})$$

where μ is the kinematic viscosity of the lubricant and K is a constant that (Carden, 2014) states to be around 0.6. Also, both “A” and “B” are empirically derived constants. However, (Gulf Science and Technology Co-T.W.Selby) does state the use of the “Walther equation”, but there is no mention of value of constant K . It is difficult therefore, to know whether the two

papers are using similar values. There is another paper published by the same author T.W Selby based around the study of low viscosity oils, through a temperature range of 80° to 150° Celsius, where, again it mentions using of the Walther equation to extrapolate the temperature viscosity dependency. However, in this paper the it was explicitly stated the methodology behind its use, such that in this case it is given as;

$$\ln(\ln(\text{Viscosity} + 0.7)) = (\ln \text{ Kelvin}).\text{Slope} + \text{intercept} \quad (\text{Selby-Savant-Inc, 2010})$$

The two methods of use are relatively consistent in their approach, with noteworthy exception being that of the value used for “K”, where 0.6 was used by (Carden, 2014); a value of 0.7 is being used (Selby-Savant-Inc, 2010) in the second of his two papers examined here. The nomenclature used by (Selby-Savant-Inc, 2010) elaborates further what is going on mathematically within the Walther equation.

What is consistent is that it appears to be a well-recognised method for extrapolating the viscosity temperature dependency from empirical values. All that is required is two published kinematic viscosity values at 40° and 100° Celsius, which are given in the material safety data sheets for all engine lubricants brought to market. The result of this extrapolation for various engine lubricant types is show below for the (Carden, 2014) paper. This technique has been used by each of the papers to take this extrapolated data and use it to fill in the gaps in the model where measured values aren’t present. Assuming that this model does hold true for a modern multi-grade engine lubrication, it may be used to model the oil’s performance with respect to its temperature during a thermal cycle. This has a clear use in the context of drive cycle simulation, if the lubricant temperature is a known or measured value then the viscosity may be inferred using the Walther relationship at any point in the drive cycle.

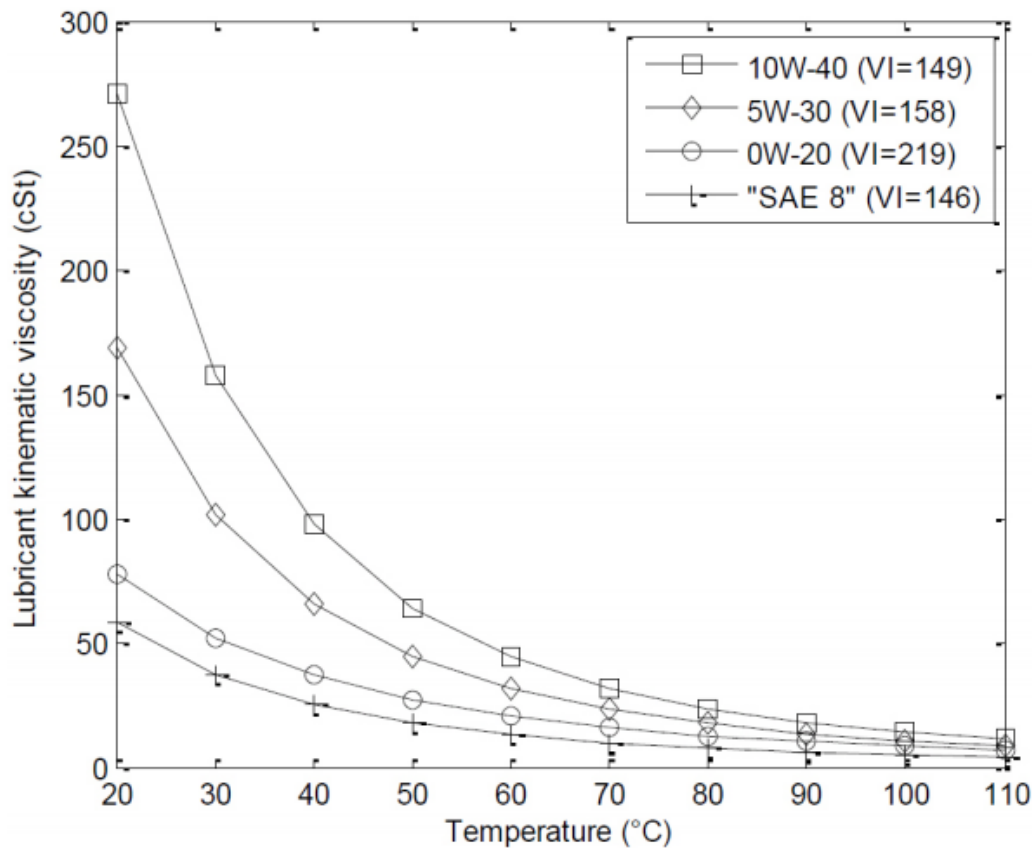


Figure 3 - Temperature viscosity extrapolation using the Walther equation; (Carden, 2014)

2.3 Viscosity modifiers

The term multi-grade oil is often used to describe a modern engine lubricant. This is no exception within each of the papers subject to scrutiny here. It must be said, that all oils analysed and modelled in ALL papers and in this report are multi-grade lubricants. (C.K Murphy and P.A Bennet Ressearch Laboratories, 1968) published a paper that concerned itself with the performance of such viscosity modifiers, in terms of what they describe as oil economy and engine wear over significant life testing. Although the intention isn't to focus upon the oil consumption or long-life wear, it does serve to compare and elaborate on the idea and function of a viscosity modifier.

First the design of a typical multi-grade lubricant must be considered, (C.K Murphy and P.A Bennet Ressearch Laboratories, 1968) introduces the concept of a base oil, as the primary oil upon which lubricant is based. This paper describes the base oil as a crude based primary mono-viscous oil, which is subject to the same temperature dependency previously mentioned. The difference is the rate of change of this viscosity. It is stated that a given base oil may well provide for adequate flow and performance at ambient temperatures but not at higher temperatures. This is where the base oil suffers rapid loss of viscosity with increased temperature. However, the idea that a modern engine oil is still based upon a crude oil is

slightly outdated, the issue being that the paper in question by (C.K Murphy and P.A Bennet Ressearch Laboratories, 1968) was published pre 1970's where research for the viscosity modifiers is relevant but the use of crude oil base oils may not be. For that reason, it would be prudent to introduce the ideas in the paper by (Petro-Canada Lubricants_H.E. Henderson, 1998). It is set forth in the "Petro-Canada Lubricants" paper that there arose a need for high quality base oils to be used to prevent oil consumption. This now introduces the idea of synthesised and blended base oils, where the base oil viscosity and characteristic can be finely controlled as per a set of specifications set out by the "SAE". What this means for all the previous papers and their research is that they are already building upon a deep engineering process. The viscosity modifier now fits the equation by providing an additive solution that effectively reduces the rate of viscosity degradation with temperature. This may be with the use of temperature sensitive polymers as (C.K Murphy and P.A Bennet Ressearch Laboratories, 1968) describes.

The implication that many oils can be manipulated in their temperature viscosity dependency is more important than the direct means of execution. By the careful consideration of the base oil and the use of polymeric additives the profile can be adjusted. This is exactly what is seen in the previous research, where (Carden, 2014) mentions briefly the use of "V.I Improvers"; it is the above that is indirectly referred to. The implication is that it is this use of viscosity modifiers that is allowing the use of such "low" viscosity oils. It not necessarily that the oil, from a holistic view, is low viscosity rather that the continued development of viscosity modifiers allows for the use of much lower viscosity base oils. It is this type of oil that both (Carden, 2014) and (Plumley, 2014) are claiming to be advantageous in the modern petrol engine.

What is left out of (Carden, 2014) is the subtle implications that come with such oils. (C.K Murphy and P.A Bennet Ressearch Laboratories, 1968) expressed concern over the long-term issues of viscosity modifiers, showing that over the long term there may have been significant loss in efficacy of the viscosity modifiers due to break down of the polymer due to shear stress. (Covitch, 1998), also presenting a paper on the polymer structure of viscosity modifiers, presents the same concern about long-term exposure to shear stress, again showing that these viscosity modifiers cannot be ensured for long term use without some degradation of the viscosity temperate profile. That is not to say that this affects the model as shown by (Carden, 2014), only that there must be overarching concerns with the use of ultra-low viscosity base oils supported by viscosity modifies.

The question, therefore, is what advantages do these ultra-low viscosity oils present in terms of increased efficiency? Would there be any advantage to further developing these types of lubricant?

2.4 Offset crankshafts

There are several papers that assert the hypothesis of using a crankshaft offset to reduce piston friction and increase thermodynamic efficiency in a typical combustion engine. Two studies that make this assertion are (GM Powertrain Division_Steve Shin, 2004), published by General Motors in response to, as it was put, “a production engine created by Toyota”. The second being (Musashi Institute of Technology_Kei Nakayama, 2000) whose paper closely aligns itself to the first in terms of test method, to explore the same hypothesis. Both papers use converted single cylinder petrol spark ignition engines with various degrees of crankshaft offset to test the theory.

The offset crank design is described by (GM Powertrain Division_Steve Shin, 2004), as an offset between the centre line of the cylinder bore and the centre of rotation of the crankshaft. It was also an assertion that with a crankshaft offset that the crank angle volume ratio changes such that the piston dwells near TDC. It is claimed that this improves the effective expansion ratio, thereby improving the combustion event. The claim is that an offset crank gives a “near constant volume combustion event” (GM Powertrain Division_Steve Shin, 2004). Should this be true it would mean that it would be close to an ideal Otto cycle. This also means that the effective expansion ratio would be improved, giving better thermal efficiency. However, (Musashi Institute of Technology_Kei Nakayama, 2000) does not make any claim that offsetting the crank should make any gain to the thermal efficiency through any geometric advantages. What they both state though, is that offsetting the crank in this way serves to reduce side wall friction on the thrust side. The theory presented is that, by utilizing an offset, the conrod angle is reduced through the high pressure firing of the engine, therefore, reducing any side loading that may arise as a component of force acting laterally.

(GM Powertrain Division_Steve Shin, 2004) used a single cylinder test engine with an offset of 18mm to the bore centre line. They claimed that this offset was selected so that the crank angle versus cylinder volume had a significant variation, shifting the TDC point to a representative 5° after crank TDC and BDC to a representative 190° after crank BDC. This relationship can be seen below in figure 4, where there is a clear geometric phase difference that arises from the crank position offset. Whereas, the test apparatus used by (Musashi Institute of Technology_Kei Nakayama, 2000) used a variable offset of 5mm, 10mm and 15mm. Although they state that this does cause geometric changes to the compression ratio, they do not explicitly state the effect it has on the phasing as the previous paper. The key information that both papers state is that the crank offset is to the thrust side, meaning that looking axially along a crankshaft that is rotating clockwise the direction of offset will be to the left.

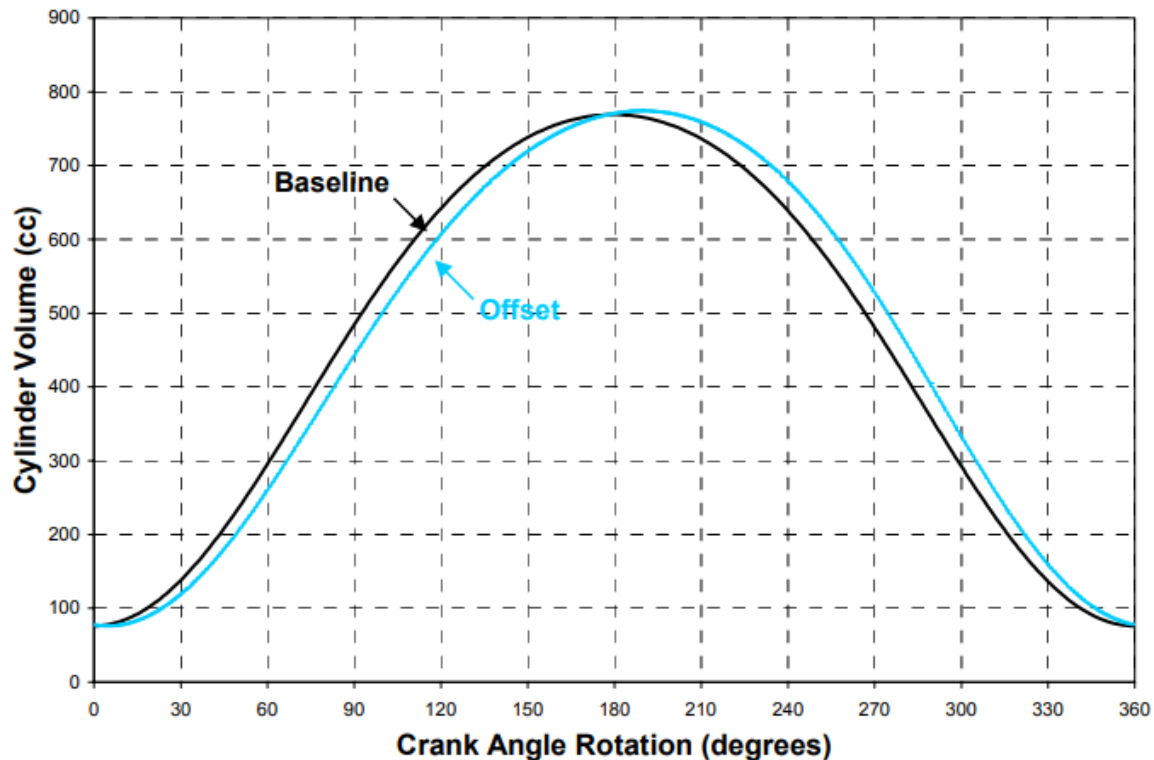


Figure 4 - Crank angle volume graph with an 18mm offset; (GM Powertrain Division_Steve Shin, 2004)

Figure 5 displays the specific test geometry of the General Motors test engine. Some interesting details are seen here that weren't explicitly mentioned in the body of the report is that the angular duration of the of both compression and expansions have changed in their proportion to each other. Further to this, it is important to understand how this change has affected other geometry and phasing within the engine. For example, the compression ratios have changed along with the swept volume, albeit a very small amount. There is a significant difference in value timing which at first glance may seem to be, obviously, to compensate for the change in phasing, assuming that the angular reference is that of crank TDC not piston TDC, as that should have been accounted for this already. However, as is stated by (GM Powertrain Division_Steve Shin, 2004) themselves, the valve overlap values change between the two configurations. In the case of the offset crank, it was stated that there is less valve overlap present. It is unclear, however, whether this was a conscious effort by General Motors to reduce valve overlap, or whether it was an inevitability of the test engine modifications. Along with this, they also performed several "sweeps" as they called them, which comprised of several runs of optimisation testing with fuelling and spark timing to find the optimal for each variable for both the standard and offset configuration. Each of these can be seen in the following Figures 6 and 7.

Specification	Baseline	Offset
Bore (mm)	93	93
Crankshaft Stroke (mm)	102	102
Conn. Rod Length (mm)	151	151
Piston Pin Offset (mm)	0.8	0.8
Crankshaft Offset (mm)	0	18
Piston Stroke (mm)	102	102.75
Displacement (cc/cyl.)	692.9	697.9
Compression Ratio	10.05	10.10
Piston TDC (°CA)	0	5
Piston BDC (°CA)	180	190
Expansion Duration (°CA)	180	185
Compression Duration (°CA)	180	175
Intake Cam Timing (°ATDC)	104	114
Exhaust Cam Timing (°ATDC)	-111	-104

Figure 5 - Geometry specifications for both standard and offset engines; (GM Powertrain Division_Steve Shin, 2004)

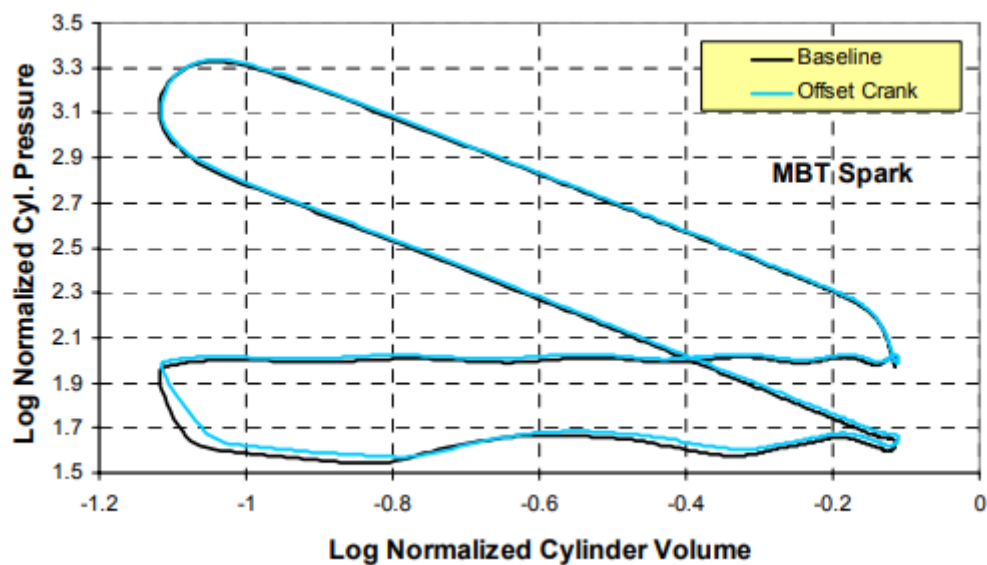


Figure 6 - Optimised log pressure volume graph for each configuration; (GM Powertrain Division_Steve Shin, 2004)

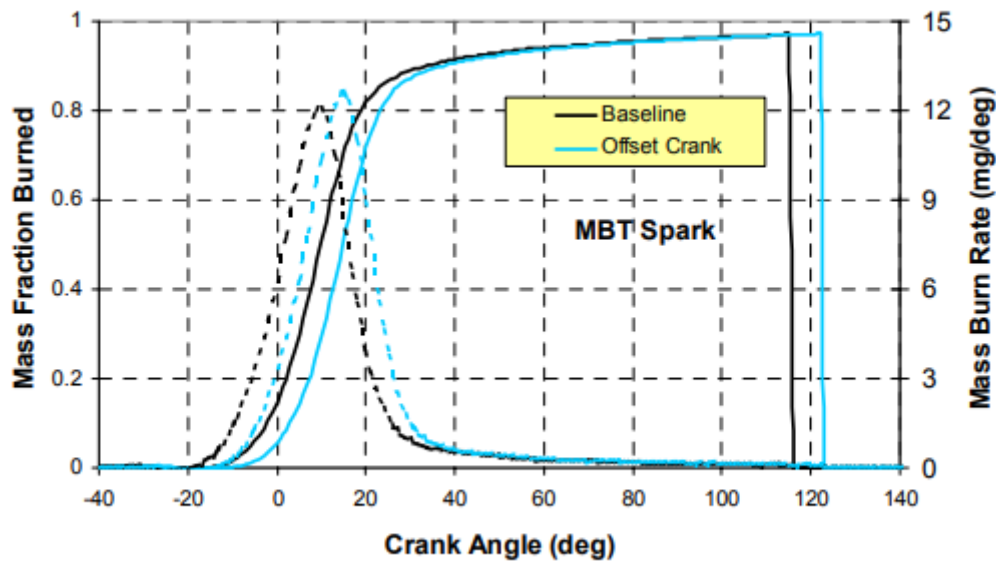


Figure 7 - Optimised MFB and pressure data for each configuration; (GM Powertrain Division_Steve Shin, 2004)

After the testing, both papers showed significant reduction in the side loadings on the thrust side of the cylinder wall by both methods of test. More notably with the testing conducted by (Musashi Institute of Technology_Kei Nakayama, 2000), with increased crank offset there was greater reduction in side force on the thrust side of the cylinder, where it was also seen that side force increased upon the anti-thrust side.

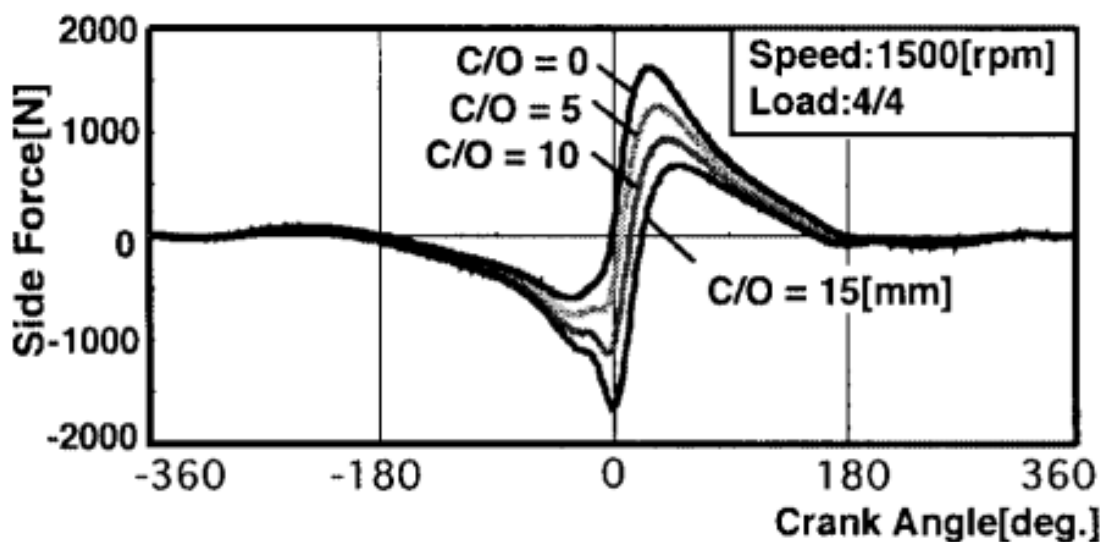


Figure 8 - Side force with varying crank offset; (Gulf Science and Technology Co-T.W.Selby)

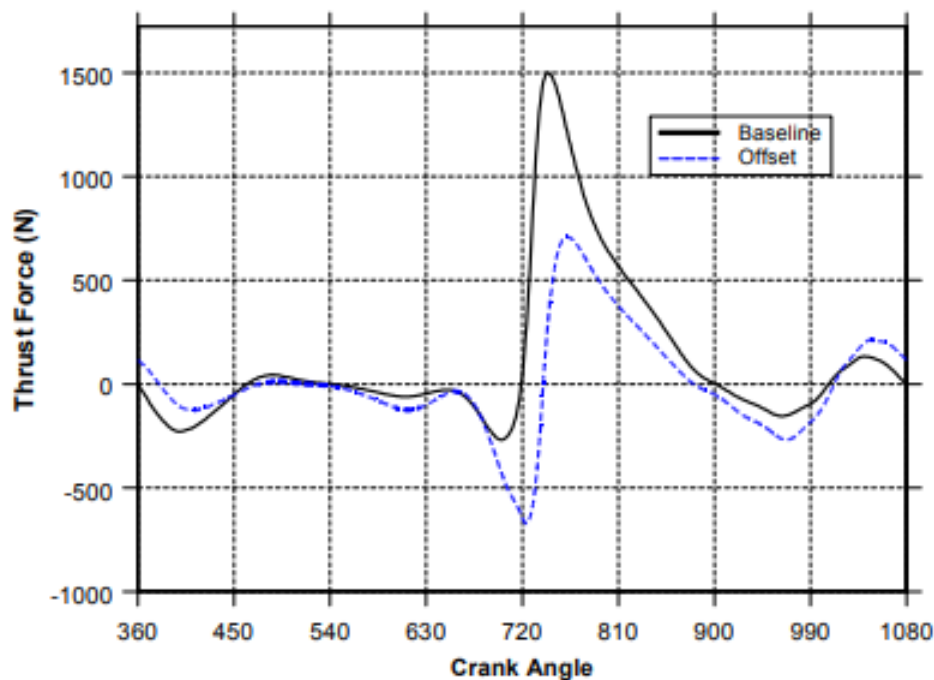


Figure 9 - Cylinder side loading with standard and offset crank; (GM Powertrain Division_Steve Shin, 2004)

Interestingly, what was stated by (GM Powertrain Division_Steve Shin, 2004), was that, despite this reduction in side force, they measured no noticeable difference in performance. The parameters by which they measured this were brake specific fuel consumption and overall torque. What this means then is that despite decreasing the side force they saw no advantage to the thermal efficiency of the engine. They also state that they observed an increase in FMEP, which seems to suggest that the overall friction effect has increased. There was no explicit explanation as to why this would be the case. Whereas, (Musashi Institute of Technology_Kei Nakayama, 2000), found there to be more subtlety to the condition, such that they state an observed reduction of piston frictional force of between 5-10% under their test conditions. This may seem to contradict the previous paper, but what they don't mention is whether they noticed an appreciable increase in engine efficiency or increased torque. What they do say is that the piston friction was subject to a great number of variables in terms of the lubrication delivery, which they say had a significant impact on the frictional results.

The questions that these papers raise are - how are the relative geometries performing in terms of their relationship to the standard engine? From where may the gains and or losses be derived? From this, are there considerations that can be made from understanding the geometric subtlety? Finally, how can understanding this be combined with further understanding of the overall lubrication system?

2.5 Research based aims

From the literature reviewed within this section, a picture began to form of a consideration for a modern powertrain solution that is focussed on optimising parasitic losses, from that most important of components... lubrication. The aim, therefore, is to better understand how each of these small aspects sum to a larger system that has a symbiotic relationship. Theoretically understanding what is critical to reducing the parasitic losses and what does this mean for the architecture of the next generation of internal combustion engines.

3. Engine Geometry

3.1 Standard engine geometry

To assess the important factors that impact the engine performance, it is prudent to first elaborate upon the standard engine geometry from a theoretical point of view. Illustrated below is what is understood of a conventional inline engine crank geometry.

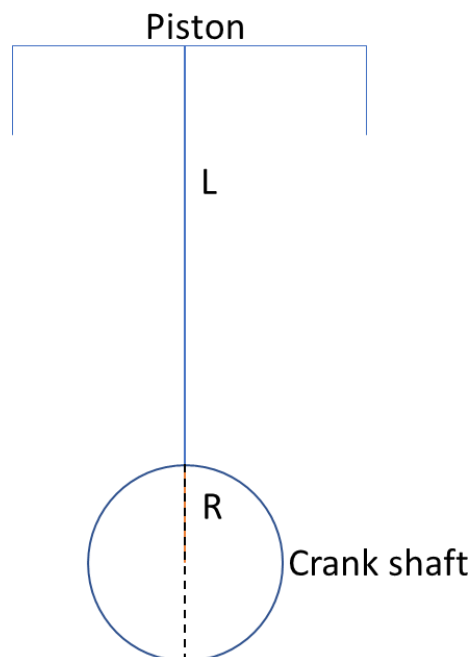


Figure 10 - Standard inline crankshaft geometry.

As the crank subtends an angle θ , the geometry can be seen to progress as shown in the next figure, from which it can be shown the piston displacement S has the following relationship.

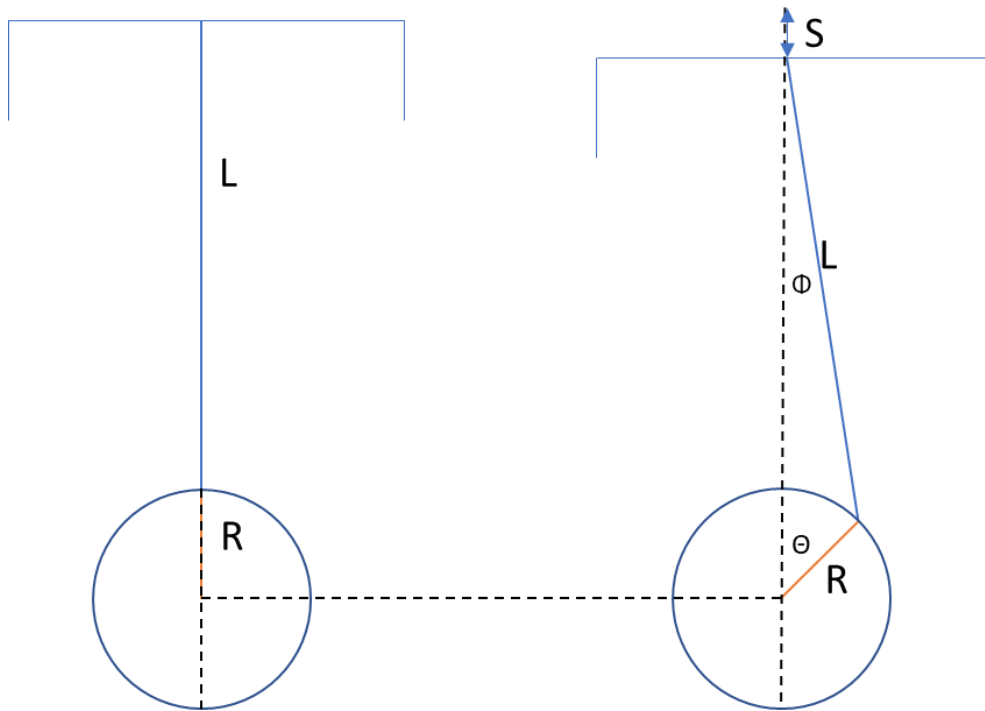


Figure 11 – Displaced piston geometry.

It can be seen from this, that when S is zero the crank and piston are at TDC. Therefore, the vertical distance between piston crown and crank centre must be equal to $R + L$, where, R is the crank radius and L is the conrod length.

$$S = R + L - \text{vertical distance}$$

So as per the diagram as the crank subtends angle theta;

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

We know that;

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

Also;

$$\cos^2 \phi + \sin^2 \phi = 1$$

$$\cos^2 \phi = 1 - \sin^2 \phi$$

$$\cos \phi = (1 - \sin^2 \phi)^{1/2}$$

Therefore;

$$\cos \phi = (1 - \frac{R^2}{L^2} \sin^2 \theta)^{1/2}$$

So, from this;

$$S = R + L - R \cos \theta - L \left(1 - \frac{R^2}{L^2} \sin^2 \theta\right)^{1/2}$$

We can consider;

$$\left(1 - \frac{R^2}{L^2} \sin^2 \theta\right)^{1/2} = \left(1 - \frac{R^2}{2L^2} \sin^2 \theta\right)$$

as an approximation because of the fact,

$$\begin{aligned} \left(1 - \frac{x}{2}\right) \left(1 - \frac{x}{2}\right) &= \\ 1 - x + \frac{x^2}{4} \end{aligned}$$

If $x = 0.09$, $x^2 = 0.0081$, therefore, when x is very small

$$\left(1 - \frac{x}{2}\right) \approx (1 - x)^{\frac{1}{2}}$$

From this the piston displacement per degree subtended by the crank can be achieved by differentiation of S with respect to angle theta.

$$\frac{dS}{d\theta} = R \sin \theta + \frac{L \cdot R \cdot R}{L \cdot L} \sin \theta \cos \theta$$

$$\frac{dS}{d\theta} = R \sin \theta + \frac{R^2}{L} \sin \theta \cos \theta$$

However, this is the piston velocity in distance per degree, where it is required in distance per unit time

$$\omega = \frac{d\theta}{dt}$$

$$\frac{dS}{dt} = \frac{dS}{d\theta} \times \frac{d\theta}{dt}$$

$$\frac{dS}{dt} = \frac{dS}{d\theta} \times \omega$$

Therefore;

$$\frac{dS}{dt} = \omega R \sin \theta + \omega \frac{R^2}{L} \sin \theta \cos \theta$$

From this, acceleration can be calculated by once again differentiating.

$$\frac{dS}{dt} = \omega R \sin \theta + \omega \frac{R^2}{L} \sin \theta \cos \theta$$

Where;

$$a = \frac{dv}{dt} = \frac{dv}{d\theta} \times \frac{d\theta}{dt} = \frac{dv}{d\theta} \times \omega$$

So,

$$\frac{d^2S}{dt^2} = \omega^2 R \cos \theta + \omega^2 \frac{R^2}{L} \cos 2\theta$$

Noting;

$$\omega = \frac{2\pi \text{ RPM}}{60} \text{ Rads/s}$$

This provides a mathematical basis on which to calculate various properties of the conventional inline geometry engine.

3.2 Offset crankshaft geometry

The offset crank geometry shall be illustrated in the same fashion below, using a similar method to approach the mathematics of the geometry.

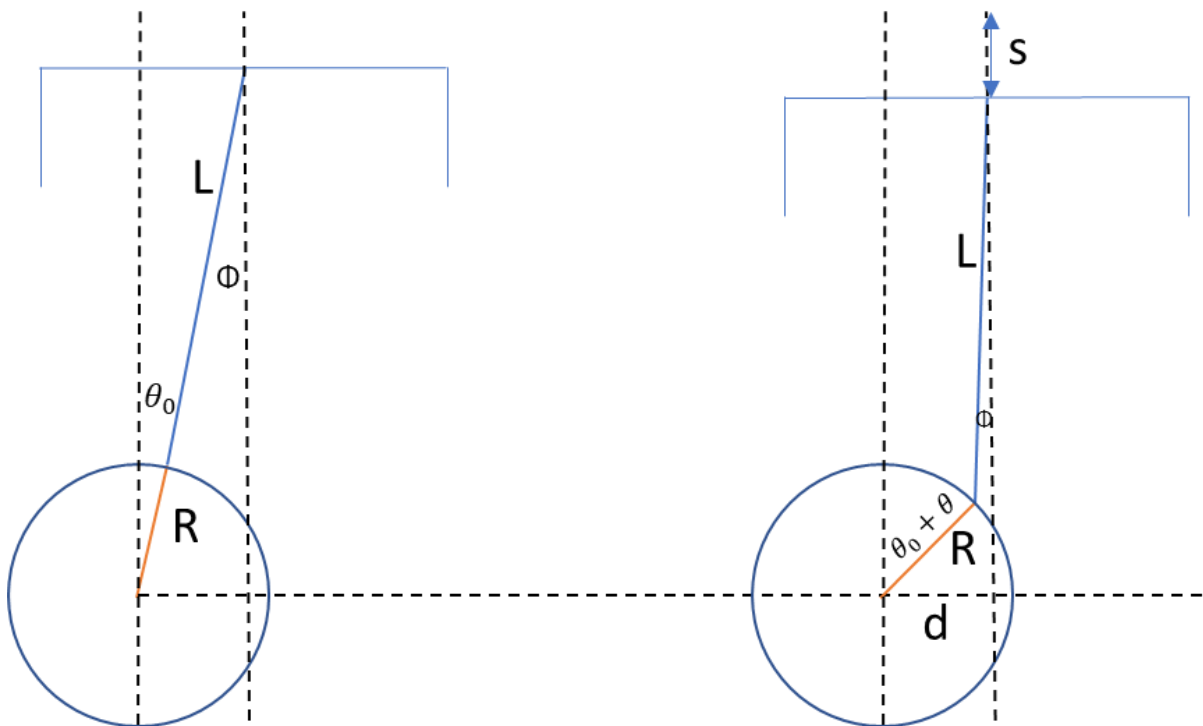


Figure 12 - Offset crankshaft geometry.

It can now be seen by virtue of the offset the geometry at TDC is shifted, such that when the piston is at TDC the crankshaft has an effective angular offset so that the vertical length between piston crown and crankshaft centre is now given by the relationship;

$$S = (L + R) \cos \theta_0 - \text{vertical}$$

$$S = (L + R) \cos \theta_0 - R \cos (\theta + \theta_0) - L \cos \phi$$

from the diagram it can be shown, because phi is negative on the anti-thrust side;

$$R \sin(\theta + \theta_0) = d + L \sin \phi$$

Rearranging for;

$$L \sin \phi = R \sin(\theta + \theta_0) - d$$

$$\sin \phi = \frac{R \sin(\theta + \theta_0) - d}{L}$$

$$\phi = \sin^{-1} \left(\frac{R \sin(\theta + \theta_0) - d}{L} \right)$$

Therefore;

$$S = (L + R) \cos \theta_0 - R \cos(\theta + \theta_0) - L \cos \left[\sin^{-1} \left(\frac{R \sin(\theta + \theta_0) - d}{L} \right) \right]$$

This describes the piston displacement as a function of crank angle. Rather than using direct differentiation for this geometry, as the assumptions made during the previous geometry were shown, it is possible to differentiate empirically within excel such that;

$$\frac{dS}{d\theta} = \frac{S_1 - S_2}{1}$$

As before;

$$\frac{dS}{dt} = \frac{dS}{d\theta} \times \frac{d\theta}{dt}$$

$$\frac{dS}{dt} = \frac{dS}{d\theta} \times \omega$$

So;

$$\frac{dS}{dt} = \frac{S_2 - S_1}{1} \times \omega$$

And therefore;

$$\frac{d^2S}{dt^2} = (v_2 - v_1) \times \omega$$

This means that now there are a set of mathematical terms that describe the motion of the crankshaft, rod and piston sub assembly for each of the geometry cases that are being considered.

3.3 Geometry comparison

For means of making an accurate and fair comparison, data for the Ford 1.6 GDI engine was used, as it was possible to gain accurate geometry measurements and use them in conjunction with cylinder pressure data. Using this to make a comparison between the standard engine and a hypothetical offset configuration of the same engine. It was deemed necessary to ensure that swept volume of the engine remained constant. The reason for doing this is so that the same pressure data may be used, given that the volume of air being ingested would be equal. This is of course an assumption that may have consequences to the validity of the model. These issues will be addressed as they arise and commented on.

The standard engine has a rod length of 0.13515 m, a crank radius of 0.0407 m, which means that the stroke of the engine S_{max} is equivalent to twice the crank radius, which for this engine is 0.0814 m. Understanding the relationship for the offset geometry, it was possible to set up a nonlinear solving algorithm to solve for the 3 variables that would give the same stroke for the offset crankshaft. To remove one variable, the crankshaft offset was fixed at 16mm as this well squarely between the values used by both (GM Powertrain Division_Steve Shin, 2004) and (Musashi Institute of Technology_Kei Nakayama, 2000). This reduced the solving time, resulting in a conrod length of 0.129 m and a crank radius of 0.04035 m. The result of which was to give a stroke of the offset crank of exactly 0.0814 m.

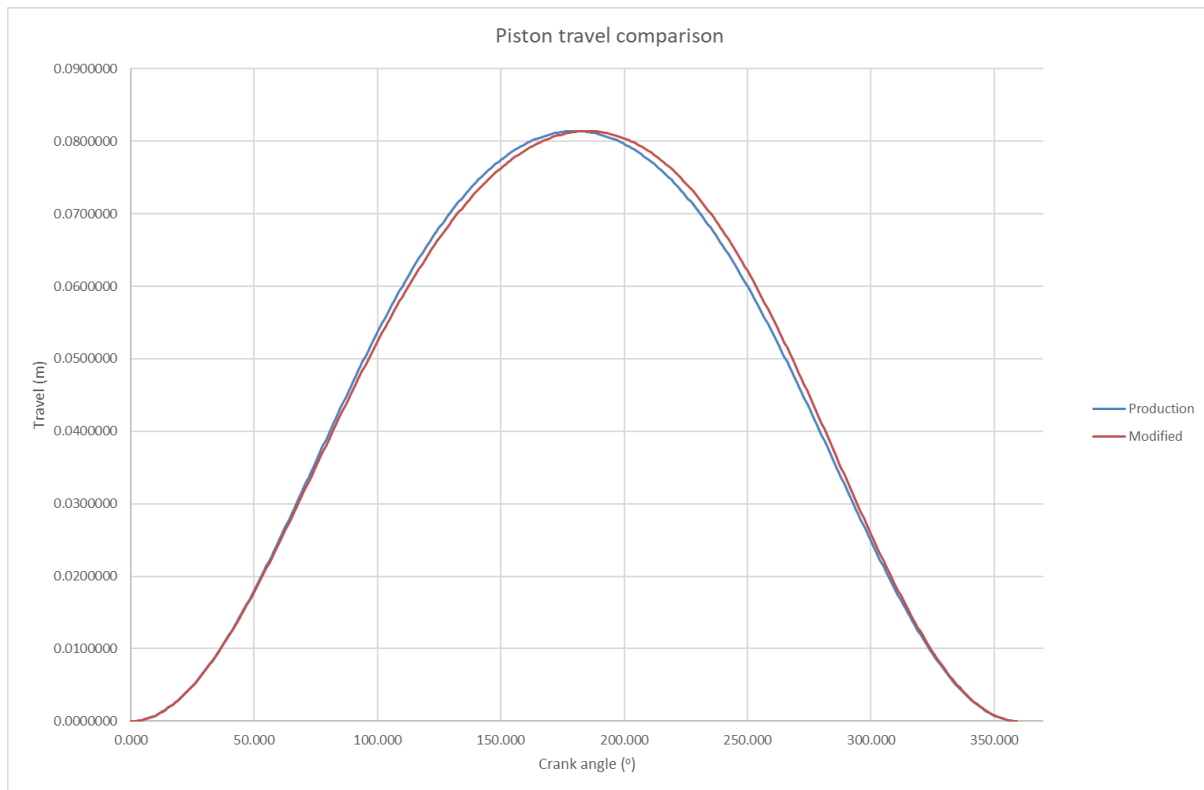


Figure 13 - Superimposed piston motion of the two engine geometries

To analyse this graph, it must first be noted that the crank angle is relative to each individual systems piston top centre. Theta Zero, as it shall be known for the offset crank, is 5.42 degrees after crank TDC. Zeroing the two in reference to each other allows them to be superimposed on one another. What is interesting to note is that it bears striking resemblance to the General Motors paper and their superimposed piston motion graph, with the graphs showing a very similar phase difference between the inline and offset crank. What was also similar was the fact that the BDC position for the model was at 185 degrees ATDC, whereas, it was report at 190-degree ATDC for the model by (GM Powertrain Division_Steve Shin, 2004). This is, of course, not concerning as the geometries were not expected to be identical to one another. The only significant discrepancy between the two is that this model does not show the piston dwell at TDC, that was seen by (GM Powertrain Division_Steve Shin, 2004). Figure 14 on the next page shows the General Motors piston motion graph, with a zoomed section graph of this model at TDC in Figure 15 below. It can be seen that no appreciable difference in piston motion occurs in the TDC region. In fact, it must be said that it would appear the only reason that the graph by (GM Powertrain Division_Steve Shin, 2004) shows this is because the two plots are shown asynchronously. This also may explain why their graph and this model are offset by the 5 or so degrees.

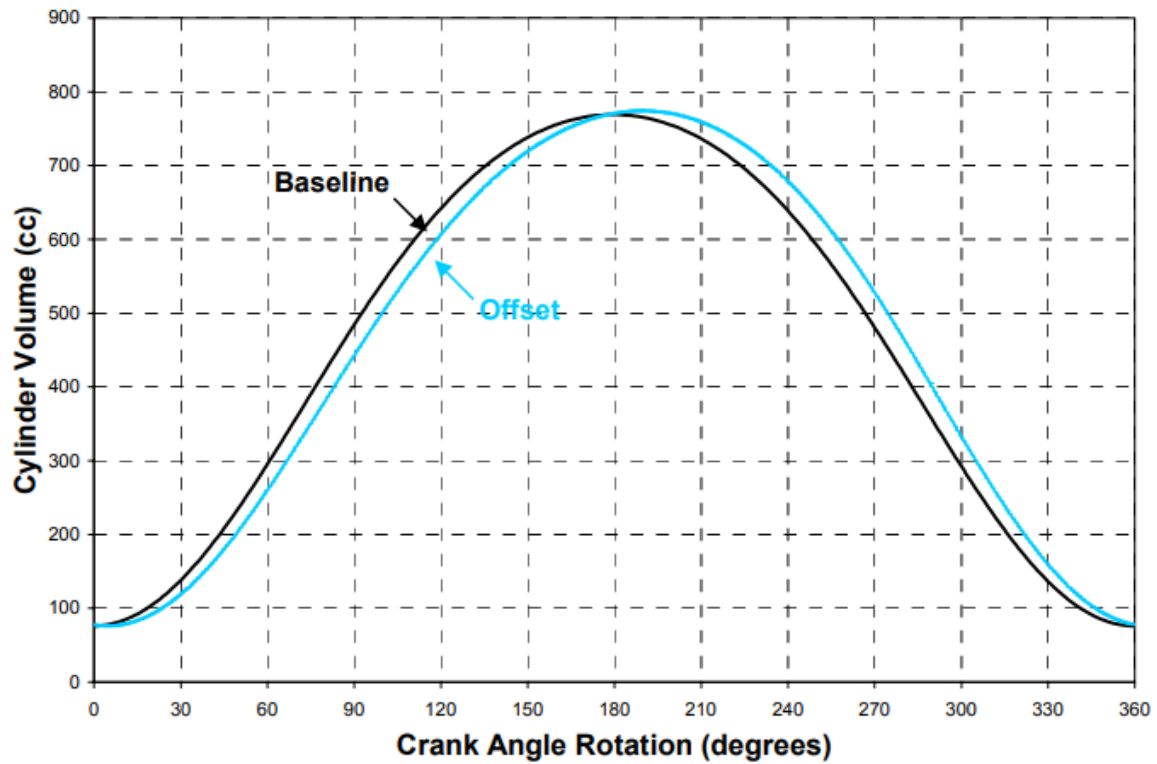


Figure 14 - General Motors superimposed piston motion by display of crank angle vs volume.

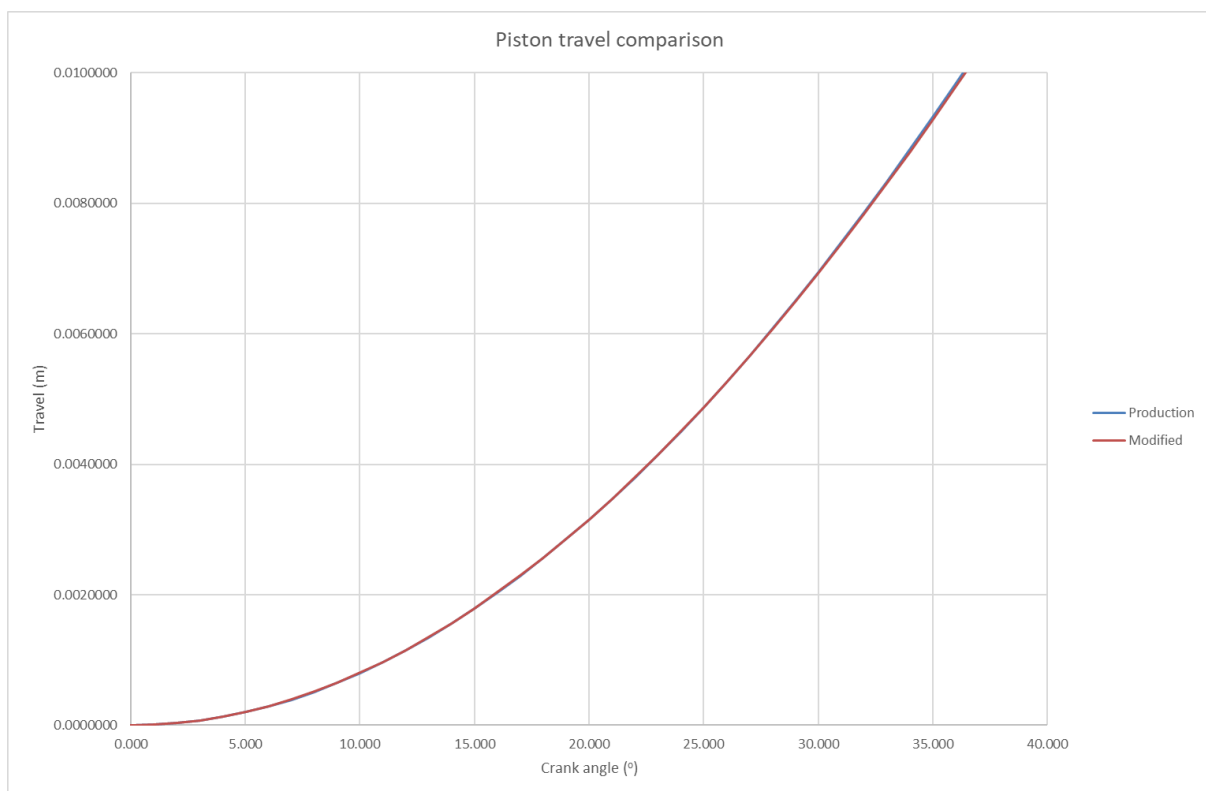


Figure 15 - Zoomed superimposed piston motion of the model to show lack of piston dwell.

3.4 Side loading

The intention of an offset crankshaft, as per those reasons expressed in the literature, was to decrease the side loadings on the piston to bore. It must be understood exactly what aspects of the geometry affect this side loading. The side loading derives from the two direct forces that act upon the piston itself, being the inertial loading and the gas forces. The inertial forces acting in the piston will be a function of its mass and acceleration such that;

$$F = ma$$

$$F = m \frac{d^2s}{dt^2}$$

These having been derived previously from the piston motion equations.

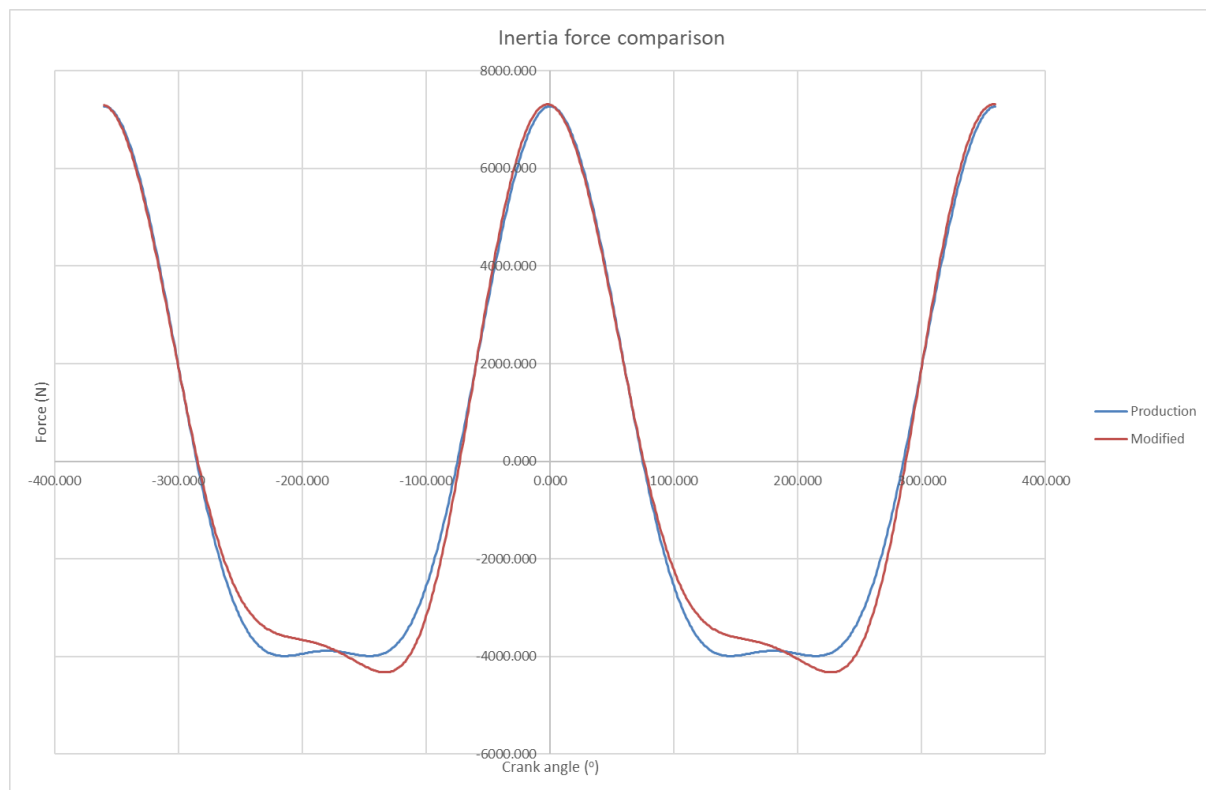


Figure 16 - Inertial piston loading comparison.

What can be seen in Figure 16 is that the difference in piston motion with respect to crank angle has the effect of skewing the inertia force graph towards the bottom of the engine's stroke. There is only a small increase in inertia loading on the compression stroke. The piston dimensions and, therefore, masses have been assumed to be identical. In doing so it means that any differences seen here are directly as a function of the changed geometry and not another variable. The piston and pin mass were measured from the Ford 1.6 GDI engine.

The second factor of the gas force seen on the piston due to the combustion process itself can be visualised by looking at the cylinder pressures present.

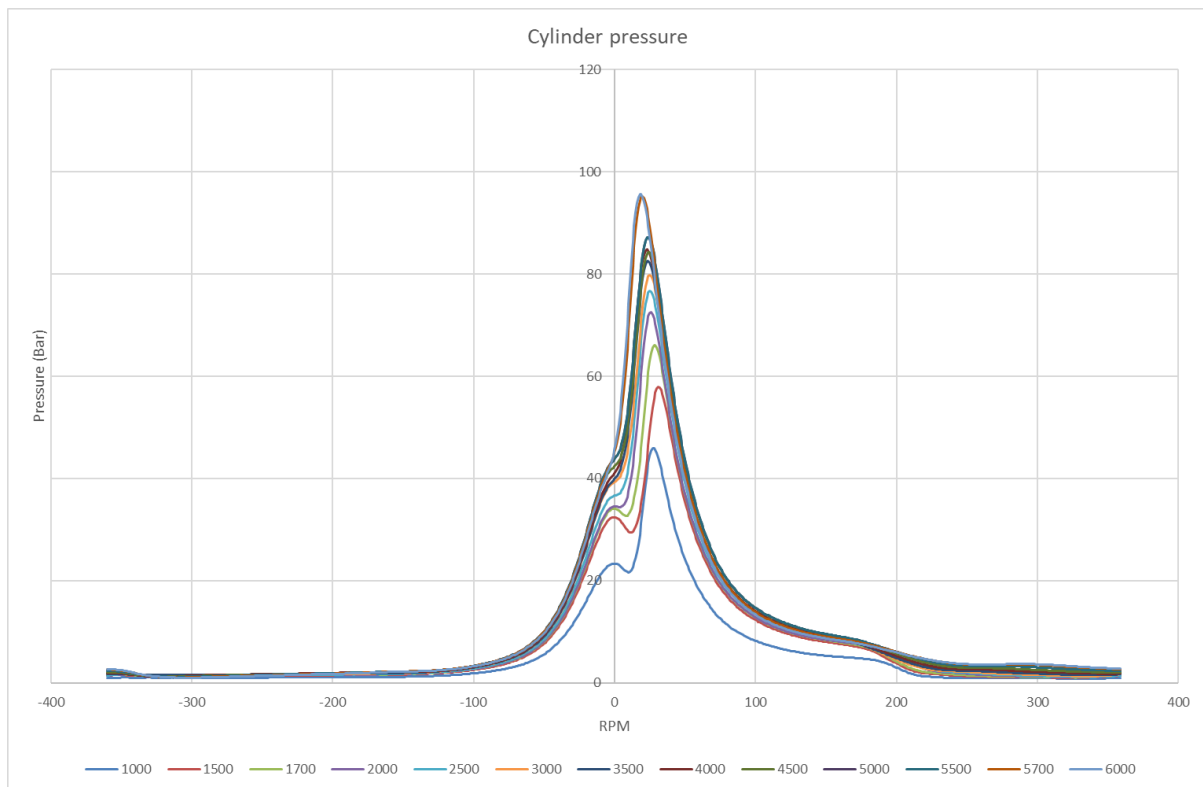


Figure 17 - Ford 1.6 GDI cylinder pressures.

What can be seen in Figure 17 raises several interesting points. Firstly, the range of magnitude of the pressure that can be seen in a modern turbo-charged engine and the range of pressures. All of these data sets are at full load, so it must be noted these are to be the maximum gas pressures seen at any one RPM site. What is of particular interest to this study is the combustion phasing. It is clear from these gas pressure data sets, that the peak pressure occurs ATDC, with peak pressure occurring for the RPM site that sees the highest overall peak pressure at 20 degrees ATDC and with the latest it occurs being at the 1500 RPM site where it occurs at 31 degrees ATDC. The position of the peak pressure occurrence can tell a story that will later be prove of great importance. The modern forced induction engine is clearly heavily knock limited as it can be seen in the lower RPM sites, combustion is being delayed greatly so that it occurs significantly ATDC. This is because the combustion must happen in a sufficiently large volume such that detonation can be prevented. This can be seen to an extreme in the lower RPM sites where expansion is visible in the pressure traces before combustion occurs. The crux of this fact is that this means the peak pressures are pushed further and further ATDC. It can be expected that in typical engine use, most users will not spend significant times under full load or high RPM sites. For this reason, it may be assumed that peak pressures will often be seen after the 20 degrees ATDC mark.

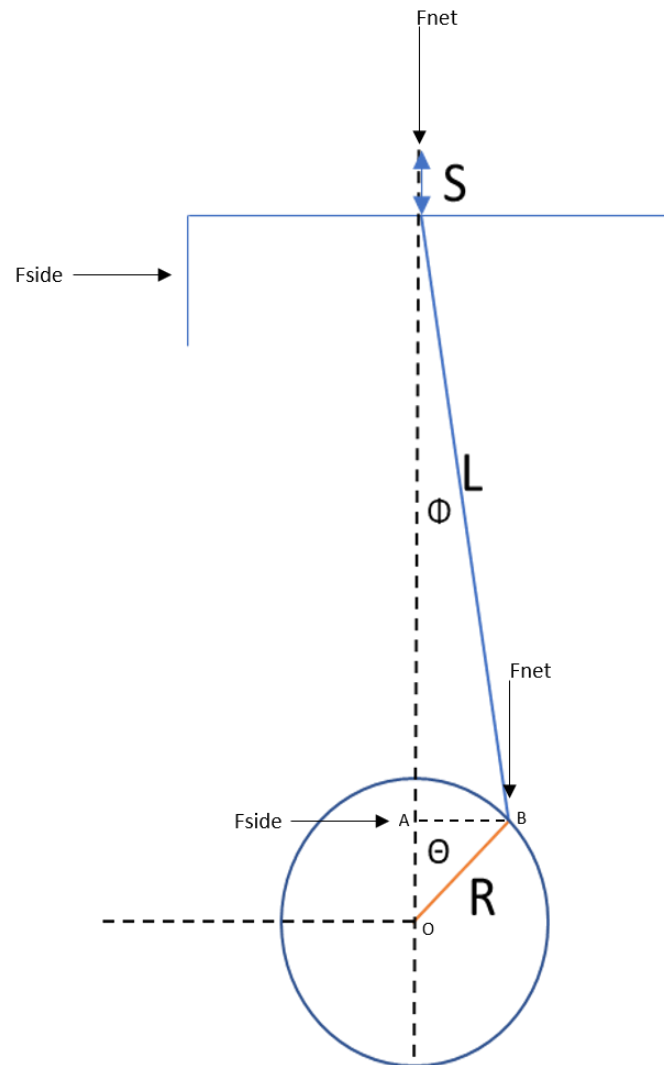


Figure 18 - Crank geometry force illustration.

It can be said that as the piston is described by rectilinear motion then the gas forces and the inertia loads may also only act along the plane of this motion. However, as the conrod is pin jointed at either end this means that the net force from both of the primary forces may be reacted at the cylinder wall as a function of this conrod angle, such that;

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

What can be said then is that the side force should increase if the net down force increases or if the rod angle increases. Knowing this the functions that describe the rod angle of each geometry models are;

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

Offset: $\phi = \sin^{-1} \left(\frac{R \sin(\theta + \theta_0) - d}{L} \right)$

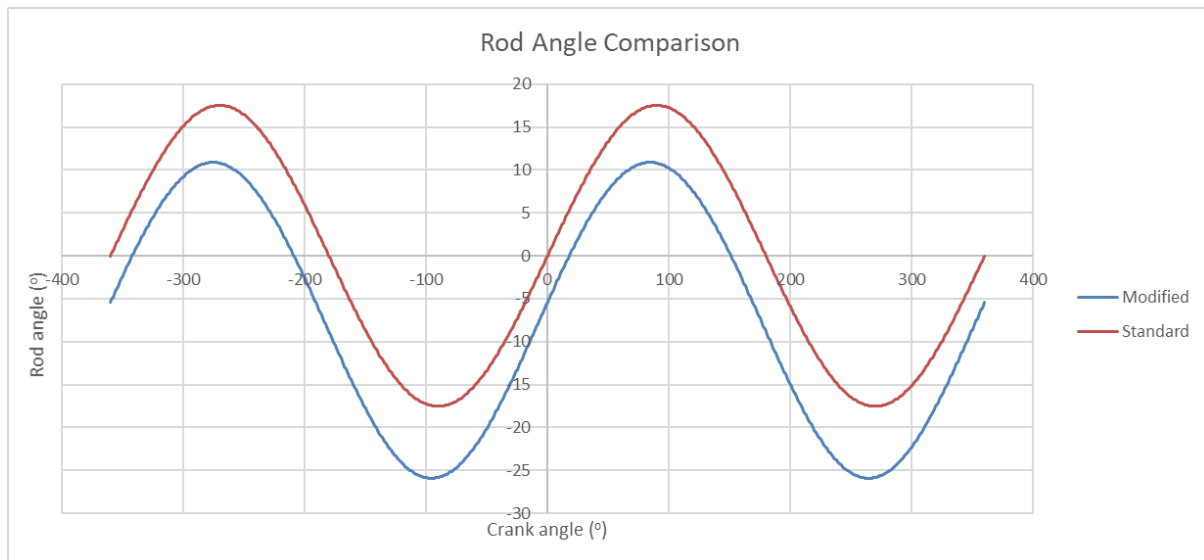


Figure 19 - Conrod angle comparison between standard and offset geometries.

Given that it is now understood that rod angle is a factor in determining the side force seen on the piston to bore region, it was sought to reduce it. However, simply having a goal of reducing rod angle is irrelevant unless it is considered in conjunction with the combustion phasing. Knowing that the $\tan$ of zero is zero and the fact that peak pressure should occur at 20 degrees ATDC, then the aim was to achieve an angle of ϕ of zero at or close to this value. It must be noted once again the crank angle θ being shown here is a zeroed reference for comparison, there is of course an angle offset between the two geometry sets. Once again, it was undertaken previously using a solving algorithm to achieve a geometry that allowed for the same stroke and therefore, volume, it has also been used to optimise for rod angle. Such that in this case the conrod angle is zero at approximately 18 degrees of relative crank angle. This can be seen as a comparison in Figure 19 above, where rod angle is greatly reduced during the peak firing region but increased before the TDC point. The reason that the rod angle isn't exactly zero at peak pressure was as a theoretical compensation for piston slap. This was mentioned by (Musashi Institute of Technology_Kei Nakayama, 2000), where having such high pressures during the rod angle transition point would lead to excessive piston slap. As such it was deemed suitable to have this transition just prior, to mitigate the issue of piston slap. Of course, this was entirely theoretical and would require real testing. What this does show is that the piston angle transition through zero can be modelled and placed quite accurately, whilst retaining all other geometrical properties.

What can now be explored is how all of this stacks up against the initial hypothesis that the offset crank angle reduces the side force upon the piston to bore region.

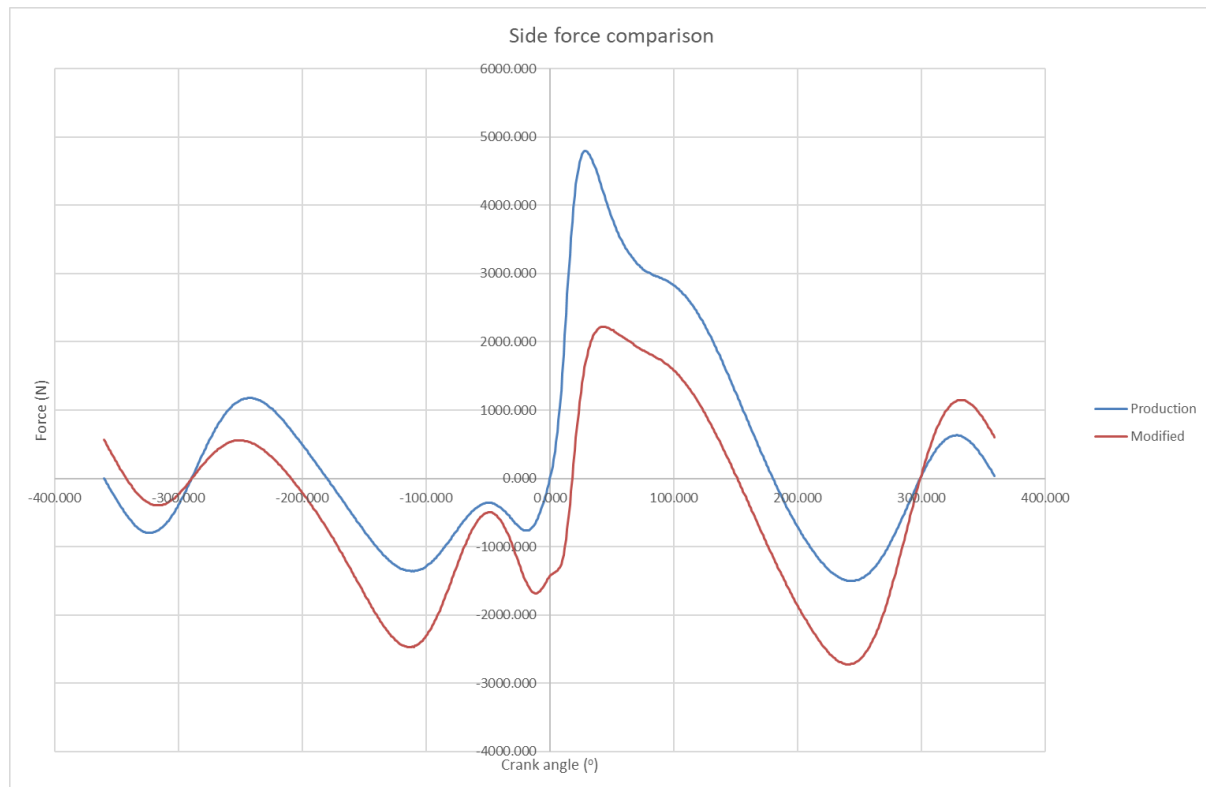


Figure 20 - Piston to bore side force comparison for each geometry.

Figure 20 illustrates the difference in side force between the standard inline geometry and the offset geometry at the 5700 RPM site. What can be seen is that there is more than a twofold reduction in force on the thrust side at its peak. However, as was to be expected from the trend in the conrod angle there is a large increase side force on the thrust side. The increase in side force is seen at TDC where it was expected due to the rod angle under the compression phase. There is also an even more significant increase close to BDC on the expansion phase. This is something that was not seen reported in either of the two papers, this could allude to the fact more than just the offset crank angle was changed in both of the paper examples. Where this model is designed to isolate the effects of just the crankshaft offset has on the system as a whole. It would be expected to see an increase in side force within this region as the conrod passes through its transitional phase at 175 degrees of relative crank angle. Along with the increased inertial loading that has shown to be present. The likelihood is that this answers one of the questions pertaining to the valve timing that was expressed in relation to the offset geometry by (GM Powertrain Division_Steve Shin, 2004), the mysterious change in cam timing that was not explained, could well have been implemented with this in mind. The fact that the displacement crank angle relationship is offset could mean that the exhaust valve timing needs to be advanced to compensate, thereby alleviating the cylinder pressure before the conrods transitional point.

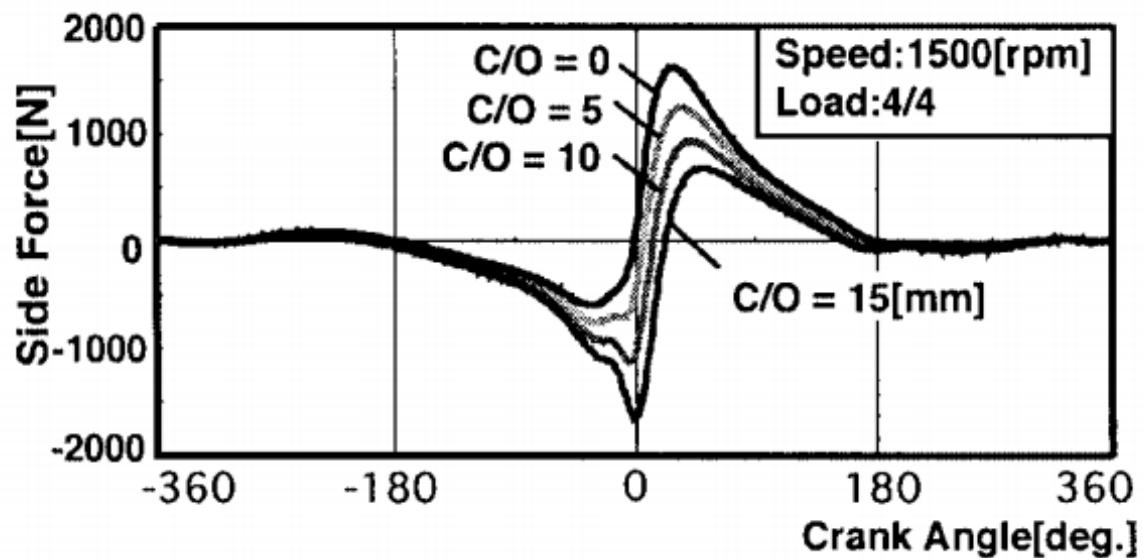
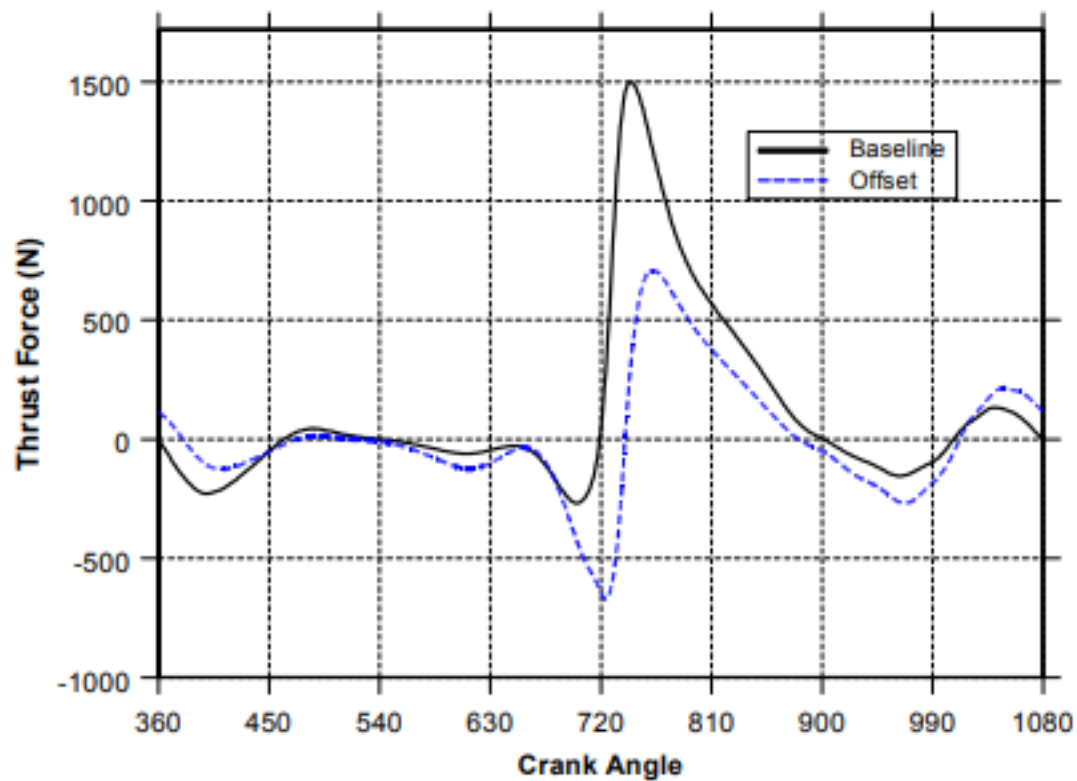
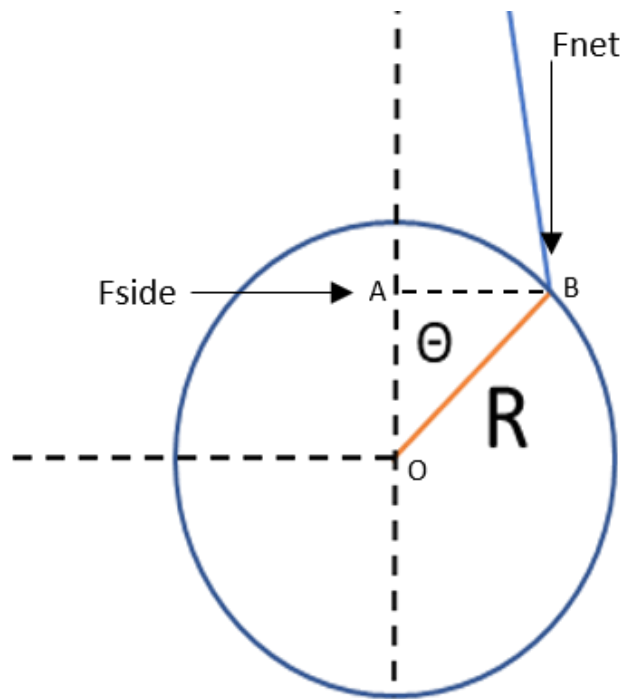


Figure 21 - Literature reference illustration of reduced side force; (GM Powertrain Division_Steve Shin, 2004)
(Musashi Institute of Technology_Kei Nakayama, 2000)

It is reassuring however, that the theoretical measurements and model closely match the trends of the research papers shown above in Figure 21. This means that the future use of these side loading simulations may be used with relative confidence within further calculations.

3.5 Torque

The close section portion of Figure 18 shown below, shows the relationship between the net force the side force and the torque acting at the centre of the crank.



Such that;

$$T = F_{side} \times OA + F_{net} \times AB$$

Knowing that;

$$T = F_{side} \cdot R \cos \theta + F_{net} \cdot R \sin \theta$$

Given that it has been shown previously;

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

So;

$$T = F_{net} \tan \phi \cdot R \cos \theta + F_{net} \cdot R \sin \theta$$

What has been shown here is exactly what has an impact on the torque acting at the crank centre. Given that the torque is the work being done at the crank, it is important now to understand what will impact this negatively. The derivation of engine torque has shown that the net down force is the key component of the engines torque meaning more gas force or less inertia means more torque, as was previously shown. However, the subtlety here is that this is a function of not only the crank angle but also a function of the rod angle as well. The issue then becomes immediately obvious, that if the side force has been reduced then the

instantaneous torque must also fall as a result. There are several complexities that arise from this small conclusion. The first is that according to both (GM Powertrain Division_Steve Shin, 2004) and (Musashi Institute of Technology_Kei Nakayama, 2000), that the reduction in side force must also result in a reduction in friction. This reduction in friction can be considered as a reduction in the work done to overcome it. However, when this is then set against the reduction in instantaneous torque the result is quite surprising.

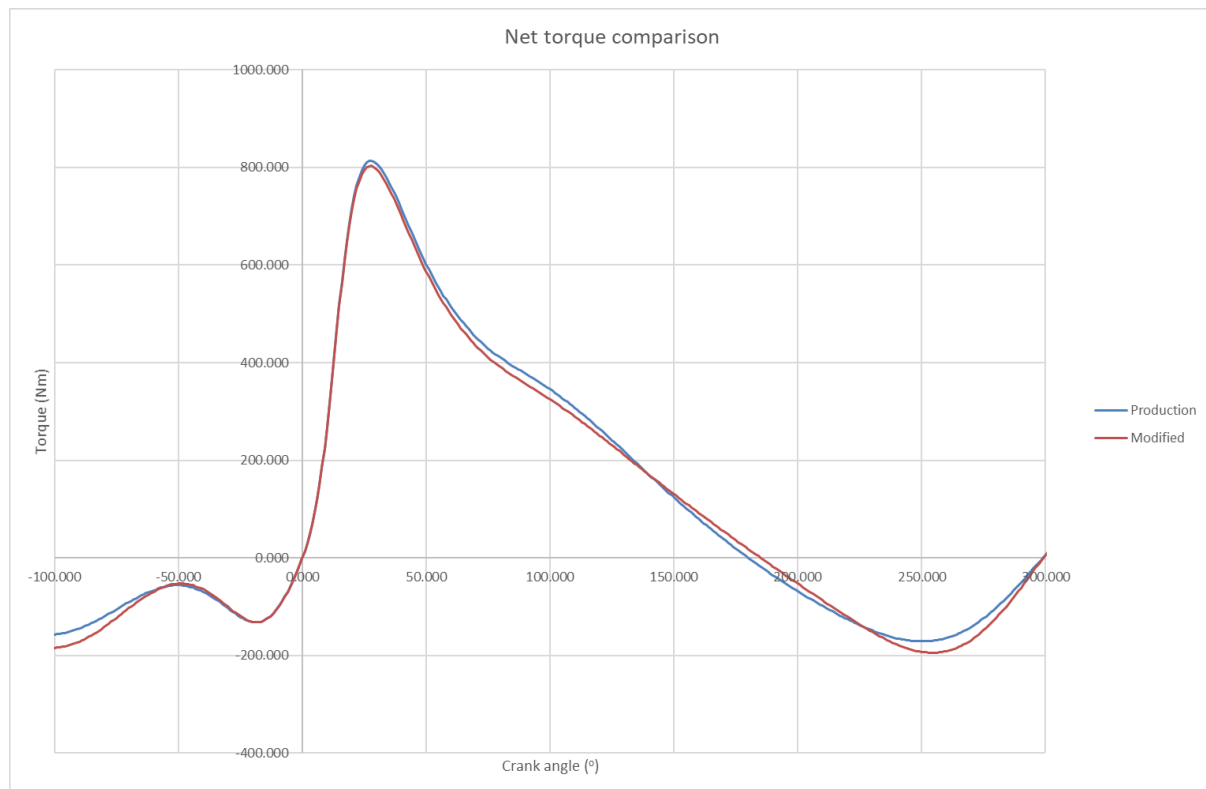


Figure 22 - Net torque comparison between the two geometries.

What can be seen in Figure 22, is the result of this subtle battle between these two properties of the side force. It can be hard to distinguish here as the result is very close but, it would appear in some areas the reduction in friction wins out against the reduced torque. Where in others the reduction in torque is enough that overall instantaneous torque is lost. This is for several reasons, one of which being that whenever the piston force is acting on the anti-thrust side then it serving to oppose the useful torque.

It is possible to see now why the conclusion made by (GM Powertrain Division_Steve Shin, 2004), was that offsetting the crankshaft had no observable benefit. This is not to say that there couldn't be of a benefit if the correct engineering considerations were made. Combustion phasing has already been discussed and as highlighted a conventional approach induces further problems. By careful consideration of the unique geometry the combustion timing could be phased such that the combustion takes place after the transition of the conrod from anti-thrust to thrust side; by the implementation of a revised valve timing that rejects the

cylinder pressure before again the conrod transition for on side to the other. All of this is hypothetical again so for the purpose of this model no meaningful conclusions can be draw from such conjecture. What can be said that it is now well understood how and why the offset crankshaft design delivers vastly reduced thrusting force under firing, which are the peak side forces seen. It is now possible to keep this revised engine geometry while exploring the tribology. Could this design facilitate efficiency savings elsewhere without thermal efficiency losses directly as (GM Powertrain Division_Steve Shin, 2004) suggests?

4. Applied Stribeck theory

Within the literature each of the studies concerned with the piston to bore tribology in modern combustion engines all cited the Stribeck theory of hydrodynamic lubrication. The idea of characterising the likelihood of the three types of lubrication regime was postulated. It is possible to asses this logic in terms of the drive to seek ultra-low viscosity and the previous technology shown in section 3. From (Carden, 2014), it was discussed that the key factors that will have an effect upon the likelihood of achieving good hydrodynamic lubrication are the absolute viscosity of the lubrication, the speed of entrainment and the normal load between the two surfaces in opposed motion. This relationship was termed Hersey's number, so the question is now how does Hersey's number with real variables?

4.1 Ring bore Stribeck theory

Something that wasn't mentioned within the section on offset crankshaft geometry was the piston rings. However, knowing that the piston rings are a vital component that are in constant contact with the bore they must now be considered. The piston rings are forced against the bore by the gas pressure present in the cylinder. Knowing that

$$F = P.A$$

then the force with which they are pressed against the cylinder bore is a function of the inner ring surface area and the gas pressure. As such it's expected the ring force will change in phase with the cylinder pressure. However, to reiterate the Stribeck relationship

$$H = \frac{\mu V}{P}$$

where in this case the speed of entrainment (V) is simply the piston velocity and of course μ is the absolute viscosity of the lubricant. By using the data from the previous section, it can be now shown how this conceptual value of Hersey's varies over a complete cycle.

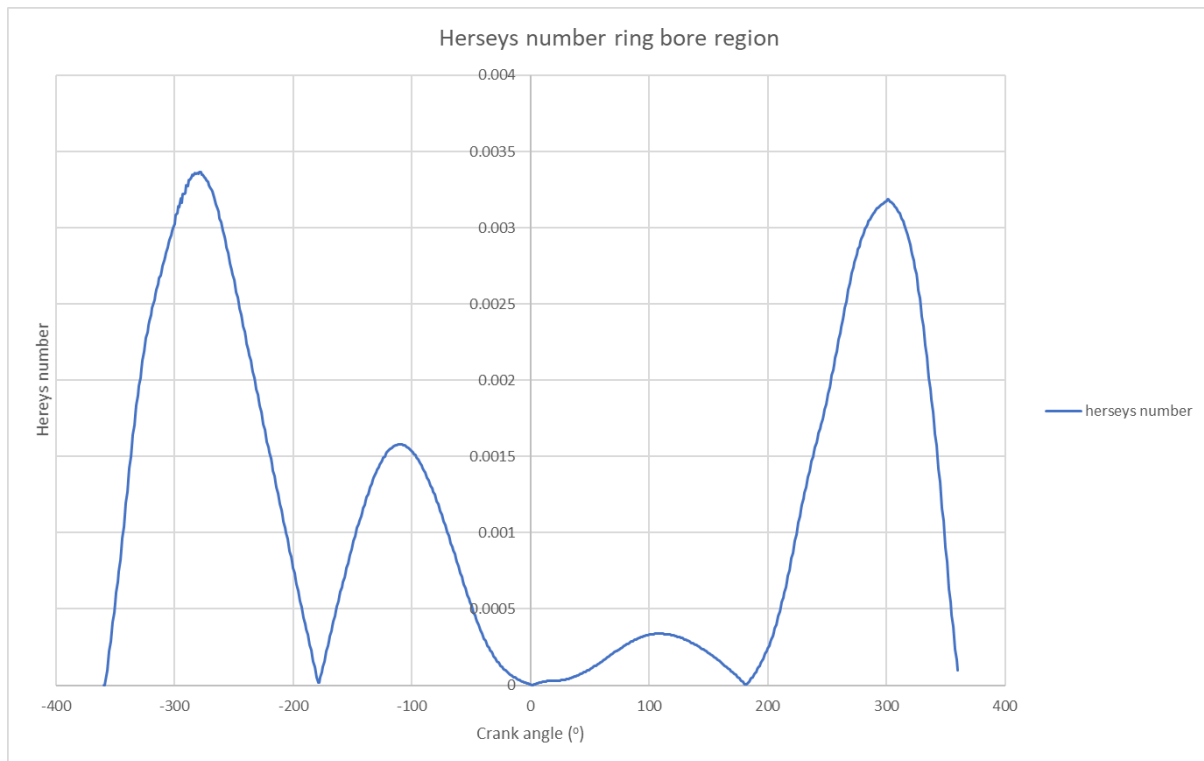


Figure 23 - Ring bore Hersey's number over a whole cycle.

What can be seen in Figure 23 is how Hersey's number varies across one full cycle, using a lubricant of a 5W-30 oil which is the general use lubricant for this engine (Ford_Motor_Company, 2019). What can be seen to be happening here is that between -360 to -180 degrees Hersey's number increases with the piston's relative velocity. This is also mirrored between 180 to 360 degrees. What can be seen to occur during the compression and expansion phases is more interesting, as the gas pressure increases on the compression phase the increased normal load sees Hersey's value halve. Then under the high pressure of combustion this value once again falls significantly. The points of interest here are, firstly the fact that whenever the speed of the piston falls to zero at TDC and BDC, the likelihood of hydrodynamic lubrication falls to zero. The second point being that the high loads under firing see the likelihood of hydrodynamic lubrication fall to by a factor of almost 6 times. Of course, this doesn't determine where in the lubricant regime any of these values lie, what it does though is set up a logic for assessing the changes being made. Furthermore, it is a way to quickly recognise how a change may affect lubricant regime. The need would arise for complex modelling or testing to truly determine the real regime of the lubrication if the need was there. So, what this means is that the same principle can be applied to the ultra-low viscosity lubricants and the geometry model to assess how each will affect lubricant regime.

There is only one more aspect to add the mix which is something that has not been addressed up to this point. So far all of the surfaces have been conceptualised as two flat surfaces. Where in reality each surface will be subject to relative roughness. The roughness

of a surface can be thought of by the way of asperities. An asperity is something that sticks out or an abnormality. As such any surface can be made of a continuous series of asperities. Because of this the surface will have a series of troughs and peaks. It is these asperities that would cause an issue with the lubricant regime. This is because the troughs and peaks effectively act as a reservoir to capture the lubricant preventing good hydro dynamic lubrication. It is for this reason that the composite roughness must be accounted for between the two surfaces. (Carden, 2014) approach this in their paper by establishing the normal distribution of the variation in asperity height, where then a mean roughness may be established. For the purpose of the Stribeck theory a new concept may be introduced, known as λ . Lambda is Hersey's number over the composite roughness (M. Knopf, 1998);

$$\lambda = H/CR$$

For this reason, from hence forth when making reference to the Stribeck theory it will be with respect to lambda rather than simply Hersey's number irrespective of whether roughness is a changing variable.

4.2 Piston bore with varied lubricant viscosity Stribeck theory

Given that the piston bore contact forces have been considered in depth previously it should be possible to now simultaneously evaluate the effect of geometric variables in tandem with the introduction of ultra-low viscosity lubrication. Figure 24 on the next page shows a close section view of the piston bore lambda values. This area of the graph was selected as, based upon the previous knowledge, it's accepted that close to TDC of the firing stroke is the area most critical when lubrication regime change may occur. It can be seen that as the piston approaches TDC the normal load increases as piston speed slows. At TDC the value rapidly changes from negative to positive, this is where the piston transitions from the anti-thrust side to the thrust side passing through zero. Knowing this, what can be determined about the characteristics of the lubrication? With respect to each of the oils, the 5W-30 appears to show far more likelihood of hydrodynamic lubrication due to the increased velocity. This is can be seen both on compression and initial expansion. What can be said then is that by changing the lubrication viscosity from the 5W-30 to the 0W-20 greatly reduces the likelihood of achieving hydrodynamic lubrication. Or more to the point, if the original engine was seeing adequate hydrodynamic lubrication regimes in the piston bore region, then this change in lubrication type presents a clear risk to the lubrication integrity. Using the Stribeck theory in this way allows the use of the real variable values to make rapid assessment how changes to the respective variable will affect the lubrication regime. Here it can be said that with the standard geometry with a lambda value reduction of approximately 8 times, there would be concern for asperity contact, increased friction and wear rate.

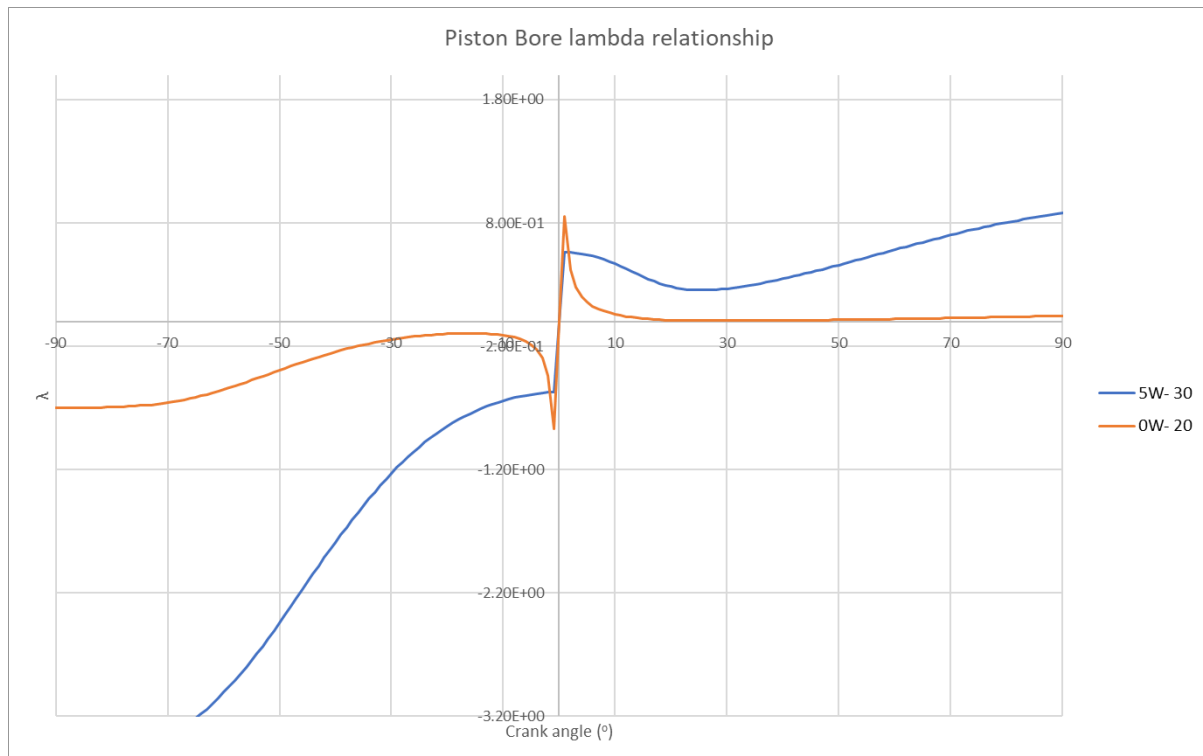


Figure 24 - Piston bore lambda value of standard inline crank geometry with respect to different viscosity lubrication.

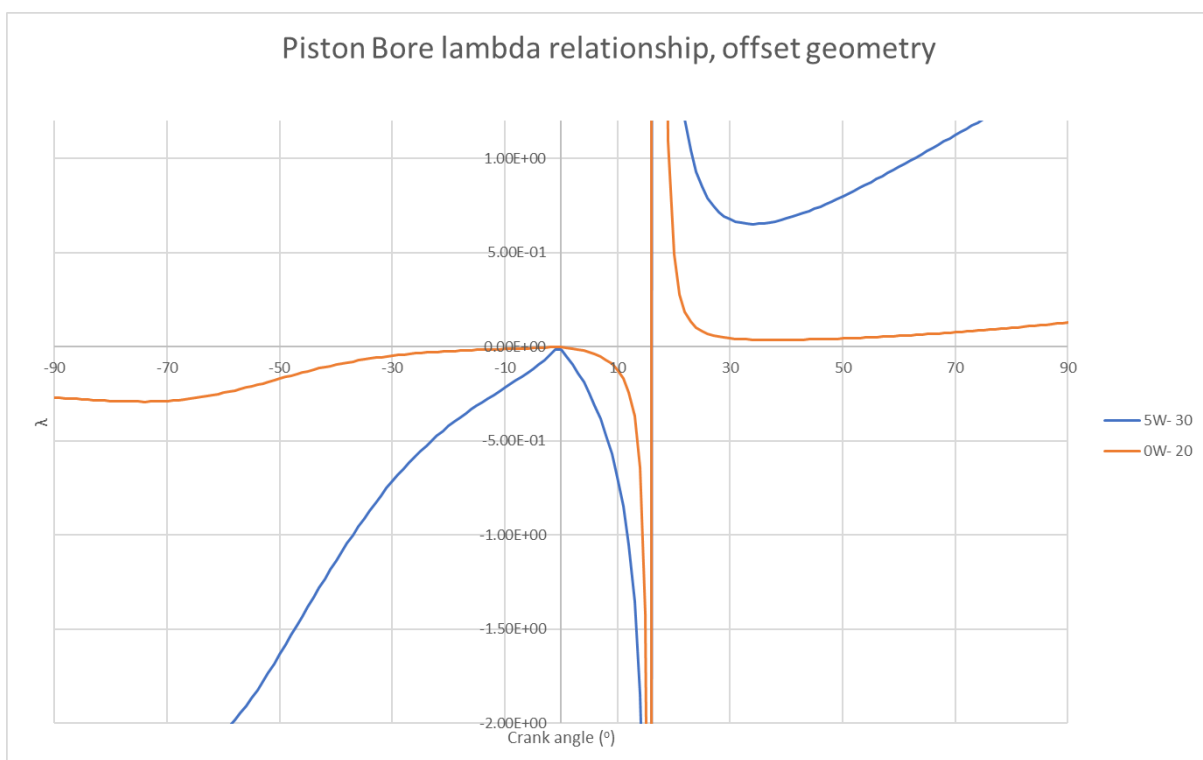


Figure 25 - Piston bore lambda value of offset crank geometry with respect to different viscosity lubrication.

Figure 25 now shows the same variables applied to the offset geometry crankshaft. As with the piston motion the complex nature of the offset crank geometry shows itself in the subtleties of the graph. In assessing the lubrication regimes, the offset crank geometry presents itself quite similarly to the standard geometry, upon compression and expansion. Where it shows clear differences is around TDC, which is where the subtle nature of the piston motion plays its role. As the piston approaches its relative TDC, the pistons velocity falls to zero which as per Stribeck theory means it falls out of hydrodynamic lubrication. However, this actually occurs on the anti-thrust side of the bore before it then starts to expand, still on the anti-thrust side. So, as the piston descends building velocity lambda increases proportionally, the crux being that as the conrod is passing through its transition the sin of the angle tends to zero with such acute angles of phi. This means that the normal load is very small here, greatly increasing the likelihood of hydrodynamic lubrication. On top of which this transition takes place at the very point that peak pressure is to be experienced, meaning that by the time the rod gains appreciable angle the loadings are already starting to reduce.

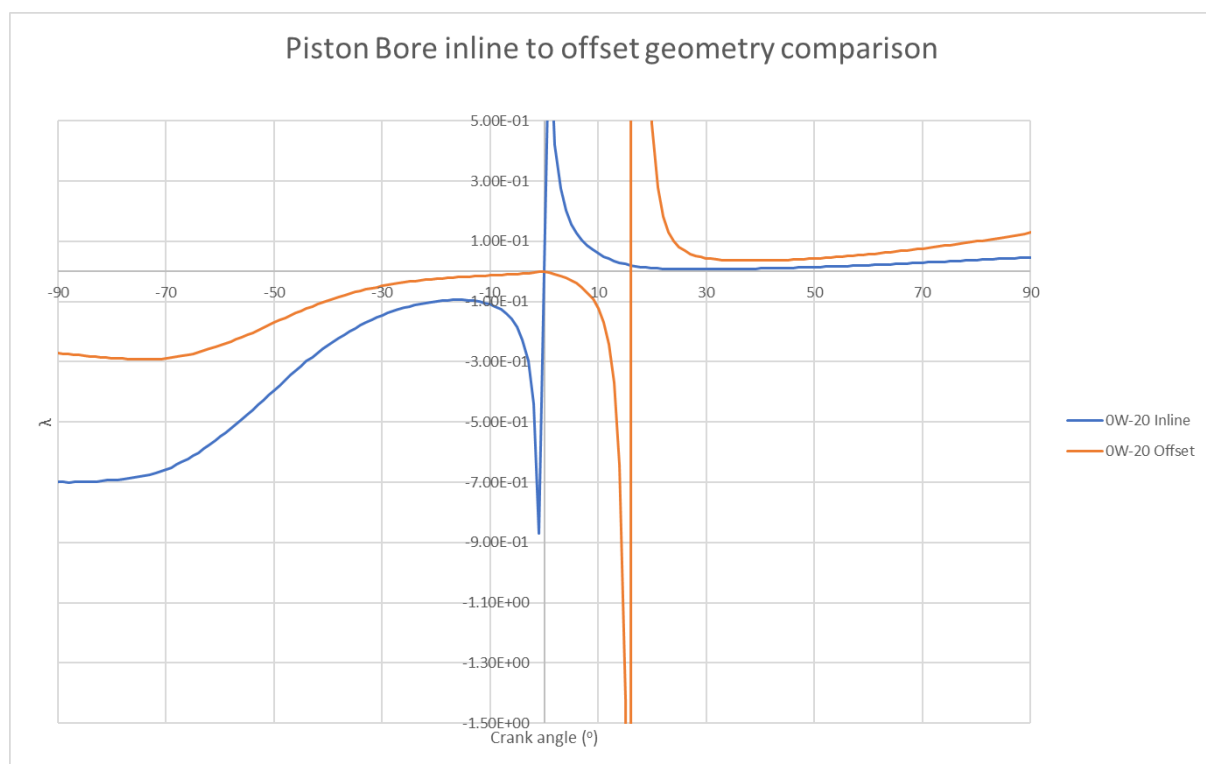


Figure 26 - Piston bore comparison of lambda with a single viscosity lubricant.

The comparison can be drawn between the two geometry sets, remembering that with the low viscosity lubricant the inline geometry saw a significant reduction in lambda value. Figure 26 now shows how each compare whilst utilising the low viscosity lubricant. Of course, given that there is greater side load on the anti-thrust side, it is expected to see worse performance for the lubricant here. However, the real area of interest is during the high-pressure region, between 10 to 30 degrees ATDC. This is where in comparison the offset crank geometry shows the potential for far better lubricant performance. This is even with the low viscosity lubricant, showing how the offset crankshaft is well suited to mitigate lubrication regime deterioration under high loads. There are several conclusions to be drawn from this. Combining this philosophy with that of the geometry shows that there is a bigger picture at play, one which wasn't approached in any one single report reviewed here. (GM Powertrain Division_Steve Shin, 2004) and (Musashi Institute of Technology_Kei Nakayama, 2000), express the functionality of the offset crank geometry to serve as a means of reducing side force, from which their conclusion was no real benefit other than its primary function highlighted. Whereas, (Carden, 2014), (Carden, 2014) and (Selby-Savant-Inc, 2010) consider the complexities of the Stribeck model and regimes of lubrication with no mention for any engineering compromises the use of low viscosity oils would require. What each of these reports fail to do is consider the power train in a holistic manner, such that the whole system is interconnected. What is being said then, is that the offset crankshaft plays a role in enabling the use of ultra-low viscosity lubricants. Each of the papers set out with the common goal of improving thermal efficiency through reduction in parasitic losses within the piston to bore region. Perhaps what (GM Powertrain Division_Steve Shin, 2004) failed to see in their research was that they saw no benefit to the offset crank geometry because it wasn't paired with a low viscosity lubricant, which would have reduced work done in shearing the oil itself, whereas the oil they had been using prior was already capable of maintaining hydrodynamic lubrication. Conversely, (Carden, 2014) is proposing a model for lubrication regimes but makes no consideration for the engine architecture requirements that come with the use of such ultra-low viscosity lubricants. Of course, there are multiple assumptions being made here, so that a system with so many variables can be compared in as simple a manner as possible, each of which has an impact on the accuracy of the direct results. It is however, reasonable to suggest this method is adequate for predicting trends and connections between said variables and the direction of the effect each change. The question now is, whether there further benefits that may be reaped by utilising ultra-low viscosity lubricants that may support the argument for offset crank geometry, to the point where the whole powertrain gains significant improvements in efficiency while retaining longevity and reliability.

4.3 Bearing loads

A similar approach may be taken when considering applying the Stribeck theory to the journal bearing in the engine's bottom end. By taking the net force and the side force present, it is possible to use them as the "X" and "Y" coordinates for a polar plot of the bearing loads at each degree of crank rotation.

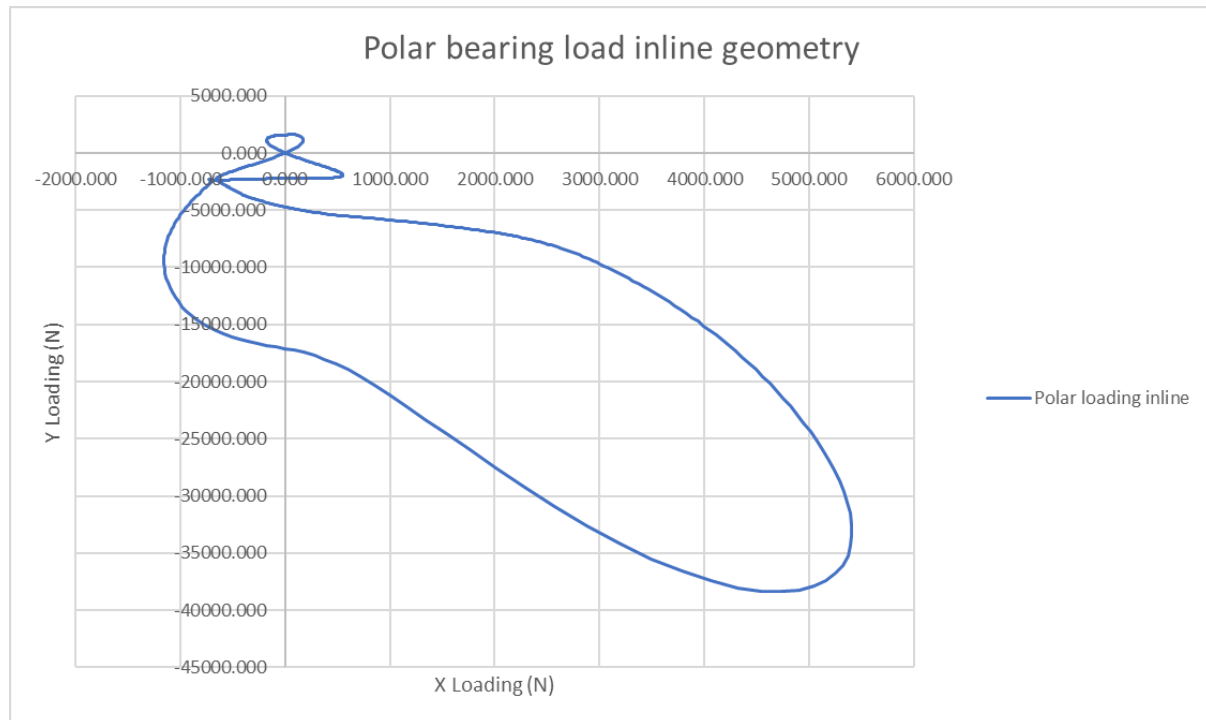


Figure 27 - Polar loading plot of inline main journal.

What can be seen is a representation of the magnitude and direction of the loading that the bearing experiences throughout the 4-stroke cycle. It has been assumed for this case that the crank is locally balanced, this is only to reduce the variables. It must be noted though, that there are variables such as this that will impact these loadings. What can be stated is the nature of the asymmetric loading that the bearing experiences because of the firing forces. It is this loading on the firing stroke that poses a risk to the lubrication integrity. Where now the loading is a composite of both the Net force acting vertically on the piston and the side force. The same can be seen in Figure 28 for the offset crank geometry. As the side loading is reduced the shape of the polar graph is more rounded in appearance. What this means is that the resultant angle of the force under firing is nearer vertical with offset crank geometry than it is with the conventional.

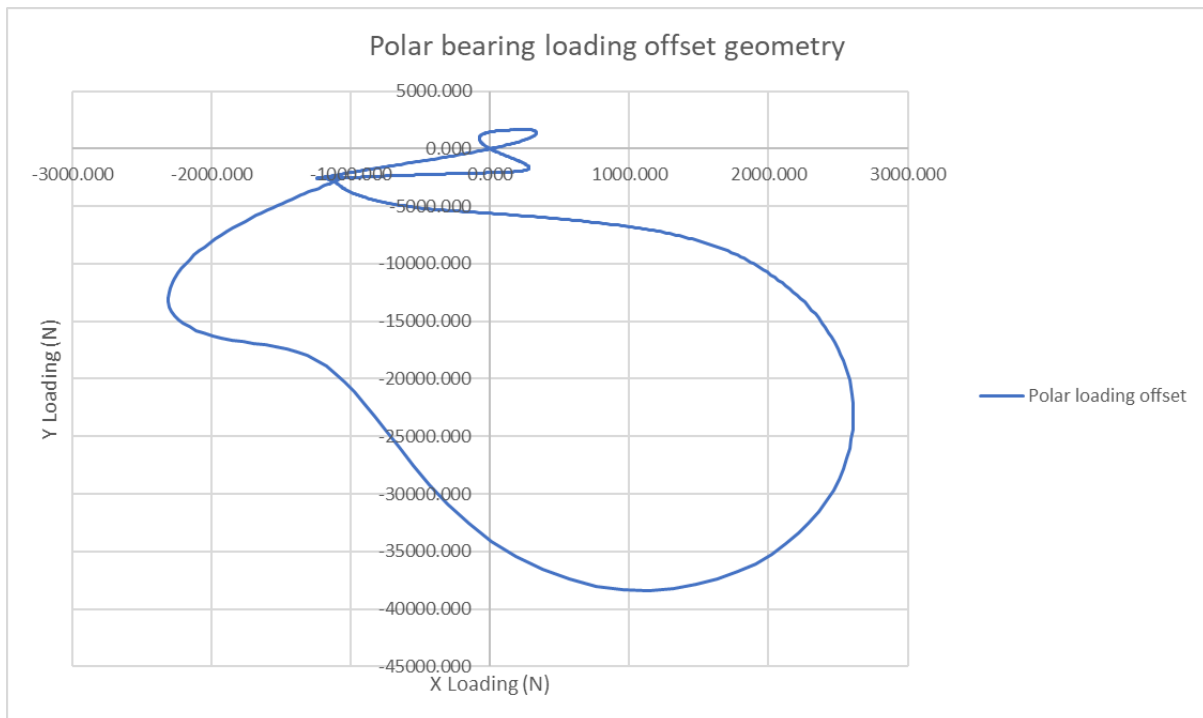


Figure 29 - Polar loading plot of offset main journal.

Although this plot gives an indication of the direction and relative magnitude of the component forces in one field of view. It is hard to get a feeling for the true scale of the forces acting normal the lubrication film. As the resultant force will always be normal to a point on the journal bearing, the absolute bearing load can be shown as a function of crank angle. This can be seen below, where the load appears strikingly similar to the cylinder pressure.

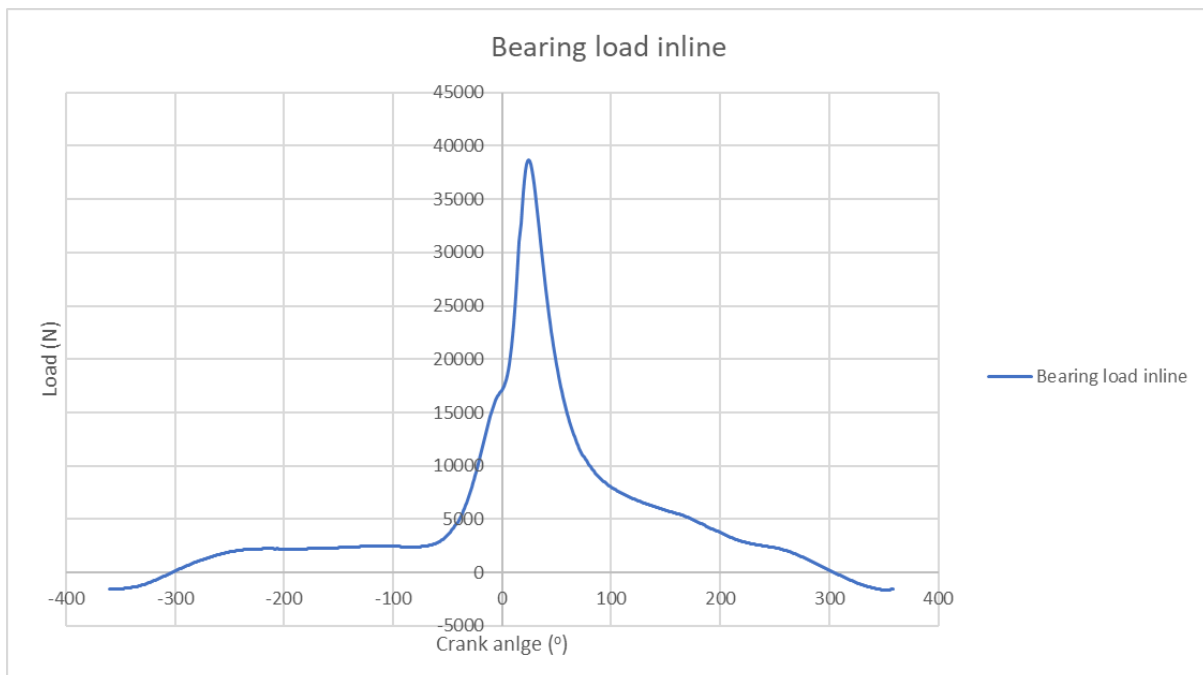


Figure 28 - Absolute bearing load inline geometry.

The true force in fact slightly greater than the gas force alone, it also becomes negative at points due to the inertial loading. It also transpires that the overall magnitude difference

between the inline and offset geometry for this case is negligible. The only noticeable difference between the two cases is the aforementioned resultant angle of the resultant load.

Applying once again the Stribeck theory to the main journal bearing of the inline geometry, it shows the relationship between the bearing load and the value for lambda.

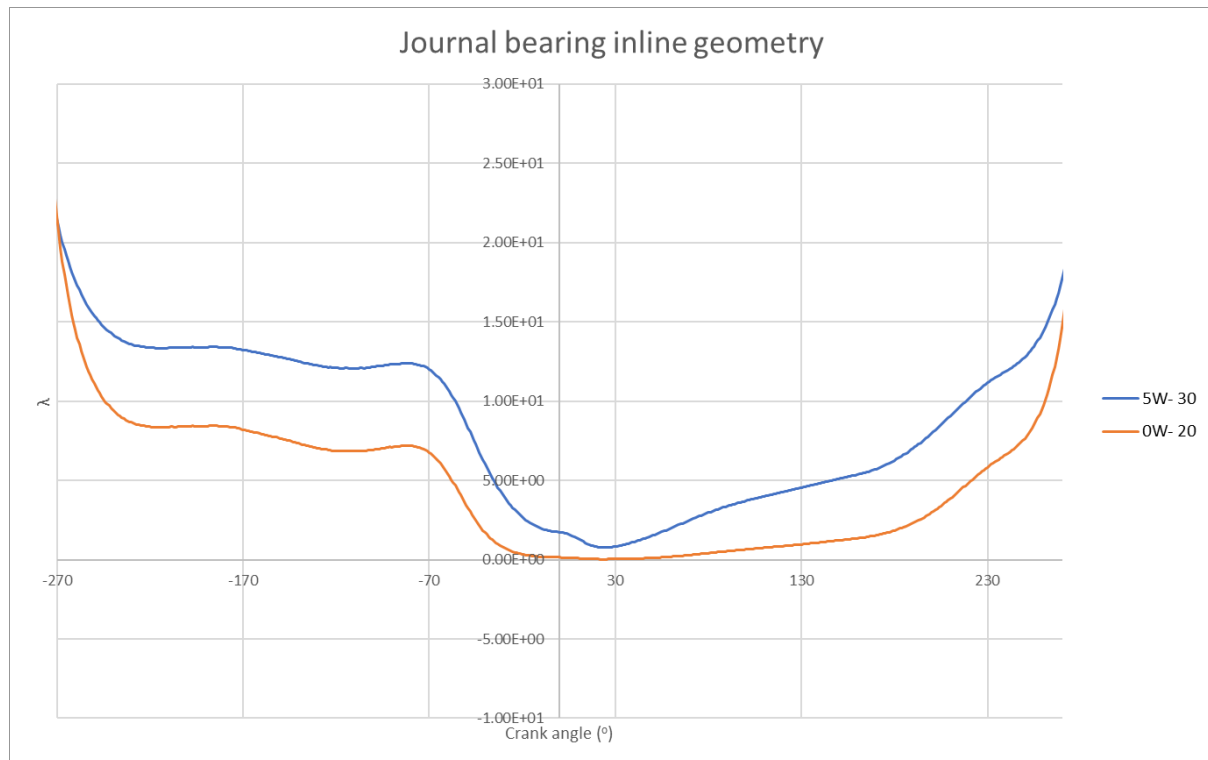


Figure 30 - Lambda load dependency at a fixed RPM site.

Given that the data is for a singular RPM site, the bearing maintains a constant speed of entrainment. As such, because the other variables are all dimensional, and the viscosity is fixed as before, the only thing that affects the likelihood of a given lubrication regime is the load. The observation is an expected one, in that as the load increases the likelihood of hydrodynamic lubrication decreases. This is not a surprising revelation based upon all the previous material presented in this paper. What is noticeable though, is the reduction in the likelihood based upon the change from the 5W-30 lubricant to the 0W-20. This shows a reduction of just over half, which means that if all else remained the same as change of lubrication runs an inherent risk. In this case Figure 31 on the next page shows that the offset geometry has no effect on the likelihood of lubrication regime. This means that unlike the piston bore area, the geometry change will not serve to support a transition to an ultra-low viscosity lubricant.

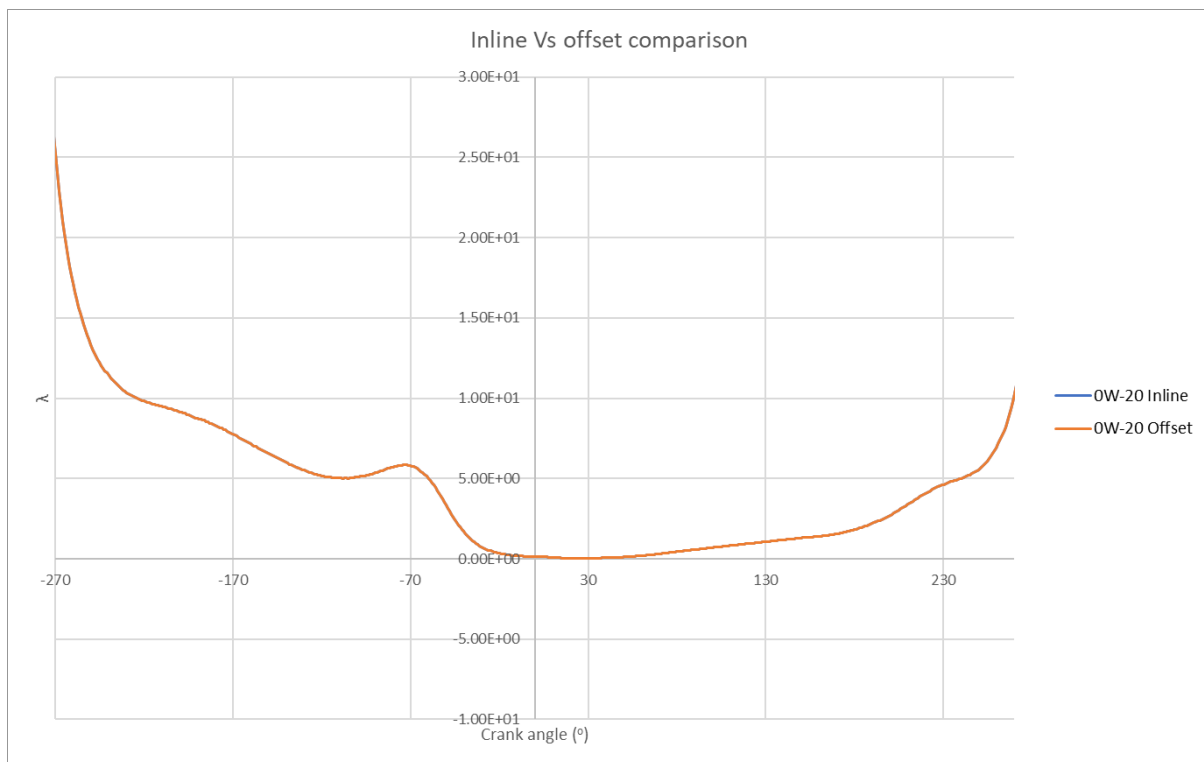


Figure 31 - Geometry comparison for low viscosity lubrication at the main journal bearing

The bearing load model along with the Stribeck analysis shows the area of concern for bearing lubrication. It also allows for consideration of bearing design. Given it is possible to calculate the angular resultant force that the bearing experiences, this can be used to work out the placement oil feed locations. The areas of peak load will see the highest hydrodynamic pressures, then knowing this location will mitigate oil starvation through reduced flow. Additionally, although the offset crank geometry offers no benefit to the bearing directly, it does shift the location of this resultant peak load so must be considered when designing a new bearing specification, which itself is the next consideration, due to the fact the offset crank geometry offers no relief for the bearing under the use of low viscosity lubrication. It must be said that further consideration for the bearing design would have to be made to ensure proper lubrication and longevity can be achieved.

5. Ultra-low viscosity influenced drive cycle simulation.

The initial goal was to understand the effect of ultra-low viscosity lubricants in a modern powertrain. It has been stated in almost every single paper cited in the literature review, that manufacturers are pushing to use even lower viscosities in their engines. The effect of doing so in terms of the risks and associated methods to cope with such lubricants has been shown. The question then, is why would an manufacturer want to use these ultra-low viscosity lubricants or justify making the modification suggested. Along with the consideration for a new proposed SAE 8 specification lubricant postulated by (Carden, 2014).

5.1 Thermo-viscous dependency

Both (Carden, 2014) and (Plumley, 2014) cite the use of the Walther equation for extrapolating the viscosity of the lubricant from its temperature. It was given as;

$$\log[\log(\mu + K)] = A + B \log T \text{ (Carden, 2014)}$$

From this, it can be shown that the viscosity is dependent on;

$$\log[\log(\mu + K)] - B \cdot \log T = A$$

3rd law of logs;

$$\log[\log(\mu + K)] - \log T^B = A$$

2nd law of logs;

$$\log \left[\frac{\log(\mu + K)}{T^B} \right] = A$$

Taking anti-logs of both sides;

$$\frac{\log(\mu + K)}{T^B} = 10^A$$

So;

$$\log(\mu + K) = 10^A T^B$$

Taking anti-logs;

$$\mu + K = 10^{10^A T^B}$$

Therefore;

$$\mu = 10^{10^A T^B} - K$$

If two viscosities are known at two temperatures, then the constants A and B can be found for each lubricant. Assuming that K is 0.6 in reference to the value quoted by (Carden, 2014). from this it is possible to show how each lubricant for the model changes with temperature.

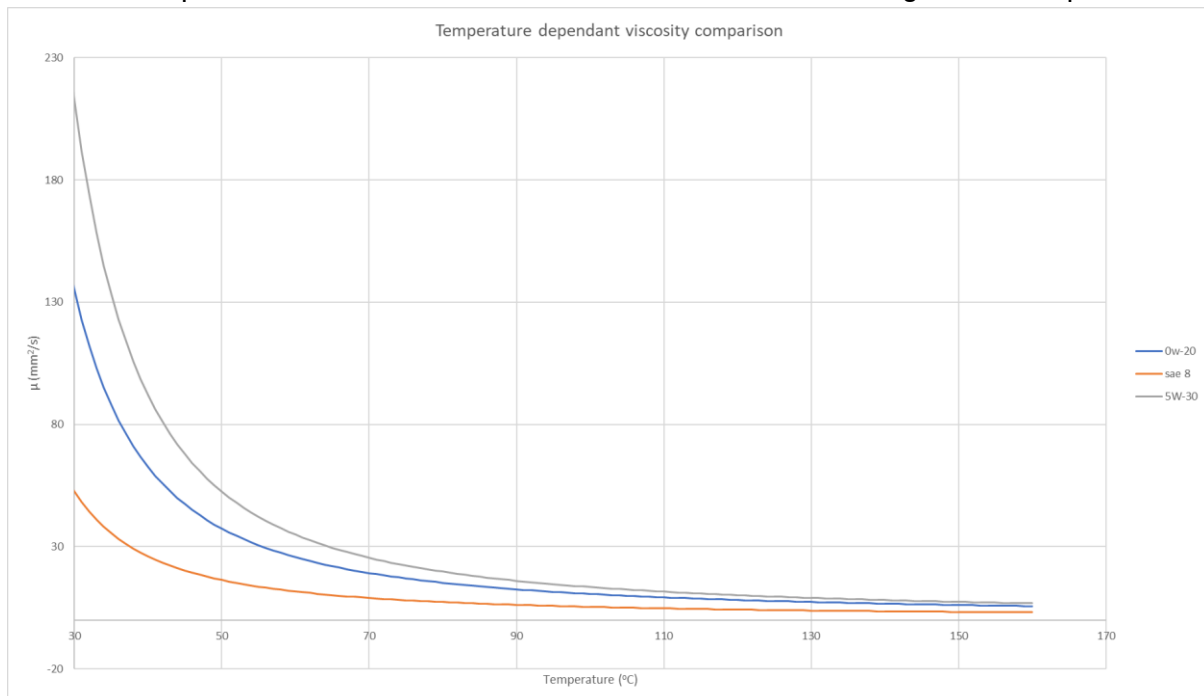


Figure 32 - Calculated viscosities using the Walther equation.

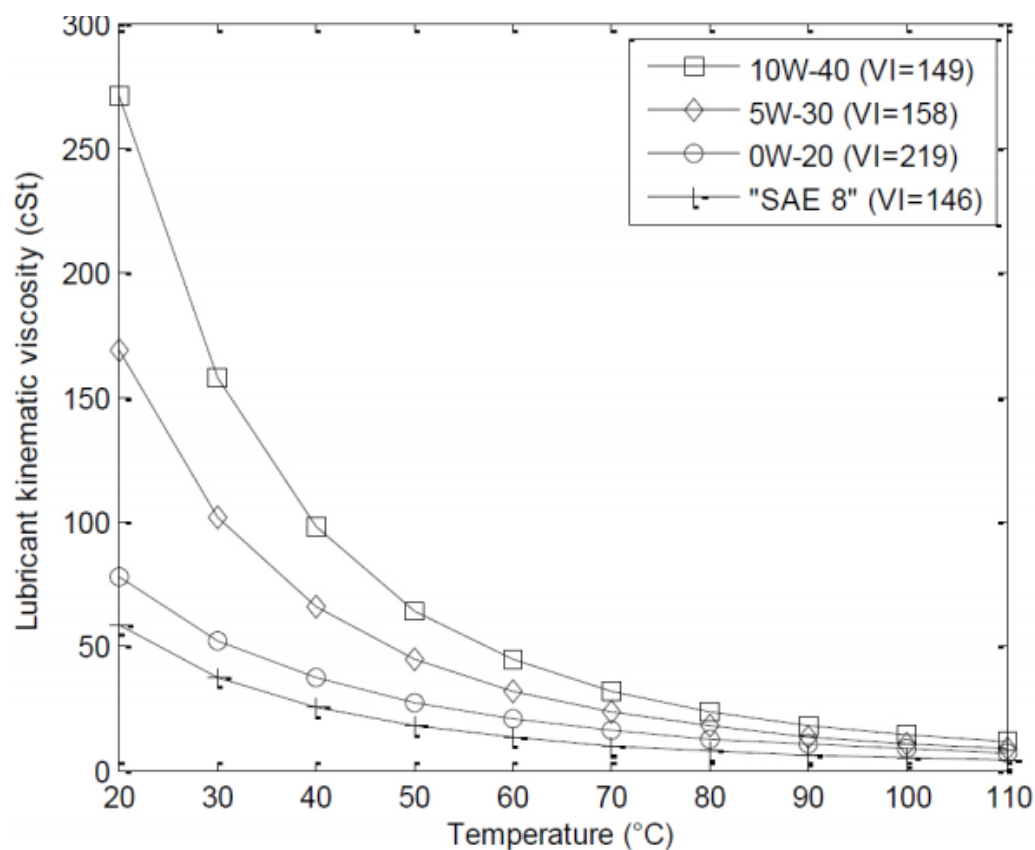


Figure 33 - Results of Walther equation according to; (Carden, 2014)

Both the calculated values and those by Carden et al, show the same trend with temperature. However, there is some discrepancy in the absolute values between the two. This is despite having used the same values for both the 0W-20 and SAE 8 lubricants.

	0W-20	“SAE 8”
Kinematic Viscosity at 100°C (cSt)	8.5	5.4
Kinematic Viscosity at 40°C (cSt)	36.9	26.1

Figure 34 - Lubricant reference data; (Carden, 2014)

Whether there are other factors that (Carden, 2014) used is unclear, however, it can be said that the method in use is well documented in other studies. As such there is sufficient confidence in the results presented here in the SI units rather than the more colloquial “Centistoke”.

What is worth making note of here is the profile of each of the lubricants, with the variation in temperature. The lubricants with the more viscous base oil, therefore, more viscous at lower temperature; has a much greater thermal dependency. Such that the viscosity falls more rapidly. Where, the viscosity of the SAE 8 oil is 20% that of the 5W-30 at 30 degrees Celsius but at 100 degrees Celsius it is 55% that of the 5W-30. What this shows is that the oils converge in their viscosity with temperature. This can be seen in the extremely Flat nature of the new proposed SAE 8 lubricant, which must utilise an extremely low viscosity base oil that is supported by viscosity modifiers. All of these oils possess properties that show the use of viscosity modifiers, particularly the lower viscosity lubricants. What can be said then is that although these ultra-low viscosity lubricants use low viscosity base oils they still seek to retain their viscous properties at higher temperatures. This is when considering absolute values rather than their proportionality. This would suggest that it is reduction in the low temperature viscosity that manufacturers are seeking.

5.2 Drive cycle simulation

To establish the real benefits of using these low viscosity lubricants it was imperative to create a simulation that shows how changing lubricant effects the real power consumption and thermal efficiency of the vehicle in a standardised format. For this reason, it was selected to use the NEDC drive cycle as the platform on which to base the simulation. The ford “Fox” platform was selected for this use, as the most comprehensive data was available for the purpose of building a simulation. For the NEDC test the vehicle speed is a fixed profile over the course of the test. Therefore, the acceleration is also fixed by the test rules, because of this the only variable is the force that must be delivered to the wheel to achieve this. There are many underlying variables that determine how much torque must be generated by the engine. If

$$F = ma$$

$$F = m \cdot \frac{d^2x}{dt^2}$$

where the mass is that of the vehicle and the acceleration is given as stated. So, the torque is the force multiplied by the wheel radius. With the addition of the wheels;

$$T = I\alpha$$

Where;

$$a = r \cdot \frac{d^2\theta}{dt^2}$$

So, the torque that must be delivered to the wheels to accelerate the vehicle and all four wheels is;

$$T = 4I \cdot \frac{d^2\theta}{dt^2} + m \cdot r \cdot \frac{d^2x}{dt^2}$$

Or;

$$T = \frac{4I}{r} \cdot \frac{d^2x}{dt^2} + m \cdot r \cdot \frac{d^2x}{dt^2}$$

This is however, the torque that must be delivered to the wheel, the engine must deliver torque through a transmission and differential which have torque multiplication ratios and their own inherent moments of inertia. Each of the components within the drive train was taken into account, to calculate the torque that must be delivered by the engine to satisfy the test conditions. Further to this, air and rolling resistance must be incorporated into the model such that an accurate total force is calculated. Once the physical properties of the vehicle were

modelled and the torque required of the engine has been determined, then the thermal efficiencies of the engine must be considered. (Bosch_Ford, 2019), published the calibration data for the power-plant in question. Each table is torque demand based as shown below.

FUELMASS kg/h	800	1000	1250	1500	1750	2000	2250	2500	3000	3500	4000	4500	5000	5500	6000	6500
FL		3.24	5.22	7.52	9.39	11.35	12.92	14.34	18.08	18.92	21.54	22.34	25.55	26.61	27.62	25.11
200						11.34	12.96	14.25	17.78	19.47						
190						10.77	12.04	13.39	16.91	18.46						
180					9.11	10.05	11.26	12.52	15.24	17.76	20.61					
170				7.51	8.77	9.38	10.52	11.72	14.06	17.14	18.91	21.41				
160				7.10	8.08	8.74	9.77	10.88	13.05	15.06	17.46	19.80	23.68			
150				6.52	7.46	8.17	9.12	10.19	11.98	14.00	16.18	18.15	20.99	24.47		
140			5.19	6.04	6.84	7.63	8.45	9.44	11.01	12.93	14.87	16.65	18.92	21.63	25.58	
130			4.73	5.46	6.16	6.96	7.73	8.55	10.05	11.89	13.84	15.40	17.30	19.71	22.93	
120			4.35	4.93	5.75	6.39	7.07	7.79	9.15	10.95	12.86	14.24	15.88	18.00	20.45	23.30
110		3.22	3.91	4.52	5.24	5.78	6.43	7.13	8.42	9.97	11.65	13.06	14.58	16.31	18.66	20.90
100		2.97	3.43	4.13	4.66	5.20	5.78	6.48	7.71	9.03	10.53	11.88	13.38	15.00	16.78	19.21
90		2.56	3.00	3.60	4.11	4.55	5.13	5.77	6.90	8.05	9.36	10.70	12.11	13.59	15.17	16.95
80		2.20	2.66	3.15	3.59	4.05	4.55	5.13	6.11	7.13	8.39	9.56	10.95	12.32	13.64	15.25
70		1.89	2.33	2.77	3.21	3.60	4.07	4.56	5.40	6.32	7.45	8.44	9.71	10.92	12.18	13.69
60		1.63	2.02	2.42	2.78	3.12	3.54	4.02	4.75	5.54	6.50	7.39	8.51	9.60	10.76	12.04
50		1.38	1.73	2.08	2.38	2.70	3.03	3.41	4.09	4.80	5.60	6.41	7.31	8.25	9.33	10.42
40	1.04	1.16	1.46	1.76	2.01	2.28	2.57	2.87	3.46	4.10	4.77	5.48	6.27	7.10	7.94	8.92
30	0.89	0.95	1.19	1.42	1.67	1.88	2.13	2.36	2.85	3.41	3.94	4.55	5.21	5.94	6.62	7.51
20	0.61	0.73	0.93	1.10	1.30	1.48	1.65	1.84	2.21	2.67	3.13	3.64	4.18	4.76	5.37	6.08
10	0.44	0.51	0.65	0.78	0.93	1.05	1.16	1.29	1.63	1.92	2.38	2.72	3.18	3.64	4.18	4.79
5	0.36	0.42	0.53	0.63	0.79	0.88	0.98	1.09	1.32	1.60	1.99	2.33	2.73	3.14	3.56	4.08
2	0.32	0.39	0.46	0.56	0.66	0.78	0.86	0.98	1.22	1.44	1.83	2.09				

Figure 36 - Fuel mass flow torque-based table; (Bosch_Ford, 2019)

	800	1000	1250	1500	1750	2000	2250	2500	3000	3500	4000	4500	5000	5500	6000	6500
FL		108.9	145.0	174.1	190.5	201.6	200.7	199.8	200.4	200.4	190.5	176.4	165.8	154.7	145.7	125.6
200						201.7	201.1	198.7	199.1	200.0						
190						190.5	188.2	187.2	188.2	188.3						
180					184.6	178.9	178.6	175.2	177.9	178.1	182.4					
170				175.0	176.5	168.7	167.7	166.1	169.5	166.9	172.7	172.3				
160				166.2	165.3	158.3	157.6	155.1	160.0	160.5	162.3	159.9	160.0			
150				154.0	154.1	150.9	147.6	146.6	148.6	150.4	151.4	149.7	149.4	148.8		
140			144.7	142.3	142.0	139.3	138.4	138.9	140.0	141.0	141.9	141.4	140.5	139.6	139.4	
130			131.1	130.3	128.9	129.0	129.5	131.7	131.9	130.4	133.0	131.1	131.0	129.1	129.9	
120			119.5	121.6	122.1	120.6	121.3	122.1	121.1	121.6	124.3	121.2	119.8	119.2	118.6	119.2
110		108.5	109.7	112.3	113.0	111.3	111.3	112.2	111.0	111.2	112.7	111.2	109.8	108.6	108.3	107.1
100		99.8	98.7	102.5	102.6	100.8	102.0	101.3	101.3	101.5	100.4	100.6	99.7	98.7	97.4	
90		90.5	88.9	92.4	92.7	90.2	90.1	89.8	89.8	90.0	90.0	90.4	89.7	89.3	88.9	
80		81.3	79.9	82.0	81.3	80.3	79.8	79.6	79.2	80.2	79.7	80.8	80.7	79.9	79.9	
70		70.8	70.8	72.1	72.4	70.5	70.9	71.6	70.0	69.4	70.4	69.2	70.6	70.4	69.9	70.4
60		60.7	60.7	62.4	62.7	60.4	60.6	61.9	60.5	60.1	60.1	58.8	60.3	60.4	59.9	59.9
50		51.8	51.6	52.8	52.4	51.2	50.3	50.5	50.7	50.6	50.0	49.2	49.9	50.2	50.0	49.5
40	44.4	43.8	42.5	42.9	42.0	40.6	40.8	40.3	40.4	40.6	39.4	39.0	39.6	40.2	39.8	39.6
30	38.0	33.7	33.5	32.1	32.1	30.7	30.3	30.2	30.7	30.9	29.3	28.9	28.8	29.6	29.3	29.4
20	25.1	24.0	23.6	22.7	22.1	20.8	20.8	21.4	19.9	21.5	19.2	18.6	17.8	19.4	18.9	19.3
10	14.7	12.6	13.0	12.4	12.7	12.0	10.8	9.9	10.7	9.9	9.9	8.4	6.6	8.7	9.1	10.4
5	9.0	7.6	7.8	7.5	8.2	7.5	6.6	5.8	6.0	5.4	5.8	3.6	2.1	4.2	3.4	5.8
2	5.1	6.4	5.3	5.2	5.1	5.4	4.1	3.7	4.5	3.4	4.0	1.5				

Figure 35 - Torque actual based on torque demand; (Bosch_Ford, 2019)

These two tables contain all of the information required determine the thermal efficiency of the powertrain. By taking the torque and converting each site to a power output, knowing;

$$WD = F \times D$$

Then for the rotational frame of reference;

$$WD/rev = 2\pi.R.F$$

So;

$$WD/rev = 2\pi.T$$

So, the rate of work done being power;

$$WD/s = 2\pi.T \cdot \frac{RPM}{60}$$

Given that the power of the engine is known at each site as is the fuel flow, it is possible to evaluate the fuel power versus the actual power created.

$$\frac{P_{actual}}{P_{fuel}} \cdot 100 = \eta$$

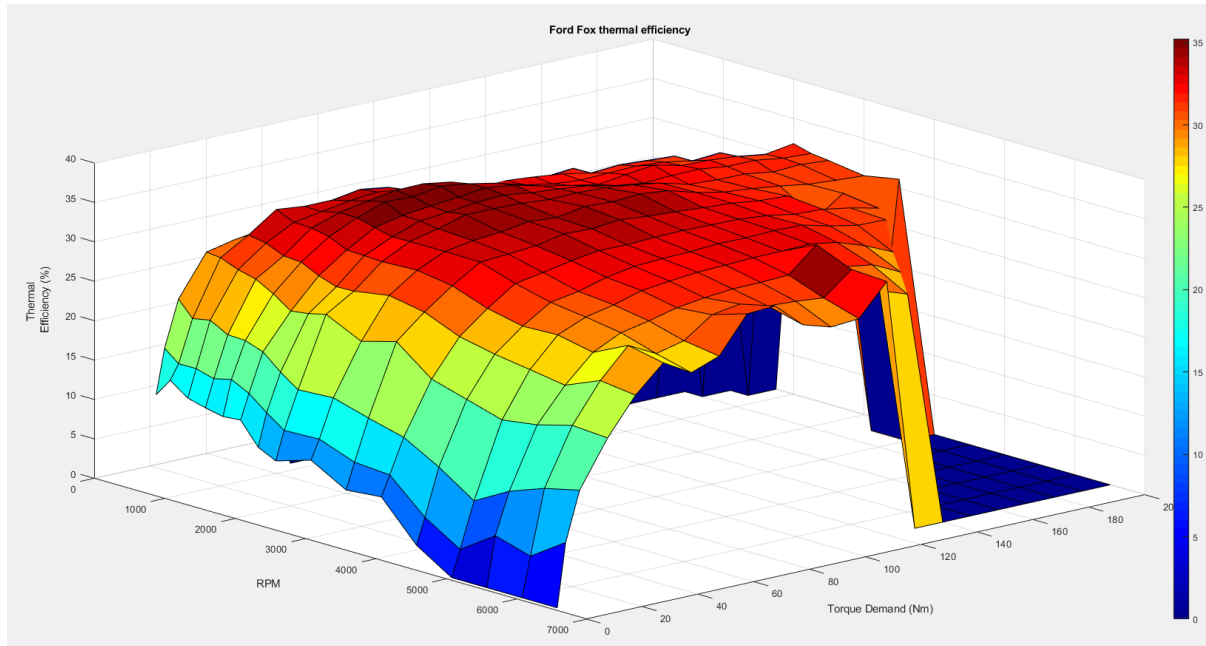


Figure 37 - Ford Fox thermal efficiency

Figure 37 shows a graphical representation of the resulting efficiency table, where the assumed LHV for petrol was 44 MJ/Kg. Now the model knows the torque demanded of the powertrain, from which the above can be used to determine the thermal efficiency of the engine given its reference to a specific site. From this the power output of the engine can be derived along with the fuel power required at each second of the drive cycle. The ultimate result of which is it is possible to determine the total energy used across the entire cycle and therefore, the quantity of fuel used.

Where the oil model comes into the drive cycle simulation is in the parasitic losses due to the shear work performed on the oil in the main and big end journal bearings. The principle follows thus

If the oil film remains attached at the boundaries of each of the surfaces in opposed motion, then the velocity at each boundary is zero. This means that there must be a velocity gradient within the lubricant, creating shear stress between the layers of fluid flow;

$$\text{Velocity gradient} = \frac{r \cdot \omega}{s}$$

where r is the radius of the journal, ω is the angular velocity difference between the two surfaces and s is the oil film thickness. The shear stress within the lubricant is

$$\tau = \frac{\mu \cdot V}{s}$$

and $V = r \cdot \omega$

Given that the geometry of the journal bearing shall be considered as a uniform cylinder, for which the reference area is;

$$A = \pi \cdot d \cdot L$$

Knowing this then, the resistive force at the journal is;

$$F = \frac{\mu \cdot V}{s} \times \pi \cdot d \cdot L$$

So;

$$F = \frac{\mu \cdot r \cdot \omega}{s} \times \pi \cdot d \cdot L$$

Where;

$$F = \frac{\mu \cdot r \cdot \omega}{s} \times 2\pi \cdot r \cdot L$$

Therefore;

$$F = \frac{\mu \cdot \omega}{s} \times 2\pi \cdot r^2 \cdot L$$

Again, given that;

$$WD = F \times D$$

Then;

$$WD = \frac{\mu \cdot \omega}{s} \times 2\pi \cdot r^2 \cdot L \times 2\pi \cdot r$$

QED;

$$WD = \frac{4\pi^2 \cdot r^3 \cdot L \cdot \mu \cdot \omega}{s}$$

Where the power loss is then;

$$P = \frac{4\pi^2 \cdot r^3 \cdot L \cdot \mu \cdot \omega}{s} \times \text{RPM} / 60$$

What this says then is that the power loss is proportional to the viscosity of the lubricant, but also the dimensions of the journal. Despite this insight being significant, more could potentially be saved by altering the dimension than the viscosity. The dimensions will remain fixed for exploring solely the lubricants impact on these parasitic losses. The way this was then implemented into the drive cycle was calculating the proportion of the power loss through the oil to the delivered power of the engine, to calculate what the thermal efficiency without the

oil. What this provides for is a negative of the oil effect, which can be applied at the end to calculate the total energy used including the oil. The 5W-30 was used as a reference, given this is the oil used by the manufacture.

The last piece of the puzzle was to account for the thermal loading of the oil. As it has been shown that the temperature oil has a huge impact on the viscosity and the viscosity is exactly what is being considered in terms of the equation for power loss. Then knowing how the oil may heat throughout the drive cycle is critical to determining the performance of the lubricant. As such the specific heat capacity of the oil was measured at an average of

2.23 KJ/Kg.K with an oil mass of 3.49 Kg (Ford_Motor_Company, 2019). The cylinder head and cylinder block mass were estimated at approximately 70Kg of aluminium, with a specific heat capacity of 0.91 KJ/Kg.K (engineeringtoolbox, 2019).

From this then it is possible to determine the rate at which the combined mass of engine and oil will increase in temperature, Knowing that

$$Q = M \cdot C_v \Delta T \quad (\text{Kambe, 2007})$$

So;

$$\Delta T = \frac{Q}{M \cdot C_v}$$

Where Q is equal to the power lost to the oil and engine for each second of cycle run time.

It is at this point having established the methodology used for this simulation, that it must be noted that is clear that there are a great number of assumptions being made here including some of the physical properties of the engine. This will indeed impact the true and absolute accuracy of the final result in terms of the energy used. This was accepted for this simulation, as it was not the subject of the calculations. What can be said is the mathematical theory behind the model is quite clear and accurate, so it can be said as a means of comparison of the effect different lubricant types, it is capable of providing accurate proportional results. As such it is with confidence that the following results are presented.

Firstly, to compare the rate of heating of the lubricant in the engine, below is shown the temperature over the course of the drive cycle for the original 5W-30 oil, 0W-20 and SAE 8.

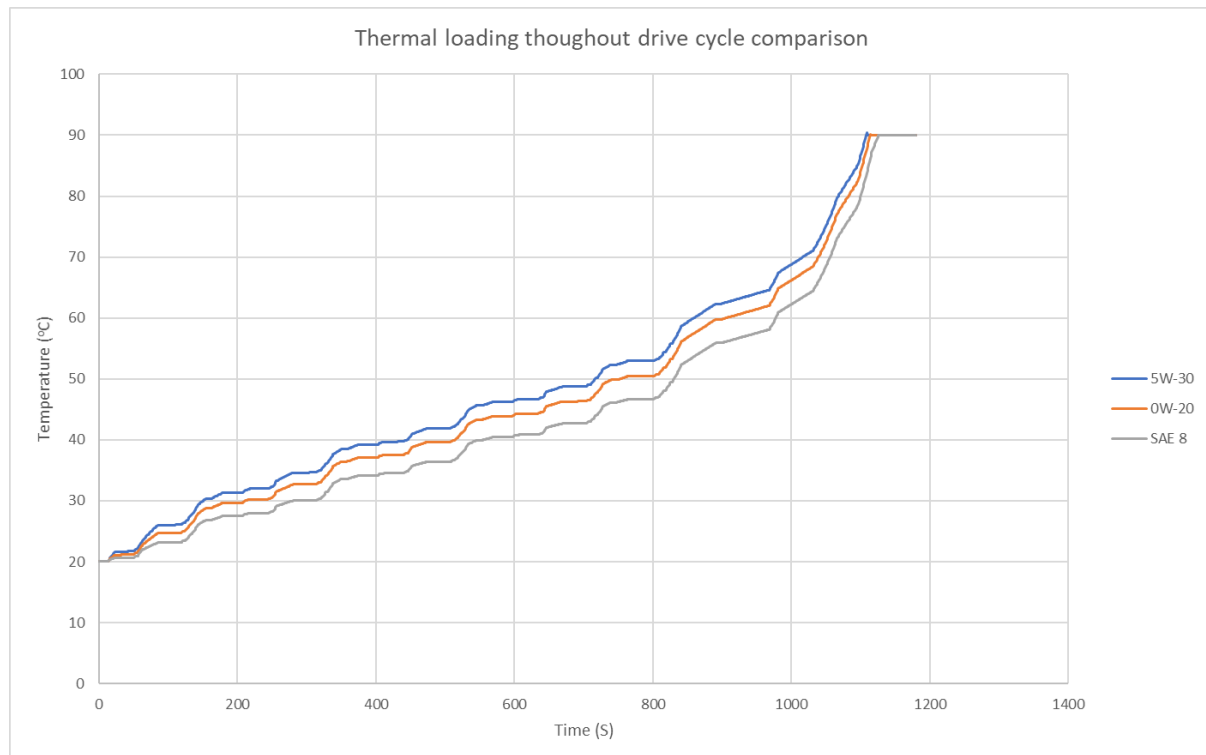


Figure 38 - Thermal loading profile of the three lubricants under examination.

The three lubricants appear to heat over the period of the test, only reaching their ultimate operating temperature at 95% completion of the test.

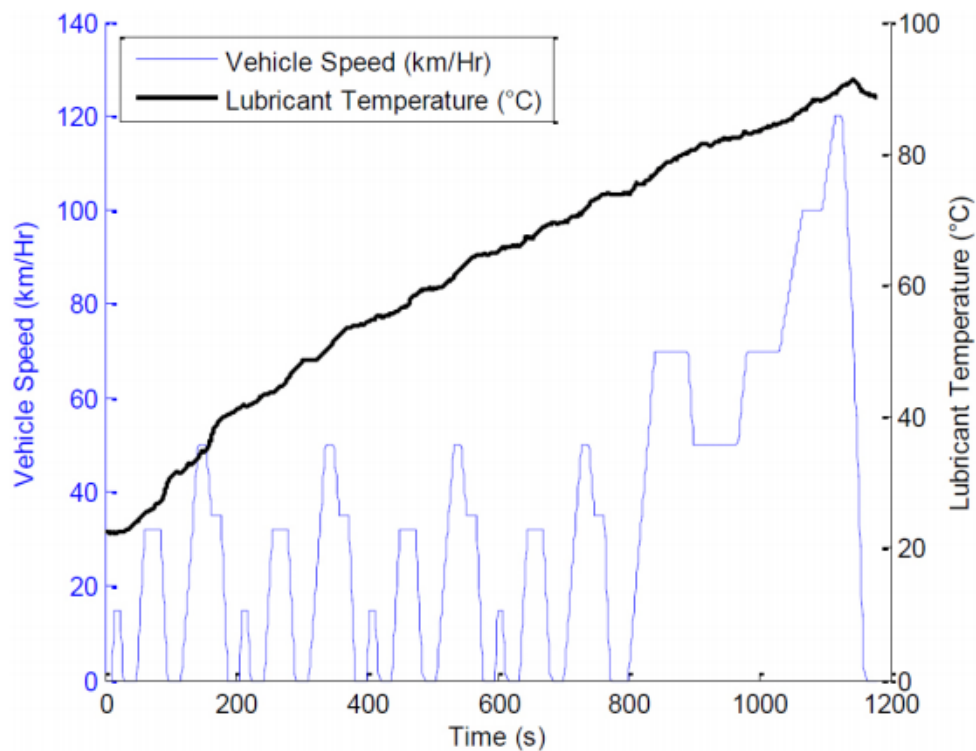


Figure 39 - Thermal loading profile for a 1.6 litre petrol vehicle; (Carden, 2014)

What can be seen then is that the simulated thermal loading profile follows that of the measured by (Carden, 2014), which reaches operating temperature at 97% of test completion. This means that there is only a 2% difference in reaching operating temperature between that of the simulation and the measured value presented in the research paper. Of course, the absolute values are not identical, as the vehicles are not the same. What can be said though is that, if the two temperature profiles are similar, then the progression in visco-temperature relationship must also be the same. The culmination of this then is to compare the overall energy used in each of the drive cycle simulations and evaluate whether the different lubricants were of any benefit.

5W-30		
Total DC WD	4.719796557	MJ
Oil DC WD	0.298723772	MJ
Total Fuel Mass	0.35568069	Kg
Total Fuel Vol	0.47424092	L
MPG	65.53535343	
Fuel / 100 Km	4.311281091	L

Figure 40 - 5W-30 drive cycle results.

The total energy used with the original lubricant was 4.72 MJ (2.p.d) where the losses to overcome work on the oil in the crank was 0.30 MJ (2.d.p). Meaning that the oil energy represents 6.3% of the total energy required to complete the drive cycle.

OW-20		
Total DC WD	4.618009758	MJ
Oil DC WD	0.224812844	MJ
Total Fuel Mass	0.344109357	Kg
Total Fuel Vol	0.458812476	L
MPG	67.73910465	
Fuel / 100 Km	4.171022506	L

Figure 41 – OW-20 drive cycle results.

The total energy used with the original lubricant was 4.62 MJ (2.p.d) where the losses to overcome work on the oil in the crank was 0.22 MJ (2.d.p). Meaning that the oil energy represents 4.9% of the total energy required to complete the drive cycle. So, not only did the

0W-20 oil yield a reduction in energy use of 2.16% but the oil also represented less of the energy lost as a proportion, with a 3.25% reduction in fuel used

SAE 8		
Total DC WD	4.453696818	MJ
Oil DC WD	0.104665489	MJ
Total Fuel Mass	0.325590646	Kg
Total Fuel Vol	0.434120861	L
MPG	71.59192079	
Fuel / 100 Km	3.946553283	L

Figure 42 - SAE 8 drive cycle results.

Finally, the total energy used to complete the drive cycle is 4.45 MJ (2.d.p), where the oil work losses now represent just 2.35% of the total, which is a 5.64% reduction in energy usage over the whole drive cycle in comparison to the 5W-30 oil, with an 8.46% reduction in fuel used. Given that the drive cycle is typically achieved using a fuel equivalence ratio of 1 due to the catalyst involvement, these figures also represent the same saving in carbon dioxide over the whole cycle.

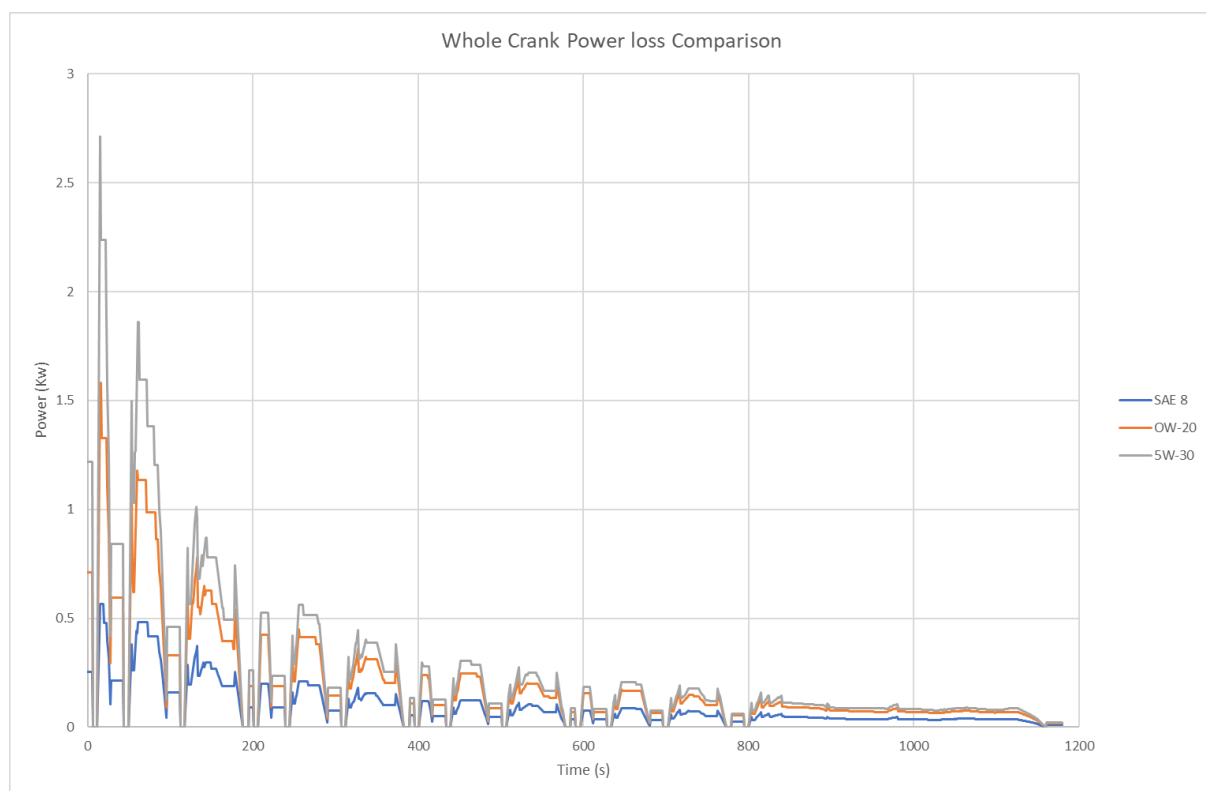


Figure 43 - Oil power loss of the whole crank comparison of each lubricant.

Figure 43 shows the parasitic losses of the whole crank assembly over the entire drive cycle, for each of the lubricants being considered. What is evident immediately is that the profile of each the cycles is that of the respective lubricants visco-temperatrue dependency curve. This is exactly what was expected and shows that the model behaves as is mathematically expected.

It can be seen then, that the critical area for the parasitic losses is at the beginning of the test, when the lubricant temperatures are low. It is here that also the SAE 8 ultra-low viscosity lubricant shows its real benefit, in that at low temperatures the viscosity is low enough to reduce the losses by 4.5 times that of the 5W-30. Towards the end of the test where the lubricants are close to their operating temperatures, they don't see as much of an advantage, although some gain is still evident. It is clear then that it is the flat profile of the SAE 8 lubricants viscosity temperature demands that gives it the advantage over the standard oil. The 0W-20 still showing significant reductions of the 5W-30 lubricant, proves that any reduction in base oil viscosity can proves beneficial. It would be this type of low viscosity base oil, supported by viscosity modifiers, that would give a manufacturer an undeniable advantage when bringing a new generation engine to test.

6. Conclusion

In the summation of this work, it is important to focus on the overlying narrative that each of its constituents tell. At the conception of this work it was to engage the question as to why manufacturers were pursuing the avenues that seemed to be new trends. These entailed the use of lower viscosity lubricants by some manufacturers and the use of and the exploration of new engine architecture by others. The questions throughout were why are these changes being made, for what reasons and why are they justified? The theme of this report has been a holistic approach to the ideas held within each of these individual questions. This means that although each one appears to be an isolated study, in fact they each affect one another in such an intrinsic way that it would be remiss not to consider them in unity.

What has been shown herein is, that when it appears that one thing doesn't have the effect that was sought that it doesn't mean that it is not of benefit. (GM Powertrain Division_Steve Shin, 2004) stated that offset crank geometry had no improvement in performance, then it was shown that in fact with an offset crankshaft, that reduction in torque due to geometry negates the reduction in side force friction. It could be though that this is where this sort of development would end. However, combining this with Stribeck theory as purported by many of the papers cited went on to show the benefits of the offset geometry. The real benefits come from the ability it gives the developers to then utilize low viscosity lubricants, without the generally accepted risks of using ultra-low viscosity lubricants. These

are the increased risk of asperity contact, degradation of the hydrodynamic lubricant regime, increased friction and therefore, increased wear. To justify the use of such lubricant types, would demand a drastic modification to the engine architecture. It was then shown that such a lubricant can yield significant gains in engine efficiency. Reduction in engine efficiency that came from the parasitic losses that come from the act of using the lubricant itself, can be mitigated by use of carefully engineered lubricants, that utilize low viscosity base oils and advanced viscosity modifiers to reduce cold lubricant losses. This then completes the loop, as the benefits that these lubricants show then validate and justify the use of offset crank geometry to support their use, all tied together by Stribeck's original theory of lubricant regime.

In all it's this holistic approach that is important in identifying what is important in terms of the key variables to drive forward the modern combustion engine. It is the overview and engineering consideration for the system as a whole that allows for such critical assessment of the symbiotic relationship that each single component has with every other. Only by understanding the what links are inherent between the lubrication and the engine architecture in this case, was it possible to see the advantages of the development to the engine geometry, in the face of research that expressed that there was no gain in its use and taken at face value would have otherwise been proved correct.

There is far more to be considered than that contained in this work, there are far more variables and assumptions to be accounted for. But for such a broad topic area that was to be expected. It would be prudent to engage in further work in all areas covered, to gain a more in-depth outlook, to remove the shroud cast by the assumptions made here and provide a more accurate analysis. This includes the rheological performance of the lubricants, because what has been modelled within this report is not the full and entire picture of the lubricant's performance. This then ought to be supported by experimental data from tests based on the hypothesis ventured in this report.

References

- Bosch_Ford. (2019). *Fox_GTDI_CalVal_Bosch2*. Bosch/Ford.
- C.K Murphy and P.A Bennet Ressearch Laboratories, G. M. (1968). V.I Improvers and engine performance. *SAE Technical Archive 980071*.
- Carden, P. T. (2014). Tribological Behavior of Low Viscosity Lubricants in the Piston to Bore Zone of a Modern Spark Ignition Engine. *SAE Technical Paper 2014-01-2859*.
- Covitch, T. L. (1998). How Polymer Architecture Affects Permanent Viscosity Loss of Multigrade Lubricants. *SAE International* .
- engineeringtoolbox. (2019, April). *engineeringtoolbox.com*. Retrieved from engineeringtoolbox.com: https://www.engineeringtoolbox.com/specific-heat-metals-d_152.html
- Ford_Motor_Company. (2019). 1.6 GDI Data set.
- GM Powertrain Division_Steve Shin, A. C. (2004). Offset Crankshaft Effects on SI Engine Combustion and Friction Performance. *SAE Techincal Paper*.
- Gulf Science and Technology Co-T.W.Selby, R. (n.d.). The Relationship Between Oil Viscosity and Engine Performance. *SAE Technical Archive 770372*.
- Kambe, T. (2007). *Elemental Fluid Mechanics*. Singapore: World Scientific Publishing Co.
- M. Knopf, C. E. (1998). Calculation of Unsteady Hydrodynamic Lubrication and Surface Contact at the Piston Ring / Cylinder-Liner Interface. *SAE Technical Paper Series* .
- Musashi Institute of Technology_Kei Nakayama, S. T. (2000). The Effect of Crankshaft Offset on Piston Friction Force in a Gasoline Engine. *SAE Technical Papers*.
- Petro-Canada Lubricants_H.E. Henderson, P. D. (1998). Categories, Higher Quality Base Oils for Tomorrow's Engine Oil Performance. *SAE TECHNICAL PAPER 982582*.
- Plumley, M. W. (2014). Optimizing Base Oil Viscosity Temperature Dependence For Power Cylinder Friction Reduction. *SAE Technical Paper 2014-01-1658*.
- Selby-Savant-Inc, T. (2010). High Shear Rate Rheology of Lower Viscosity Engine Oils Over a Temperature Range of 80° to 150°C Using the Tapered Bearing Simulator (TBS) Viscometer. *SAE internattional 2010-01-2288*.

Appendix

Experimental values for specific heat of general engine lubricant:

The data is quite messy with many introduced experimental errors, so the average was taken to three significant figures. Each of which is taken from the 6 repeated experiments

		Oil Mass (Kg)			1.013		
power (Kw)	power (Kw)	time (s)	time/step (s)	Q (J)	temp (C°)	ΔT	Specific Heat Capacity (KJ/Kg.K)
1097	1.097	0	0	0	25		0
1092.3333	1.0923333	0	0	0	30	5	0
1081.6667	1.0816667	26	26	28.123	35	5	5.55248437
1083	1.083	38.2	12.2	13.213	40	5	2.608608095
1083	1.083	48.6	10.4	11.263	45	5	2.223731491
1082.6667	1.0826667	58.8	10.2	11.043	50	5	2.18029615
1082.3333	1.0823333	70.2	11.4	12.339	55	5	2.436051333
1083.3333	1.0833333	80.4	10.2	11.05	60	5	2.181638697
1082	1.082	89	8.6	9.3052	65	5	1.83715696
1082.3333	1.0823333	97.4	8.4	9.0916	70	5	1.794985192
1082.3333	1.0823333	106.4	9	9.741	75	5	1.923198421
1082.6667	1.0826667	116	9.6	10.394	80	5	2.052043435
1082.6667	1.0826667	125.8	9.8	10.61	85	5	2.09479434
1082.6667	1.0826667	136.8	11	11.909	90	5	2.35129977
1082.3333	1.0823333	148.6	11.8	12.772	95	5	2.521526818
1081.6667	1.0816667	158.4	9.8	10.6	100	5	2.092859493
1081	1.081	170	11.6	12.54	105	5	2.475735439
1081.3333	1.0813333	181	11	11.895	110	5	2.34840408
1081.3333	1.0813333	195	14	15.139	115	5	2.98887792
1081.3333	1.0813333	206	11	11.895	120	5	2.34840408
1082.3333	1.0823333	214	8	8.6587	125	5	1.709509707
							2.231617857
							AVG

Drive cycle vehicle data:

LHV (KJ/Kg)	44000000									Efficiency	Ratio
Vehicle Mass (kg)	1650		Gravity (m/s^2)	9.81		Drag coefficient	0.38		1	0.94	3.73
Wheel Rolling Radius (m)	0.318		Rolling Resistance Coefficient	0.01		Air Density (kg/m^3)	1.2		2	0.94	2.05
Engine idle (rpm)	800		Frontal Area (m^2)	2.26		Wheel Inertia (kgm^2)	1.2		3	0.95	1.36
Engine Inertia (kgm^2)	0.2		Engine Displacement (cc)	999		Drive Shaft Inertia (kgm^2)	0.09		4	0.97	1.03
Engine idle (rpm)	83.7758041								5	0.97	0.82
Specific heat capacity OIL (KJ/Kg.K)	2.2								6	0.96	0.69
Sump mass of oil (Kg)	3.4891								7		
Specific heat capacity AL (KJ/Kg.K)	0.91								8		
Sump mass of Block (Kg)	60								9		
									Final Drive	0.94	3.82

Journal data:

Big-end radius	0.02	m
Big-end width	0.04	m
Crank main radius	0.0225	m
Crank main width	0.05	m
Oil film thickness	0.000001	m