Improved Fuel Efficiency by Lubricant Design : A Review

R.I. Taylor & R.C. Coy

Shell Research & Technology Centre, Thornton,
P.O. Box 1, Chester, CH1 3SH, UK

Abstract :

1. Introduction

The Kyoto Agreement on climate change has led to many countries agreeing to cut their CO_2 emissions relative to 1990 levels during the next decade. A major source of CO_2 emissions arises from the use of cars and trucks, and a significant amount of effort has been spent on manufacturing more fuel efficient cars. On the engineering side, manufacturers of passenger cars have introduced 4 valves per cylinder, roller follower valve train systems, lighter aluminium engines, smaller engine bearings, and gasoline direct injection engines. Similarly, there have been many advances in the design of heavy duty diesel engines over recent years including the introduction of high pressure fuel injection systems, the increasing use of 4 valves/cylinder, improved electronic management systems, and the introduction of two piece, articulated, pistons.

Significant savings may also be achieved by simply changing from a “standard” engine oil (e.g. an SAE-15W/40 grade) to a more fuel efficient engine oil. It has been estimated[1] that if all US car owners used an engine oil that gave them 0.5% fuel economy improvement, then the total cost savings would be $370 million per year. It is also worth noting that US manufacturers benefit significantly from more fuel efficient oils : a 1 mpg benefit in CAFÉ (Corporate Average Fuel Economy) is worth approximately $100 million since the manufacturer has more flexibility in the mix of vehicles which it can sell. Therefore, both Governments and OEMs (Original Equipment Manufacturers) are key drivers to more fuel efficient lubricants. Hence, the lubricant industry is actively pursuing fuel efficient, friction modified, engine oils with viscosity grades such as 0W/20, 5W/20, 5W/30, 0W/30, 0W/40.

In order to qualify a lubricant as a “fuel economy” oil, both the American Petroleum Institute (API) and ACEA have developed engine tests that measure the fuel consumption of candidate oils relative to reference oils, and the Effective Fuel Economy Increase (EFEI) of the candidate oils has to be better than that of the reference oil by some pre-defined limit in order to claim the oil is fuel efficient. In API tests, the current specification is ILSAC GF-2, which uses a Ford 4.6 litre V8 engine for its fuel economy test. A new specification, ILSAC GF-3 is currently under development, which will use the same engine, but aims to increase the role of mixed/boundary friction in the test cycle. In Europe, a fuel economy test has been developed using a Mercedes Benz M111 2.0 litre engine.

Whilst improved fuel consumption *per se* is seen as desirable world-wide, there are some subtle geographical nuances. Clearly, in the USA, the drive towards improved fuel consumption in passenger cars is driven by legislation via the CAFÉ limits, and manufacturers that do not meet the prescribed limits face swingeing financial penalties. It is also worth commenting that in the past, US cars have tended to use large engines (>4.0 litres) and so fuel consumption has generally been high. In Europe and Japan, where the use of small engines (<2.0 litres) is far more common, fuel consumption has been relatively lower, and so the focus has been on other factors too. In Japan, emissions control has been seen as a high priority. In Europe, the emphasis has been on durability, since very high speed driving occurs in certain European countries. Therefore, in Europe, until recently, passenger car lubricants have been required to have HTHSV (high temperature high shear viscosity, measured at 150°C and a shear rate of $10^6 s^{-1}$) greater than 3.5 mPa.s. It is also worth commenting that in Europe, tax rates on fuel are high, and so the drive towards better fuel consumption is partly driven by consumers. This helps to explain why diesel engined passenger cars are commonplace in Europe.

Whilst this review is mainly concerned with fuel economy in passenger cars, it is worth noting that hauliers clearly have a great interest in heavy duty trucks with improved fuel economy, since fuel is a major cost in a trucking operation. In the review some comments are made regarding the ability of engine and transmission lubricants to contribute towards improved truck fuel economy. However, since there are at present no heavy duty fuel economy engine tests in place, and as durability is still the major concern in heavy duty diesel engines, less emphasis is placed on heavy duty diesel engines in this paper.

In this review, a summary of the lubricant factors that influence fuel economy are elucidated[2-8], past, present and future fuel economy engine tests are described, and their appetites summarised, and the role of engine friction[9] and fuel economy engine test modelling[10] is discussed. In addition, other consequences of using fuel economy oils are discussed, and data is presented showing that with current fuel economy oil formulations, engine durability is maintained.

2. Lubricant Factors Affecting Fuel Consumption

It is generally accepted that both the piston assembly and bearings are predominantly in the hydrodynamic lubrication regime, whereas the valve train is in the mixed/boundary lubrication regime[2,3]. Therefore the simplest approach to developing a fuel efficient lubricant is to reduce the viscosity (to give benefits in pistons and bearings) whilst at the same time adding an effective friction modifier (which gives benefits in the valve train). However, it is still necessary to pass all other relevant engine tests, it is also desirable to retain a low volatility, and it is essential that engine durability is maintained. More detailed formulation factors also affect the fuel savings achieved (e.g. is the lower viscosity achieved through using a low base oil viscosity and a lot of Viscosity Index Improver, or a higher base oil viscosity and less VII). Questions that need asking in such developments are : What friction modifier should be used ? Will the friction modifier interfere with the antiwear additive ? What base oil should be used (synthetic or mineral oil based) ? What Viscosity Index Improver should be used ?

In addition to these purely lubricant issues, other factors need to be considered. Engine design will have a big influence on the effectiveness of the lubricant in reducing fuel consumption. For example, an engine with 4 valves/cylinder, with sliding contact direct acting valve trains, will have a high proportion of valve train friction, and so a lubricant containing a friction modifier will be effective at reducing fuel consumption, but the same lubricant will not be so effective at reducing fuel consumption in an engine that uses a roller follower valve train system.

Also, the driving cycle is of great importance. For drivers that make a large number of short trips, the engine is never fully warmed up, and minimising the viscosity at the low temperature end is important, whereas drivers that mainly use motorways, when the engine is fully warmed up, will require oils that are optimised at the high temperature end. Higher fuel savings are more likely to be achieved for the driver making numerous short trips[11,12]. The importance of cold starts is one of the reasons for the recent proliferation of 0W/x and 5W/x oils in Europe.

Figure 1 summarises typical European driving habits, and emphasises the importance of short trip driving patterns.

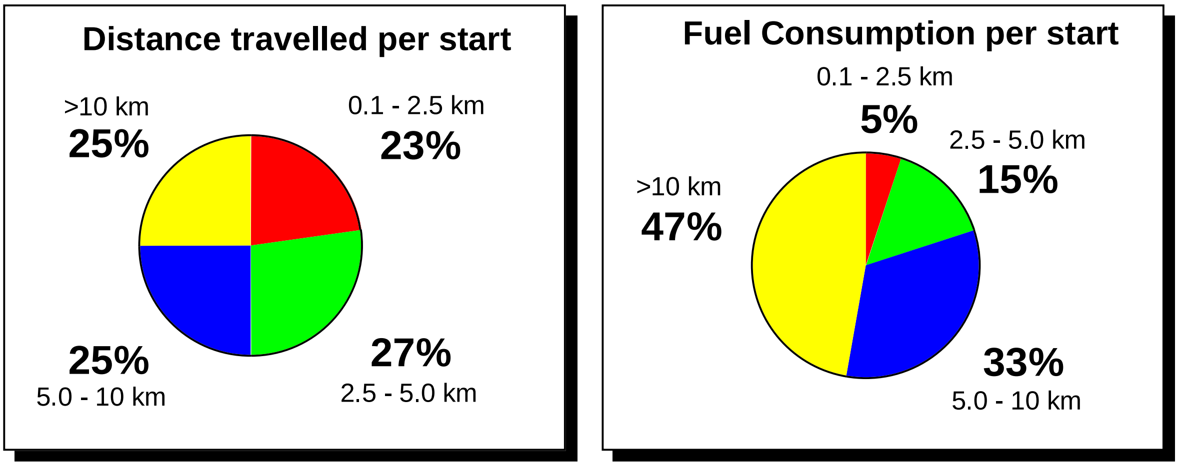

Figure 1 : Average distance travelled per engine start, and average fuel consumption per engine start, for a typical European car user. Note that 53% of total fuel consumption is for journeys less than 10 km in length

3. Modelling of Engine Friction & Fuel Economy Engine Tests

Modelling activity can be broadly split into two types.

- A number of groups, such as those at GM[13,14], Ford[15], Shell[9,16], The University of Leeds[17], Nissan[18], Toyota[19], and Southwest Research Institute[20] have lubrication models for the three main engine components : the piston assembly, the valve train system, and the bearings. These models enable estimates of friction (and wear) to be made for different engine speeds/loads/temperatures. The models have the advantage that predictions can be made for different engines under a wide variety of operating conditions relatively quickly. The disadvantage of this approach is that a relatively large amount of data is required to model each engine component, and some data, such as combustion chamber pressures, component temperatures, can be difficult to obtain.
- An alternative approach, adopted by Ford[21], BP[22], Ethyl[23] and Paramins/Imperial College[10], is to measure viscometric parameters of lubricants that are representative of hydrodynamic, mixed and boundary lubrication, and then empirically fit Effective Fuel Economy Increase (EFEI,%) to fuel economy engine test results. The advantage of this technique is that it is simple and quick. A disadvantage is that engine test results must be available before predictions can be made, it is necessary to ensure that the laboratory viscometric measurements are representative of the engine operating conditions.

3.1 Detailed Engine Friction Modelling

Some results from the first approach are summarised in Table 1, from the GM FLARE program[13], and Table 2 summarises some results from Shell[9] obtained for the Mercedes Benz M111 2.0 litre gasoline engine. (Note that the results have been converted to FMEP (Frictional Mean Effective Pressure) to make comparisons easier).

	Power Loss (kW) (FMEP figure in kPa)	
	2000 revs/min	5000 revs/min
Bearings	0.90	5.00
Piston Skirt	0.95	5.35
Piston Rings	1.17	2.96
Valve Train	1.55	2.60
Total	4.57 (54.8)	15.91 (76.4)

Table 1 : Results from GM FLARE software for a 5.0 litre gasoline engine for an SAE-10W/30 engine oil

	Power Loss (kW) (FMEP in kPa)		
	SAE-10W/30	SAE-15W/40	SAE-20W/50
Bearings	0.55	0.59	0.63
Piston Assembly	0.52	0.64	0.80
Valve Train	0.38	0.29	0.14
Total	1.45 (34.8)	1.52 (36.5)	1.57 (37.7)

Table 2 : Results from Shell engine friction model for Mercedes Benz M111 2.0 litre gasoline engine at 2500 revs/min

Table 3 shows some results from the Leeds University model[17], as applied to a 1.8 litre Ford Zetec engine.

	Power Loss (kW) (FMEP figure in kPa)	
	2000 revs/min	5000 revs/min
Bearings	0.275	1.203
Piston Assembly	1.480	3.660
Valve Train	0.446	1.031
Total	2.201 (73.37)	5.894 (78.59)

Table 3 : Results for Leeds University total engine friction model[17] as applied to a Ford Zetec 1.8 litre gasoline engine (lubricant not specified)

Figure 2 shows the relative distribution of power losses amongst the three main engine components for the three different models (and engines) above at speeds of 2000 revs/min or 2500 revs/min. (Note that the actual power loss distribution, and the overall power loss, are very sensitive to the temperatures assumed in the engine components, since lubricant viscosity varies strongly with temperature.)

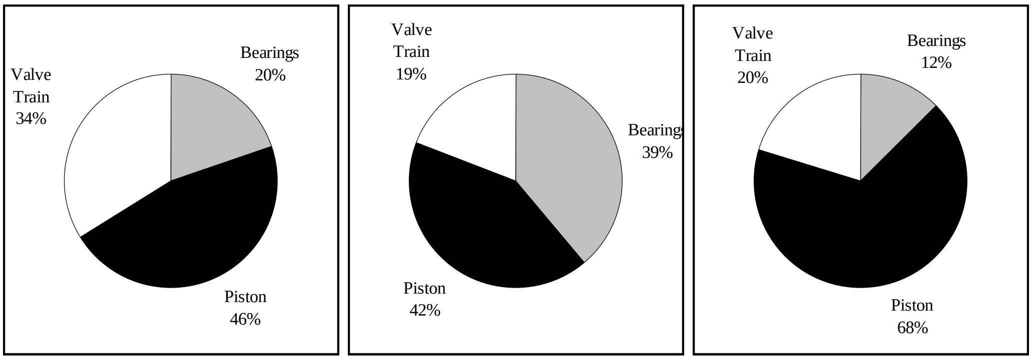

Figure 2 : Relative losses in bearings, valve train and piston. Left graph shows results from GM FLARE model for a 5.0 litre gasoline engine @ 2000 revs/min. Centre graph shows results from Shell model for Mercedes Benz M111 2.0 litre gasoline engine @ 2500 revs/min (assuming an SAE-15W/40 lubricant). Right graph shows results of Leeds University model for a Ford 1.8 litre Zetec engine @ 2000 revs/min

A couple of comments are worth making about Figure 2, and the results contained in Tables 1-3. The relative proportions of losses in the three main components are in relatively good agreement for the GM and Shell models. The Leeds model seems to have much higher piston assembly losses compared to the other models. From the results contained in the tables above, the Shell model seems to underestimate the FMEP (Frictional Mean Effective Pressure) compared to the GM and Leeds University model. However, since the temperatures in the engine components are not specified in the GM and Leeds model, direct comparisons are not straightforward.

Figure 3 shows the friction breakdown according to the Nissan model[18] for three hypothetical 2.0 litre gasoline engines. Engine D is an in-line 4 cylinder engine with bore x stroke = 82.5 mm x 93.5 mm, with a double overhead cam (DOHC), with 16 valves. Engine E is an in-line 6 cylinder engine with bore x stroke = 70.0 mm x 86.6 mm, with a DOHC, with 24 valves. Engine F is a V-type 6 cylinder engine with bore x stroke = 75.0 mm x 75.5 mm, with a DOHC, with 24 valves.

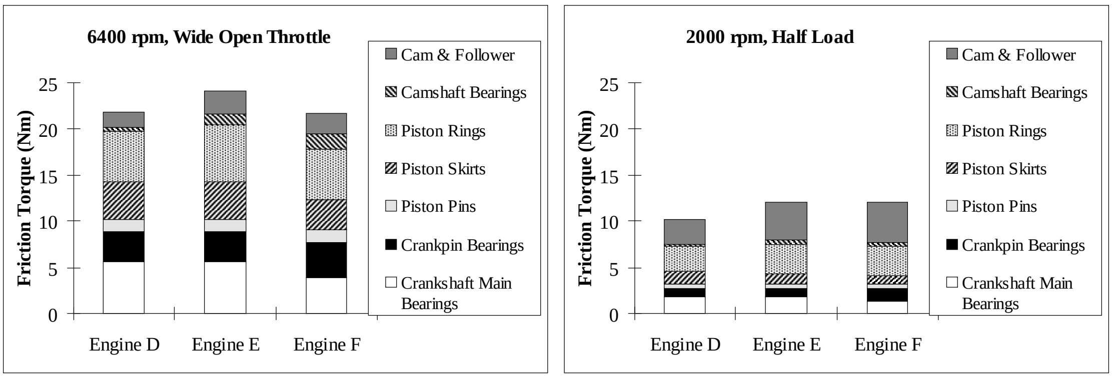

Figure 3 : Nissan predictions for 3 hypothetical 2.0 litre gasoline engines[18]

The results of the Nissan predictions are fairly easy to understand. Engine D has less valves than engines E or F (16 compared to 24) and so the valve train frictional torque is less for this engine (as is the camshaft bearing friction). Engine F has lower crankshaft main bearing friction since it is a V-type 6 cylinder engine, and so there are less main bearings than for the in-line engines. However, Hamai[18] does not explicitly state what oil viscosity was assumed, and does not use the model to explore the lubricant sensitivity of the engines. His conclusion was that, of the three hypothetical engines considered, the in-line 4 cylinder engine offered the best prospects for achieving low friction, under both low and high speed engine operating conditions.

The work by Southwest Research Institute (SwRI)[20], based on a generic 2.0 litre gasoline engine with a single overhead cam (SOHC), concluded that "the model predicts that the friction of the piston rings is the highest single component in engine friction, except at high engine speeds, where the predicted windage is greater. Next after the piston rings was the piston body friction. The remaining components were relatively small, and in order of importance were the accessories, the cam bearing friction, cam/tappet friction, the main bearing, the crank pin, and oscillatory friction in the valve train, in that order." (Windage refers to losses due to air motion in the crankcase, and this loss is significantly affected by the proportion of oil mist in the air.) The work done by SwRI was based on a generic 2.0 litre four cylinder gasoline engine with a SOHC.

Figure 4 shows the breakdown of mechanical losses for a motored car engine according to Lang[24] of Daimler Benz, and Figure 5 shows a similar breakdown for a 1.3 litre gasoline engine at 5000 revs/min and full load, according to Hoshi[25].

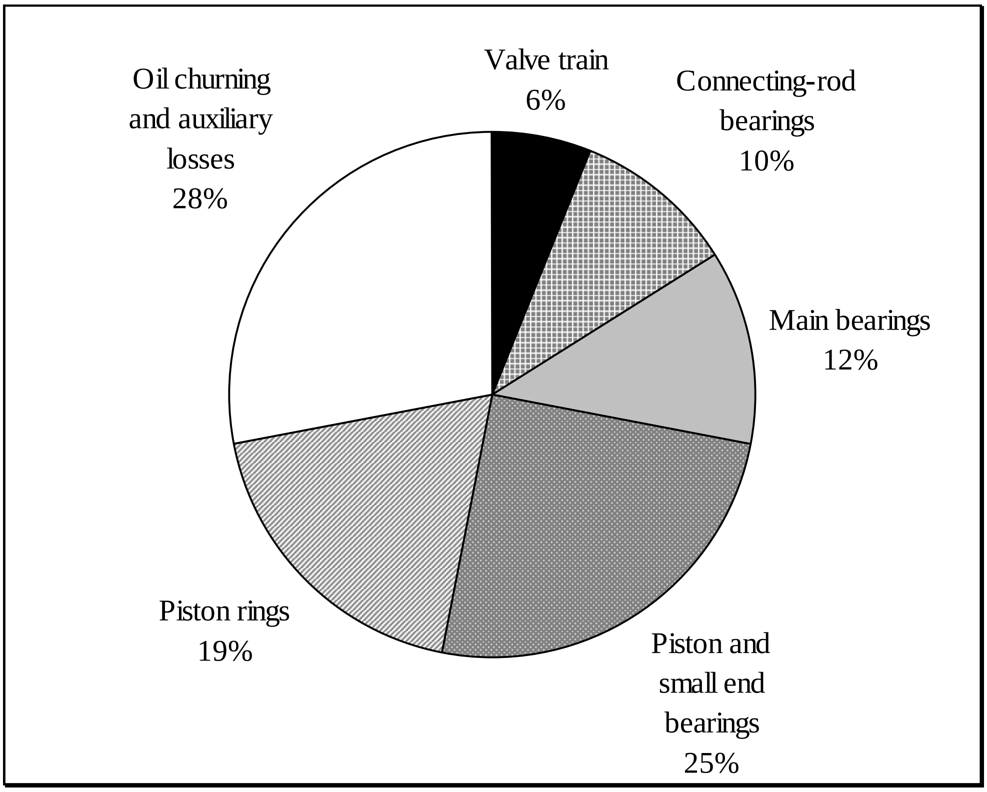

Figure 4 : Breakdown of mechanical losses for a motored car engine[24]

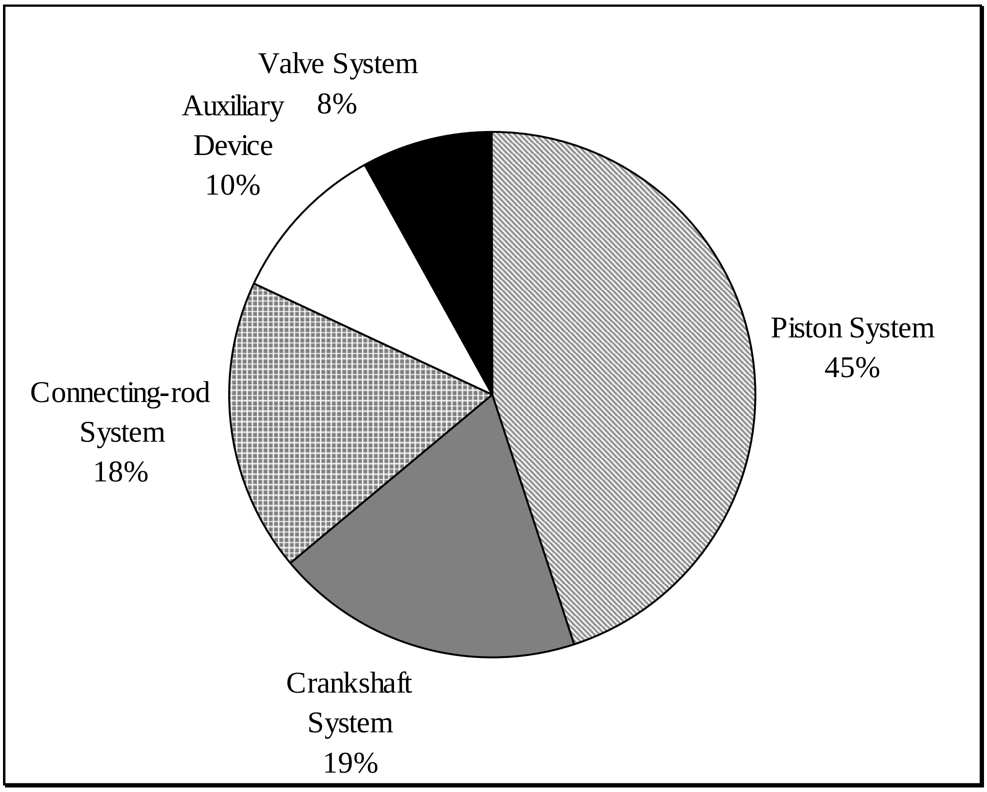

Figure 5 : Total Power Losses for a 1.3 litre Engine at 5000 revs/min and Full Load[25]

Of the published engine friction models, the Shell model has concentrated on including realistic lubricant viscometry (i.e. variations of viscosity with temperature, shear rate and pressure.) Figure 6 shows the complicated way in which the viscosity of a lubricant varies with both temperature and shear rate for an SAE-15W/40 lubricant.

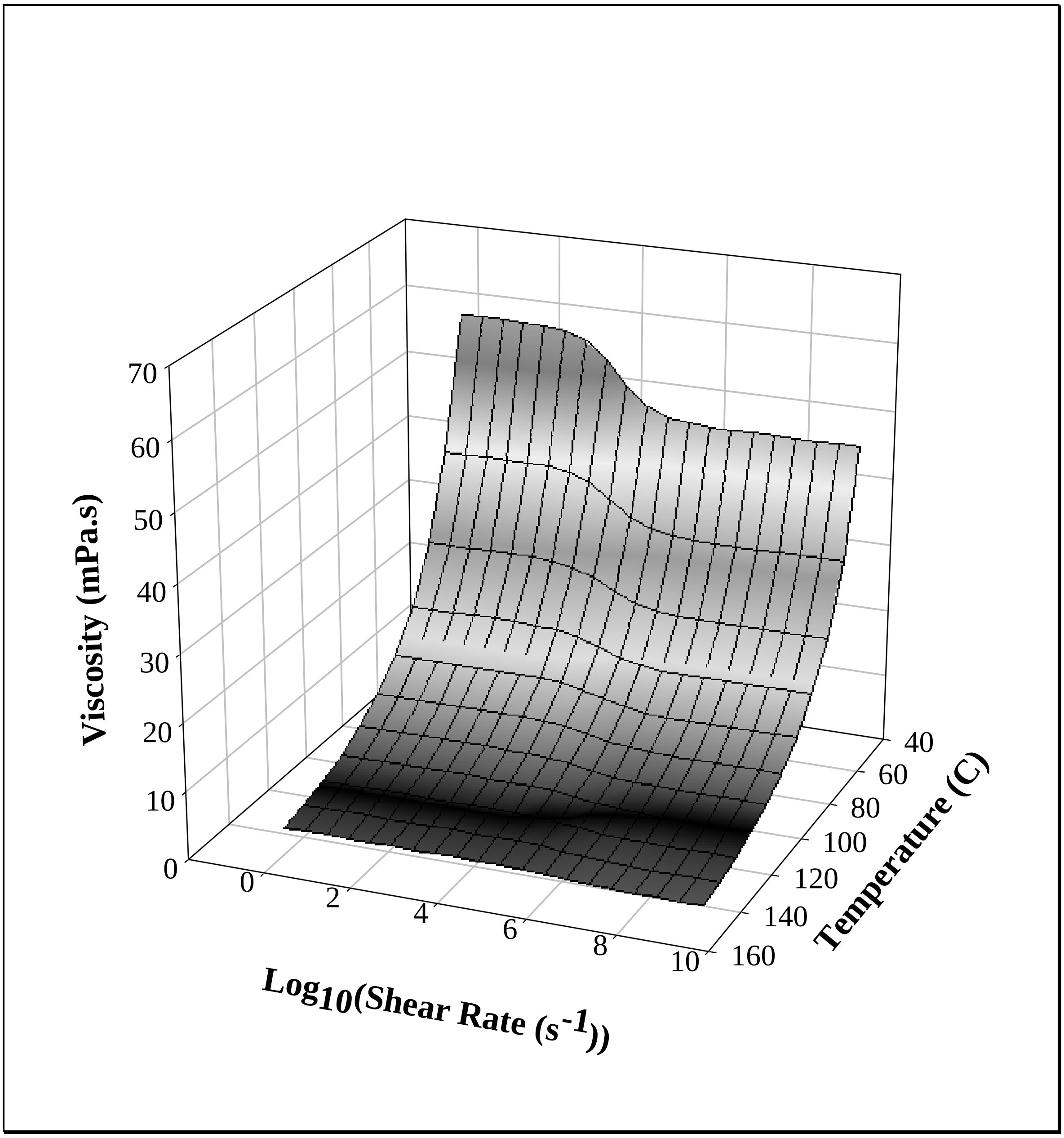

Figure 6 : Variation of viscosity with temperature and shear rate for an SAE-15W/40 oil

One way in which the Shell friction models have been used is to study the sensitivity of the Sequence VI-A fuel economy engine test (and more recently, the proposed Sequence VI-B fuel economy engine test) to lubricant viscometry. Figure 7 shows the variation of viscosity with shear rate for two early fuel economy oil formulations, Oil A, Oil B (both of which have a HTHS viscosity of 2.9 mPa.s) and the BC-2 reference oil (used in the VI-A test). The shear flow curves are shown for temperatures of 100°C and 150°C.

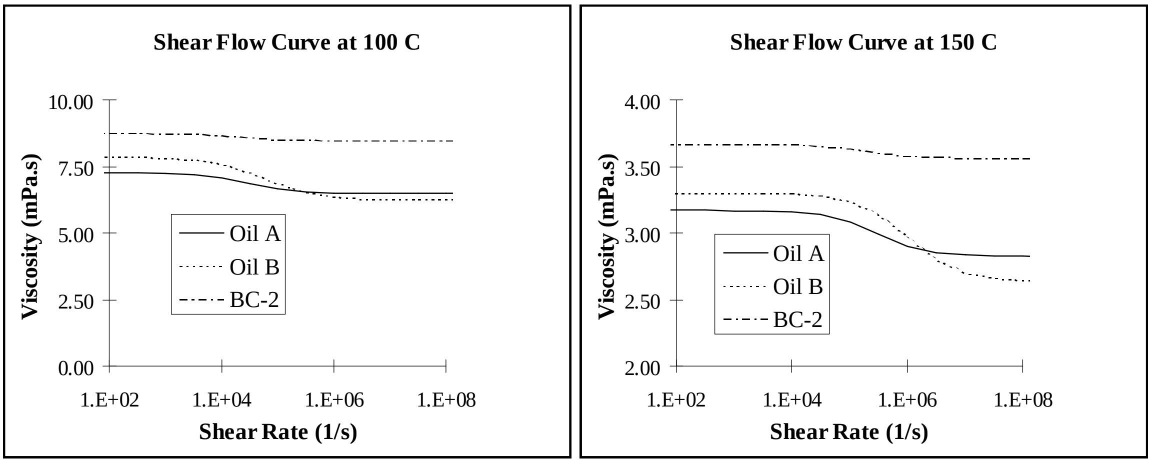

Figure 7 : Shear flow curves for three different lubricants at 100°C and 150°C

The engine used in the Sequence VI-A fuel economy test has a valve train that uses roller followers. Therefore, the valve train contribution to total engine friction is very small (as demonstrated by the small friction modifier response of the engine, as will be discussed in more detail later). Hence, when modelling friction in this engine one only needs to consider the bearings and the piston assembly. Figure 8 shows the results of such a simulation, for all six stages of the Sequence VI-A engine test, for oil B. Figure 9 shows the total power loss for each stage for each of the three oils, oil A, oil B and BC-2.

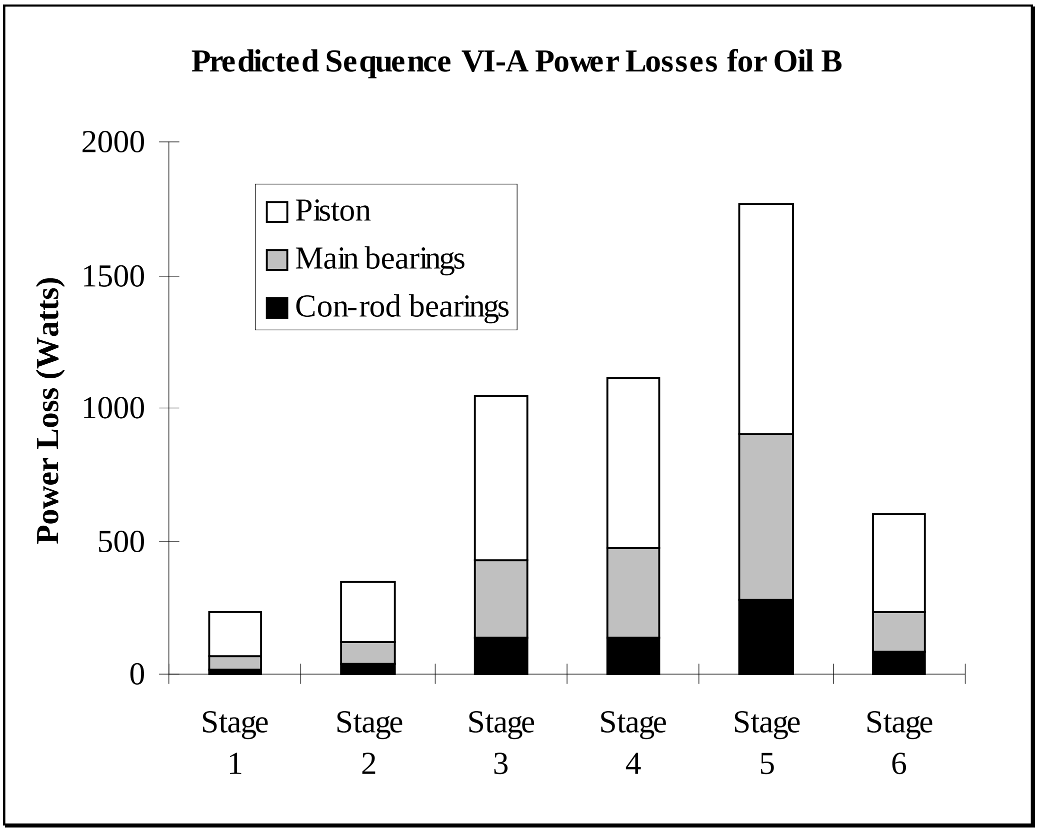

Figure 8 : Predicted Sequence VI-A friction losses for oil B, and the relative contribution of piston assembly, main bearing and con-rod bearing friction losses

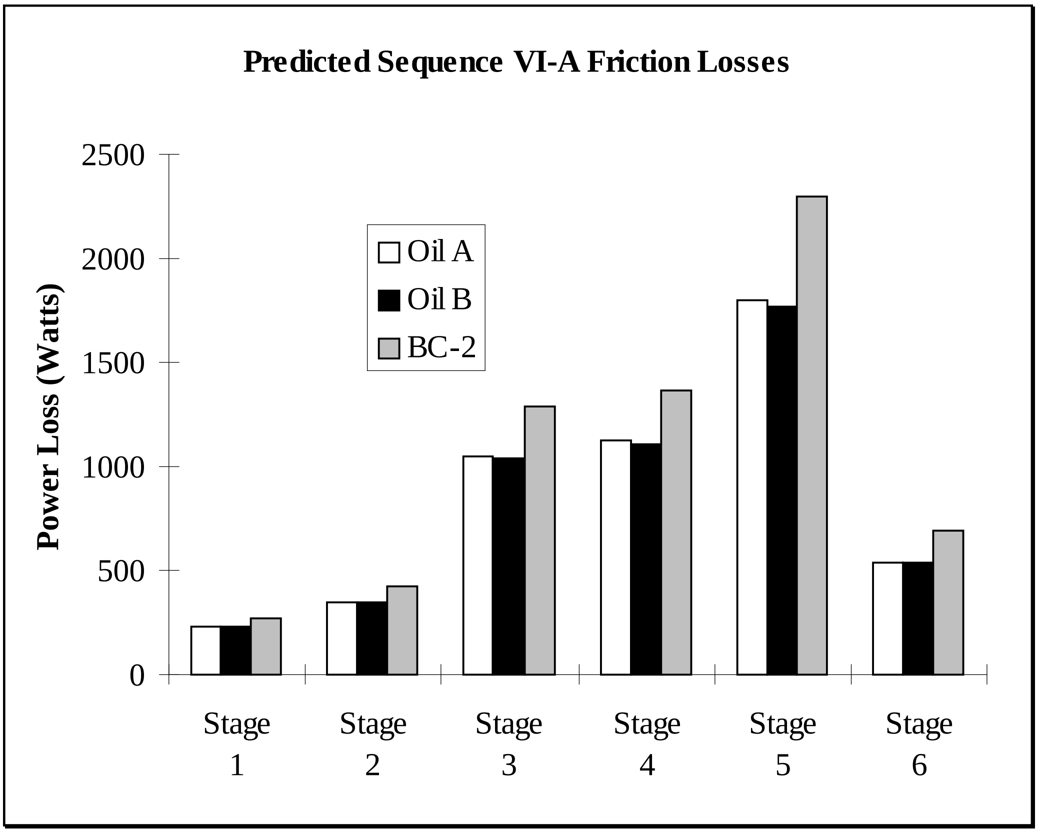

Figure 9 : Predicted Sequence VI-A friction losses for oil A, oil B and BC-2 lubricants

The modelling work described above showed that both oil A and oil B gave a sizeable reduction in friction loss compared to the BC-2 reference oil. Since oil A and oil B had nominally the same HTHS viscosity (2.9 mPa.s), the results demonstrated that the oil with the lower base oil viscosity (in this case oil B) should give lower friction.

Engine test results with these two oils are shown in Figure 10. Note that the abbreviation “EFEI” is the Effective Fuel Economy Increase relative to the reference oil, after suitable weighting factors are applied to each of the six stages of the engine test.

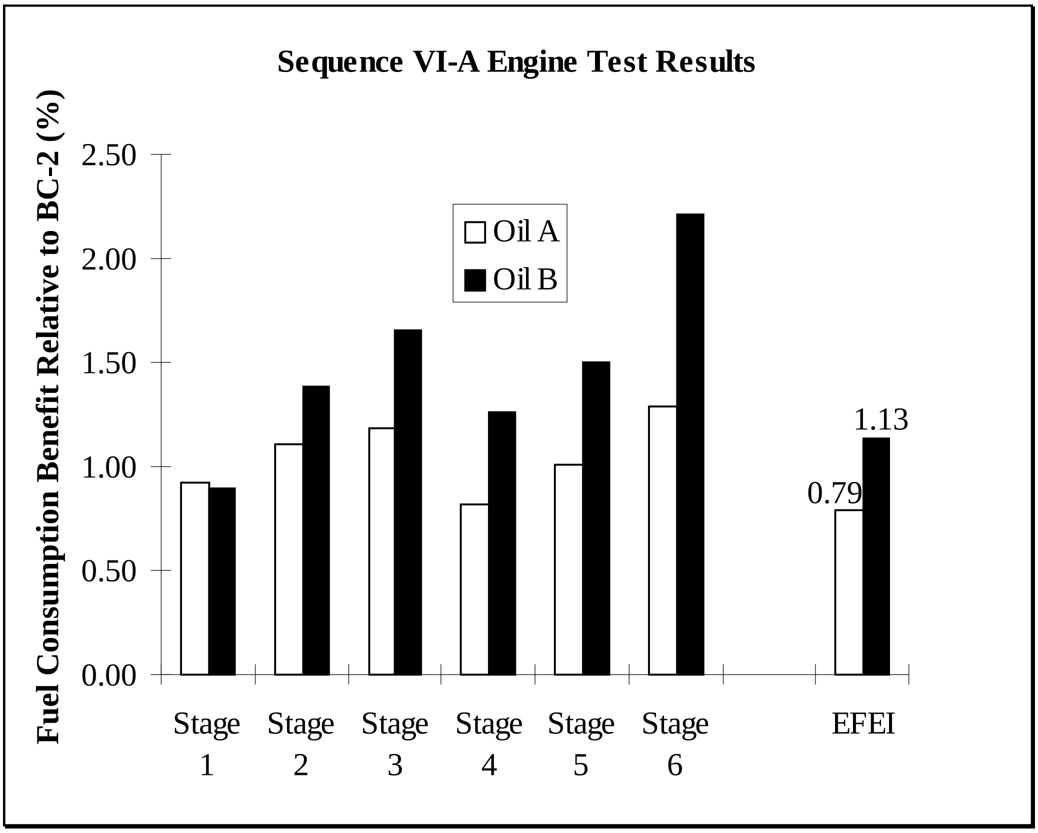

Figure 10 : Sequence VI-A engine test results

The modelling work also gives an estimate of the relative proportion of boundary friction in the Sequence VI-A engine test. Table 4 shows the percentage boundary loss in each of the 6 stages, together with the absolute power loss, for oil B. In Table 4, an additional stage has been considered, which is essentially the extra stage introduced for the Sequence VI-B test (oil temperature of 125°C, speed of 1500 revs/min, and high load (98 Nm)).

Stage	Total Loss (W)	Boundary Contribution (%)
1	233.7	6.3
2	347.0	2.0
3	1043.0	0.5
4	1110.3	1.7
5	1765.9	0.6
6	537.6	0.6
“New”	601.9	6.4

Table 4 : Summary of total losses for oil B together with percentage boundary friction

In conclusion, modelling of total engine friction from first principles can be used to analyse standard fuel economy engine tests, and if lubricant rheological parameters are adequately accounted for, insights for good formulation strategies for meeting test limits can be obtained. However, the models can also be used for other engines, other operating conditions, and in some cases can also be used for estimating wear rates.

3.2 Empirical Fuel Economy Engine Test Modelling

An alternative approach to full engine friction modelling has been developed by a number of authors[10,21,22,23,26]. The aim of this simpler approach relies on having a good set of engine test results, and effectively involves finding a correlation function between the fuel economy benefit and representative rheological properties (e.g. a viscosity value that is representative of hydrodynamic lubrication, a friction coefficient that is representative of boundary friction, and a parameter such as the pressure-viscosity coefficient that is representative of EHD/mixed lubrication). Moore[22] reports that the general correlation function for fuel economy increase is of the form :

$$FEI = a - b \cdot \eta - c \cdot \mu - d \cdot \alpha \qquad \ldots(1)$$

where a, b, c and d are constants, η is a high shear viscosity, μ is a boundary friction coefficient and α is the pressure-viscosity coefficient. The boundary friction coefficient, μ, is generally measured in a reciprocating friction rig (Moore[22] uses the Plint TE-77 High Frequency Friction Machine). The pressure-viscosity coefficient, α, is not always used in correlation functions, since it is not that straightforward to measure.

Moore[22] has reported correlation functions for the Sequence VI and VI-A engine tests. For the Sequence VI engine test, his correlation function is :

$$EFEI(\%) = 8.647 - 1.252 \cdot \eta_{150} - 15.62 \cdot \mu_{100} \qquad \ldots(2)$$

For the Sequence VI-A engine test, his reported correlation function is :

$$EFEI(\%) = 6.238 - 1.697 \cdot \eta_{150} - 4.051 \cdot \mu_{100} \qquad \ldots(3)$$

His conclusion was that “the relative importance of boundary friction in the Sequence VI-A test is much less than that in the Sequence VI”. In engine design terms, this is straightforward to understand, since the Sequence VI-A engine employs roller follower valve trains whereas the Sequence VI engine used sliding followers.

Gangopadhyay et al[21] carried out a similar analysis but used high shear viscosities at temperatures appropriate to the engine test, rather than at the single value of 150°C. This approach is, in principle, capable of distinguishing between oils that have the same HTHS viscosity (at 150°C) but different base oil viscosities. The equations proposed by Moore above would not distinguish between such lubricants.

Bovington and Spikes[10] use a similar model but split up the total friction into three contributions, namely hydrodynamic, traction and boundary. The “traction” part seems to be related to friction in EHD (elasto-hydrodynamic) contacts in the engine. Their conclusion for the Sequence VI and VI-A engine tests are summarised in Figure 11.

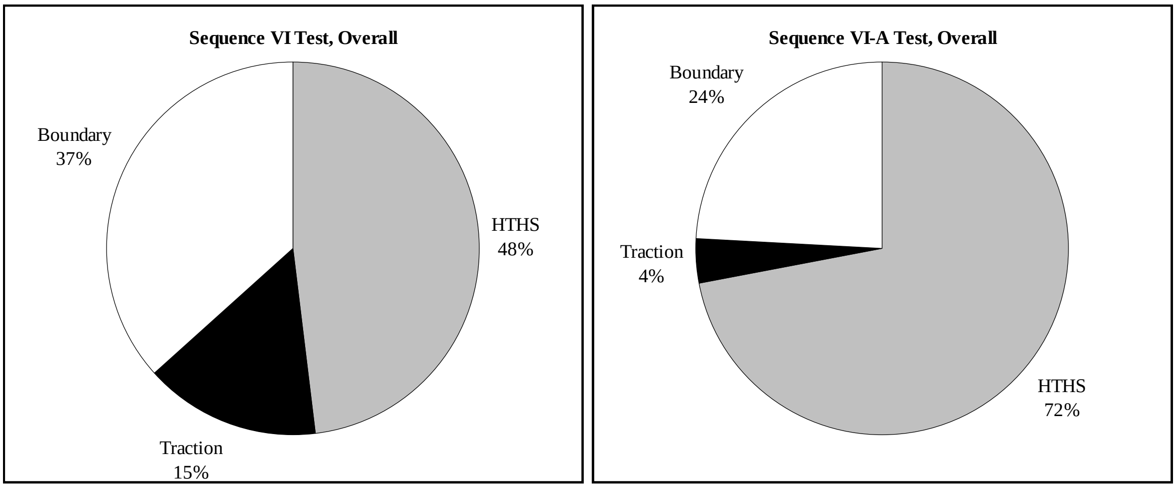

Figure 11 : Summary of Bovington's results[10] for Sequence VI and VI-A engine tests

The empirical approach outlined here can, of course, be applied to field trial results as well as to industry standard engine tests.

General Motors[26] have reported that the following correlation functions are reasonably good across a wide range of their engines :

$$\%FE = 2.752 - 0.267 \cdot (KIN100) \quad \ldots(4)$$

where %FE is the percentage increase in fuel economy relative to a BC reference oil, and KIN100 is the kinematic viscosity measured at 100°C. The following equation was also reported to work :

$$\%FE = 3.823 - 1.214 \cdot (HTHS) \quad \ldots(5)$$

where HTHS is the HTHS viscosity measured at 150°C.

Devlin[23] has also reported on the fuel economy performance of GM vehicles. Figure 12 summarises his conclusions. Basically, he finds that the combined highway and city fuel economy (COMFE) for GM vehicles shows higher boundary friction than that found in the Sequence VI-A engine test.

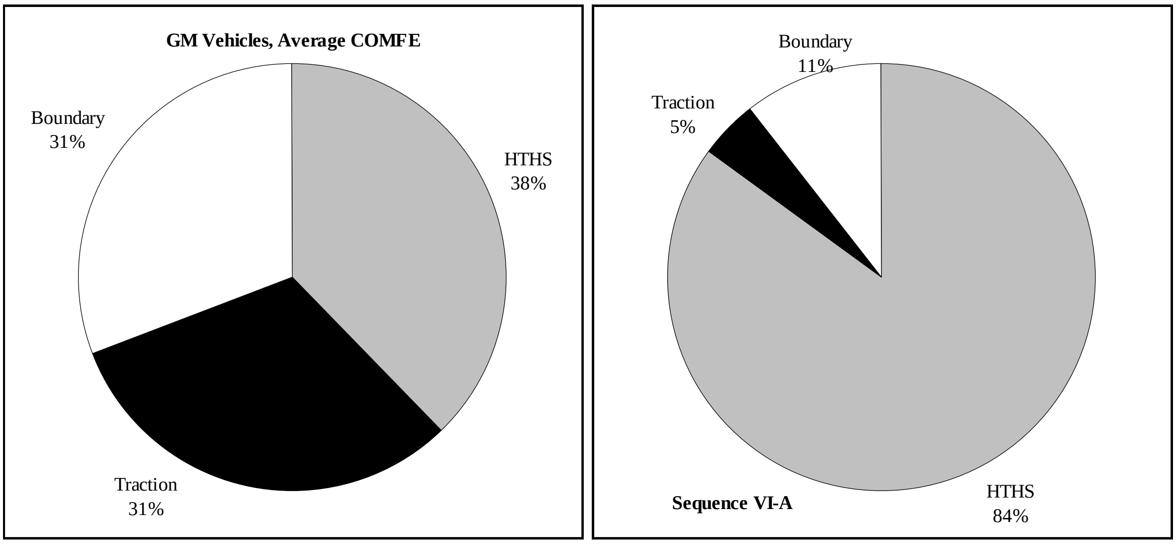

Figure 12 : Relative fuel consumption factors in GM vehicles & Sequence VI-A engine test

As far as the authors are aware, no correlation functions have been reported for either the Mercedes Benz M111 or the proposed Sequence VI-B engine tests.

In summary, the empirical approach has the advantage of simplicity. However, it relies on there being a significant amount of engine test data already available, and it is necessary to choose viscometric and boundary properties that are representative of the engine test. New correlation functions need to be developed if the engine test conditions are changed.

4. Engine Test Results

In this Section, a selection of engine test results are summarised for the Sequence VI-A and Mercedes-Benz M111 fuel economy tests.

Figure 13 shows typical results obtained from the Sequence VI-A engine test.

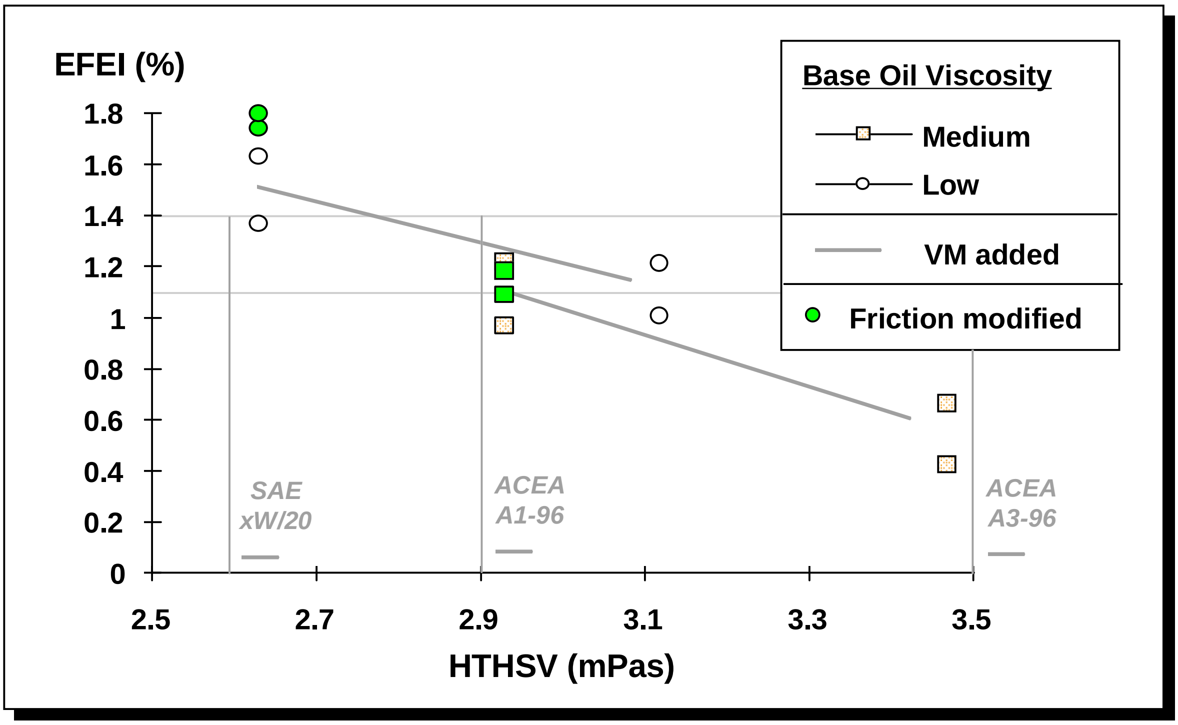

Figure 13 : Summary of Sequence VI-A engine test results

Conclusions from Figure 13 are that, in the Sequence VI-A engine test, the Effective Fuel Economy Increase (%) is almost linear with decreasing HTHS viscosity, and that friction modifier effects are small, with perhaps a 0.2% benefit at 2.6 mPa.s, but no discernible benefit seen at 2.9 mPa.s. Figure 13 also suggests that for a given HTHS viscosity, a lower base oil viscosity would give a higher friction benefit. It should be noted that the reference oil used in the Sequence VI-A engine test has a HTHS viscosity of around 3.5 mPa.s. A Sequence VI-B test has been proposed for ILSAC GF-3, which uses the same engine as for the Sequence VI-A test, but has had some of the stages altered in order to try to get a larger friction modifier response[27]. In addition, two fuel economy determinations will be carried out, one of which is after 16 hours (essentially the fuel consumption benefit of the fresh candidate oil), and the other is carried out after a further 80 hours of aging (this is the fuel consumption benefit of the aged oil). These modifications were made firstly to increase the friction modifier effect in the engine, and secondly to ensure that such friction modifier benefits were retained during the lifetime of the oil in the engine.

The Mercedes Benz M111 engine test appetite is less well known than that of the Sequence VI-A engine. However, Shell has run a matrix of oils in the M111 test. The oils had different HTHS viscosities, but contained no friction modifier. Figure 14 shows the results obtained, together with results obtained for an that does contain a oil friction modifier. The reference oil, RL-191, whose viscosity is 3.9 mPa.s is also included on the graph.

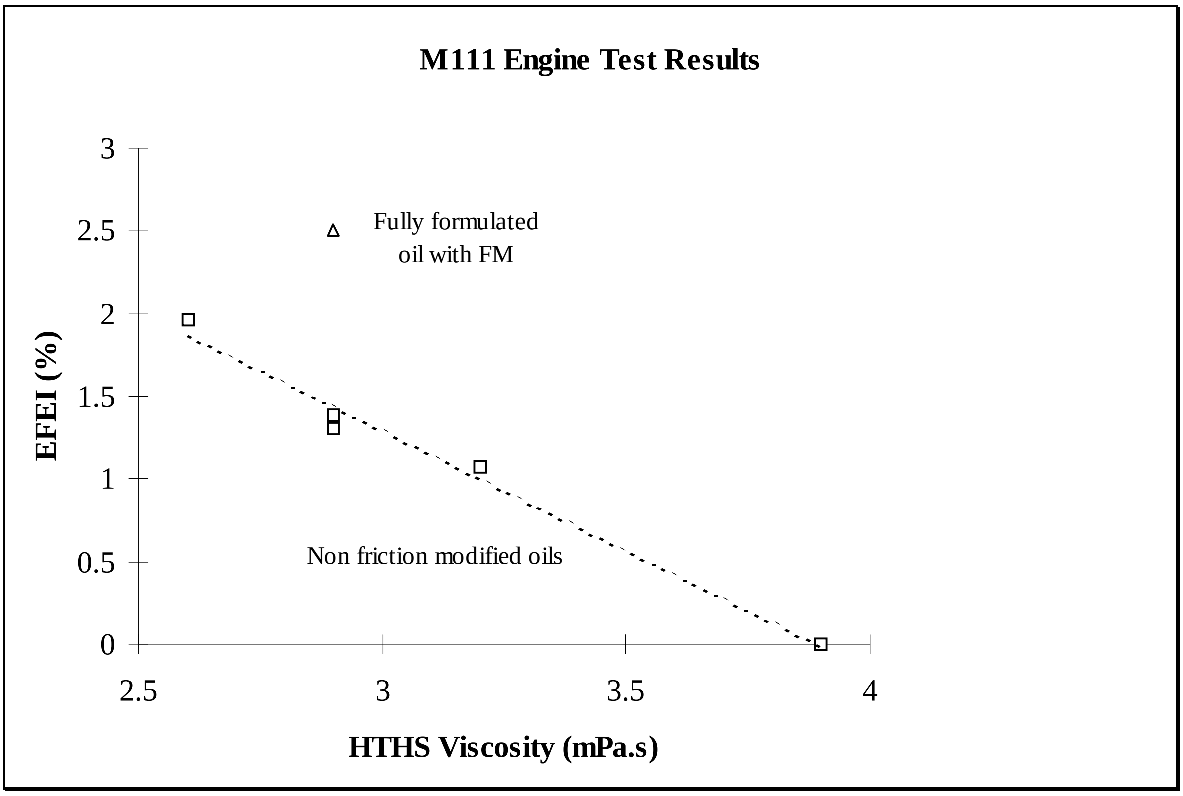

Figure 14 : Preliminary M111 engine test results

The results contained in Figure 14 are interesting since they show a clear, linear trend with HTHS viscosity, but in addition show a very significant effect due to the presence of friction modifiers. At a HTHS of 2.9 mPa.s, roughly 1.5% EFEI can be achieved due to the lower oil viscosity compared to that of the reference oil, but another 1.0% can be achieved simply by adding an effective friction modifier. As the ACEA A1/B1 limit is 2.5% EFEI, and the minimum allowable HTHS viscosity is 2.9 mPa.s, this limit can be achieved. Higher EFEI values can be obtained by using a lower base oil viscosity at a given HTHS viscosity, in a similar way to that seen in the Sequence VI-A engine test.

Apart from the standard fuel economy engine tests, there are many other demonstrations of a fuel economy benefit being obtained by using lower viscosity oils. Examples are :

(1) An SAE-5W/20 lubricant (with a HTHS viscosity of 2.9 mPa.s) gave the following fuel consumption savings in a field trial in Germany. Figure 15 shows the results.

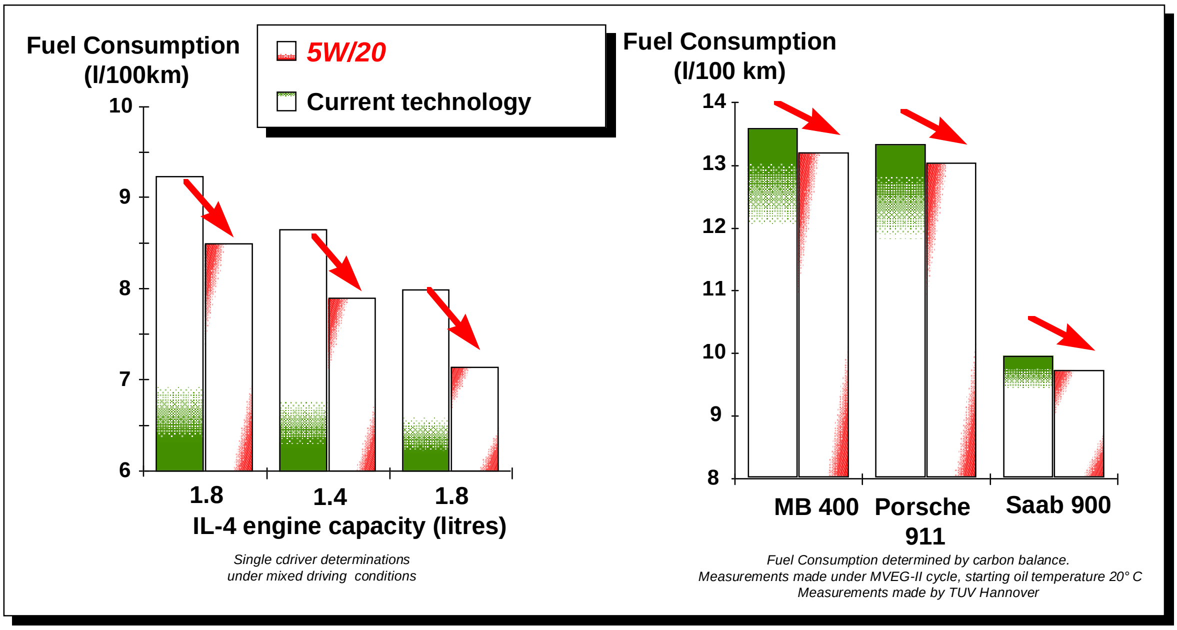

Figure 15 : Demonstration of fuel economy benefits that can be obtained in current production engines using an SAE-5W/20 lubricant

(2) An independent test laboratory showed that the same lubricant gave a power advantage over other lubricants tested (this is another indication that the engine friction is lower). Figure 16 shows the results.

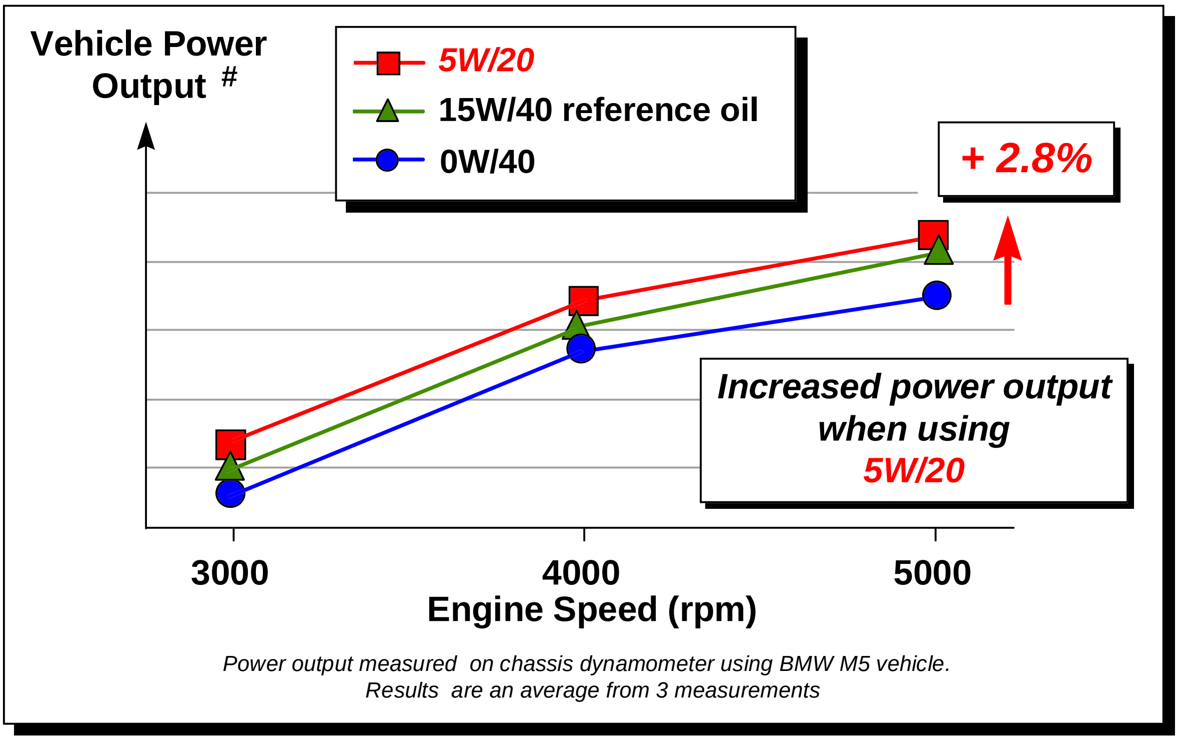

Figure 16 : Demonstration of power output increase that can be obtained when using an SAE-5W/20 lubricant. Measurement carried out by an independent laboratory

(3) Using a 2.5 litre V6 engine (with roller followers), the following fuel economy benefits were seen for SAE-5W/30 and SAE-10W/40 oils compared to SAE-20W/50 grades. Figure 17 shows the results.

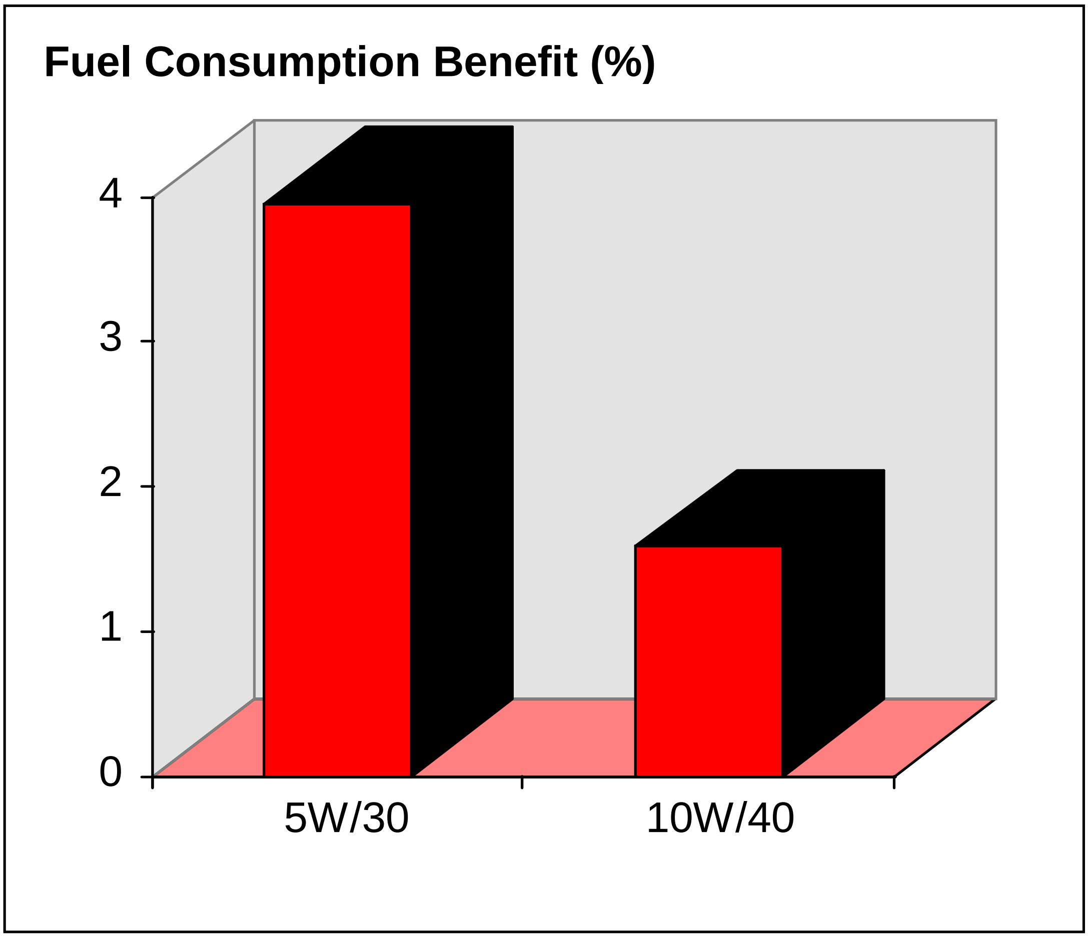

Figure 17 : Measured fuel consumption benefit compared to an SAE-20W/50 lubricant. Measurements carried out on a Ford 2.5 litre V6 engine, operating under cyclic conditions

Currently, there are no fuel economy engine tests available for heavy duty diesel engines. However, there is some data to show that benefits can be obtained by moving to lower viscosity lubricants. Figure 18 shows data obtained from a chassis dynamometer test on 6 tonne trucks on the ECE-15 cycle showing the potential advantages to be obtained using an SAE-10W/30 lubricant compared to an SAE-15W/40 lubricant[28]. In addition, field trial data has been obtained by Shell that suggests fuel consumption savings of up to 4.8% can be achieved in Volvo FH12 engines when using an SAE-5W/30 lubricant compared to an SAE-15W/40 lubricant. There are, however, some subtle differences between heavy duty diesel engines and passenger car gasoline engines.

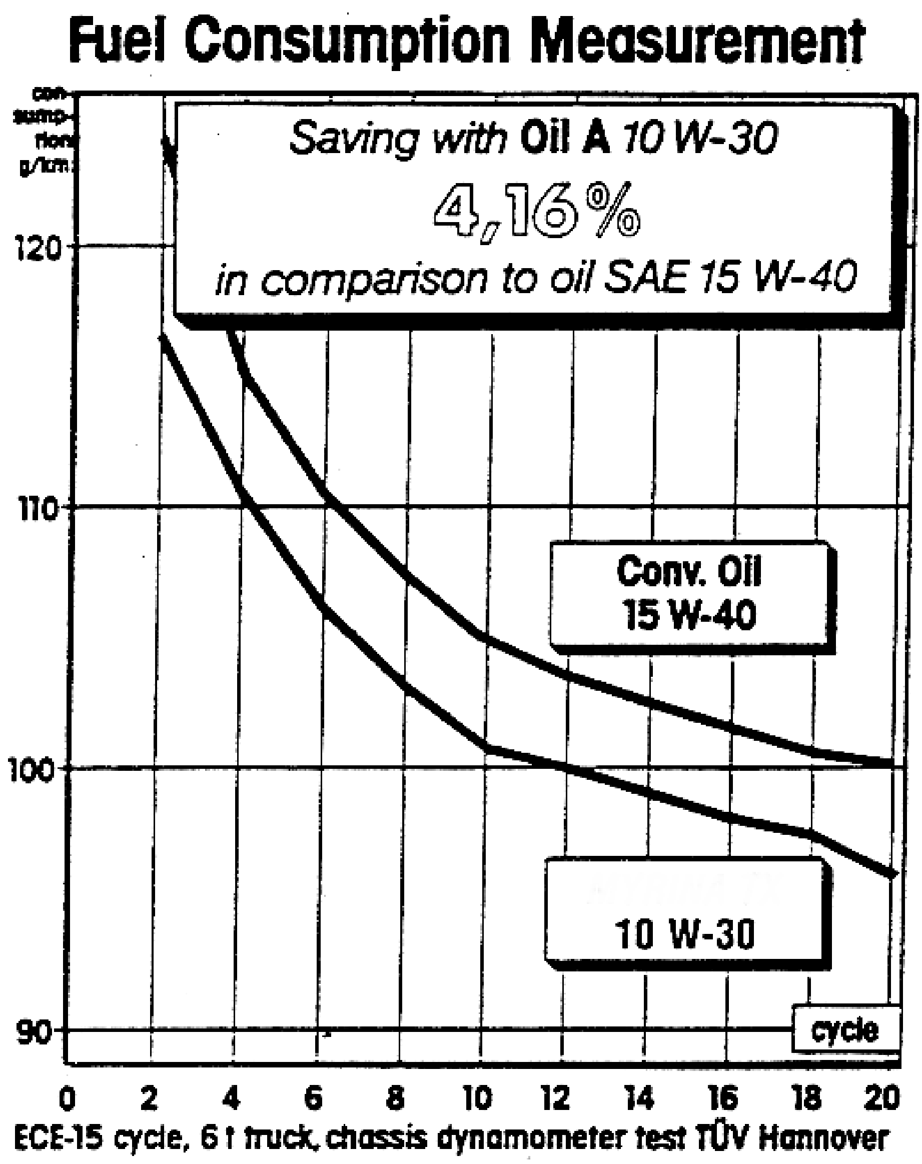

Figure 18 : Fuel savings that can be achieved in 6 tonne trucks over the ECE-15 cycle with an SAE-10W/30 lubricant compared to an SAE-15W/40 lubricant

- Firstly, duty cycles are quite different, with heavy duty diesel engines often operating under high speed and high load for long periods of time. Therefore, the impact of cold-starts on fuel consumption is typically less than for passenger cars. Also, the lubricant related losses are a smaller fraction of the total losses in a truck, due to the high load operation, so fuel consumption savings are often less than those found in passenger cars.

- Secondly, heavy duty diesel engines are more "hydrodynamic" in their lubrication behaviour than passenger car gasoline engines. This is because valve train friction is relatively less important in a heavy duty diesel engine than in a passenger car gasoline engine[29]. Figure 19 shows the calculated relative friction breakdown for a 2.0 litre gasoline engine and a 4.0 litre diesel engine. This effect manifests itself as a lack of friction modifier response in the majority of heavy duty diesel engines, as has been found in a number of field trials reported in the literature[30].

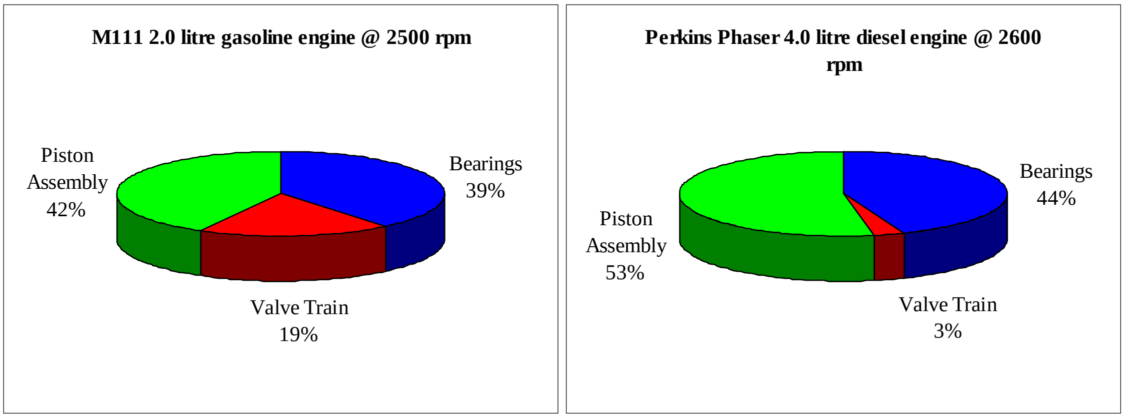

Figure 19 : Predicted engine friction breakdown[29] for a 2.0 litre gasoline engine (total loss = 1.5 kW) and a 4.0 litre diesel engine (total loss = 4.66 kW)

- Thirdly, there is an opportunity for heavy duty diesel engines to combine engine oil and transmission oil effects to produce a greater fuel saving. Bartz[5,31] has calculated the energy savings that can theoretically achieved by minimising losses in the engine and transmission. His general conclusions are that the savings that can be achieved from the engine (savings of order 3-5%) are greater than those that can be achieved from the transmission (savings of order 1-4%), but that the total savings that may be achieved by combining optimised engine and transmission lubricants could be significant. Simner[32] has also attempted to quantify the potential fuel economy benefit of using lower viscosity transmission lubricants.

- Finally, to meet NO_x emission limits, the traditional approach has been to retard the injection timing. This has the effect of pushing up the fuel consumption, and increasing the amount of soot in the lubricant. Future limits (e.g. EURO 3) are such that this approach, if it still works, will cause the typical fuel consumption to be approximately 10% higher than that in current engines. This has forced manufacturers to consider alternative approaches, the most commonly mentioned being the introduction of Exhaust Gas Recirculation (EGR). This latter approach is thought capable of meeting NO_x emission limits, whilst maintaining fuel consumption at today's levels. The drawback of this approach is that soot loading of the lubricant is likely to be substantially higher. The challenge for future heavy duty diesel engine lubricant formulators is to develop oils that can handle soot effectively, whilst maintaining durability and good fuel consumption.

Impact on Durability

Durability in Gasoline Engines

The potential disadvantage of moving to lower viscosity lubricants is the thinner oil film that is expected to exist between lubricated contacts within the engine. However, it should be remembered that in Europe, current oils have a relatively high viscosity (>3.5 mPa.s) compared to those marketed in the US and Japan. The move from oils that have High Temperature High Shear Viscosities (HTHSV) of 3.5 mPa.s to oils with a HTHSV of 2.9 mPa.s is not expected to have a major effect on engine durability for modern gasoline engines. Indeed, some of these engines may well be running on 2.9 mPa.s oils in the USA or Japan. Durability may well be of more concern when moving from oils with a HTHSV of 2.9 mPa.s to lower values (e.g. to 2.6 mPa.s).

The issue of durability is also not just limited to lubricant viscosity, but more generally to engine component design. Finger follower valve train systems, such as the Peugeot TU3 valve train, and the Ford Sequence VE finger follower valve train system, were capable of exhibiting high wear even with 3.5 mPa.s oils, if the anti-wear package used was sub-optimal. Bell[33] has shown that direct acting bucket tappet systems have inherently less wear than finger follower systems. Bell comments that "modern passenger car engines incorporating direct-acting cam/tappet valve trains are therefore expected to be less susceptible to wear and failure, and hence more tolerant to measures that could be taken to improve fuel economy and reduce phosphorus levels, than the engines that are used in the current valve train wear specification tests for motor oils." In addition, engines that have 4 valves per cylinder (rather than 2) tend to use lower spring loads, which will also help reduce wear (although it may cause other problems such as exhaust valve stick). The move towards roller follower valve train systems should also help to alleviate some of the concerns about valve train durability.

Bearing durability is also an area of concern, although it should be remembered that there are three important physical effects which help ensure bearings survive. One is that typical automotive lubricants have viscosities that are very sensitive to pressure (the commonly used Barus equation suggests that viscosity increases exponentially with pressure), and so as oil film thickness decrease, pressures rise, leading to higher oil viscosities, which help support the bearing loads. Secondly, the squeeze term in Reynolds' equation (which is often neglected) helps ensure thicker oil films. Thirdly, and perhaps most importantly, bearings deform when pressures are too high, again helping to sustain oil film thicknesses.

Over the years it has also been postulated that the inherent viscoelasticity of multigrade oils bestows a load bearing benefit on bearings. Okrent[34] suggested that at higher eccentricity ratios, the elasticity of a multigrade oil (which arises due to the polymer additives in the oil) gives a larger load bearing capacity that would be the case for an equivalent viscosity oil that did not have any elastic behaviour. Such an effect has been confirmed experimentally by Williamson et al[35].

In our laboratory, it has been observed that in a modern gasoline engine, well designed automotive bearings can be lubricated with oils as thin as 2.3 mPa.s without any observable wear on either con-rod or main bearings.

The assumption that lower viscosity lubricants automatically give rise to thinner oil films in key lubricated contacts in a gasoline engine is also open to question, particularly in the case of piston rings. Laser Induced Fluorescence measurements have found that, in a Nissan gasoline engine, the mid-stroke top ring oil film thickness was greater for an SAE-5W/20 lubricant than it was for an SAE-15W/40 lubricant. These effects were also observed in our laboratory for monograde lubricants. Similar effects have been observed by S.L. Moore of BP[36]. Figure 20 illustrates the observations. A qualitative explanation of such an effect could be as follows : There are two routes by which lubricant reaches the top piston ring. Route #1 (the "conventional" route) is that oil is left on the liner by the passage of the preceding ring. The higher the oil viscosity, the larger will be the oil film thickness left on the liner. Route #2 involves oil being transported to the top piston ring via the ring gaps (such flows have been observed by Nakashima et al[37]), and this is thought to favour lower viscosity lubricants. The precise balance between oil transported by the two routes will determine whether the oil film thickness under the top ring is greater for a lower viscosity oil or not.

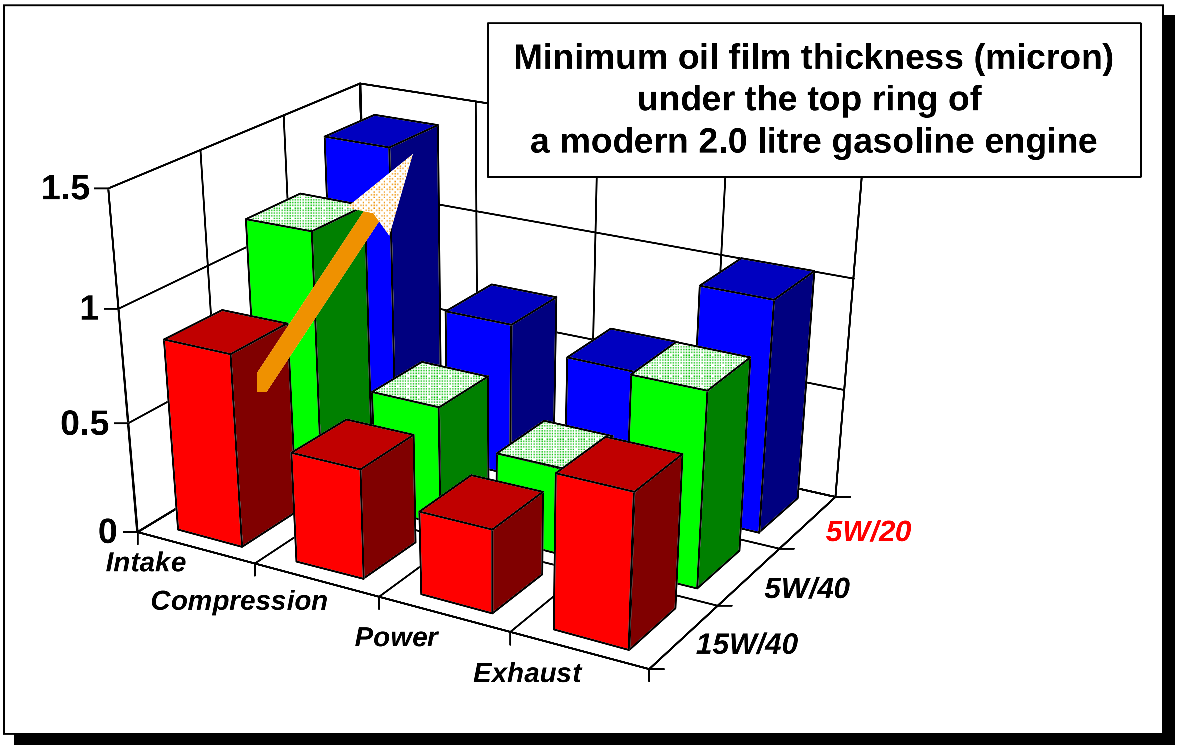

Figure 20 : Top ring oil film thicknesses (measured at mid-stroke) for a Nissan 2.0 litre gasoline engine

It must be pointed out, however, that despite the arguments outlined above, lower viscosity lubricants still have to be extensively field tested to ensure that durability is maintained. Figures 21-24 show data from field tests carried out on a Ford Mondeo, equipped with a 2.5 litre V6 engine. The lubricant was an SAE-5W/20 oil with a HTHS viscosity of 2.9 mPa.s.

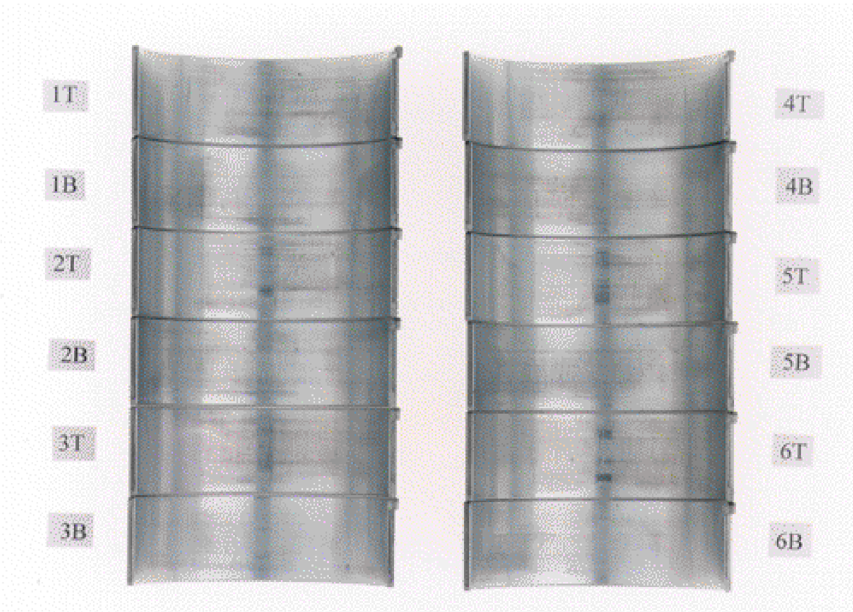

Figure 21 : Virtually no bearing wear observed after an arduous 12 day durability test using an SAE-5W/20 lubricant with a HTHS viscosity of 2.9 mPa.s

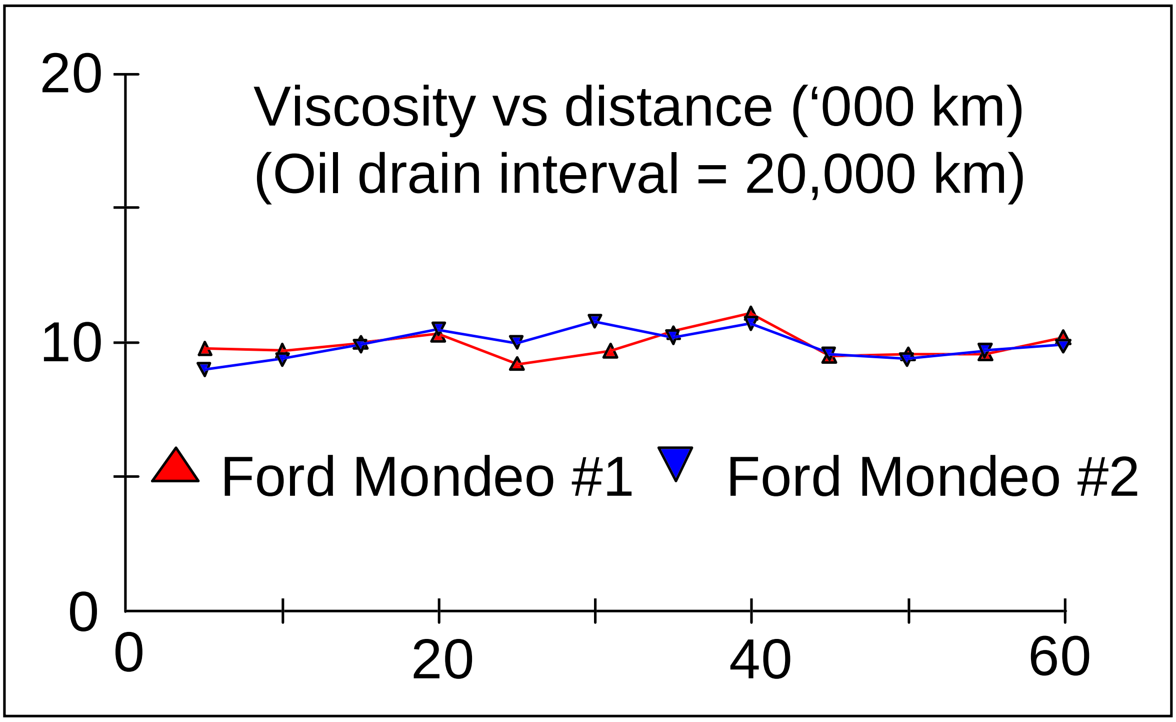

Figure 22 : Good viscosity stability demonstrated over 3 oil drain periods with an SAE-5W/20 oil

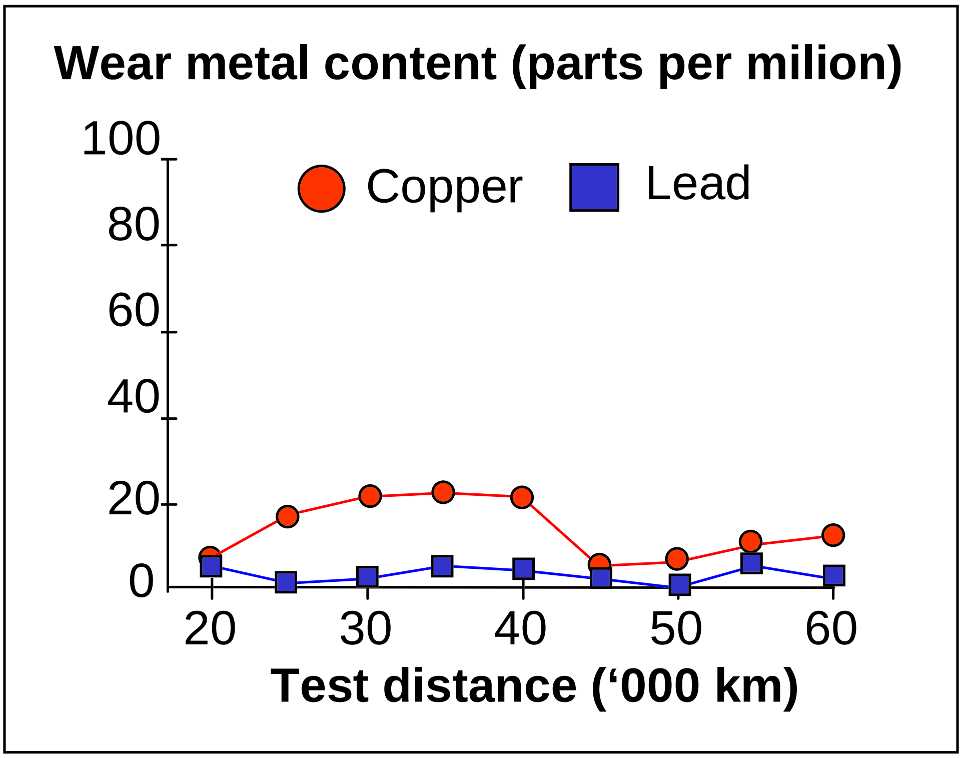

Figure 23 : Very low wear metal content in oil demonstrates no significant durability concerns with this engine using a 5W/20 oil with a HTHS viscosity of 2.9 mPa.s

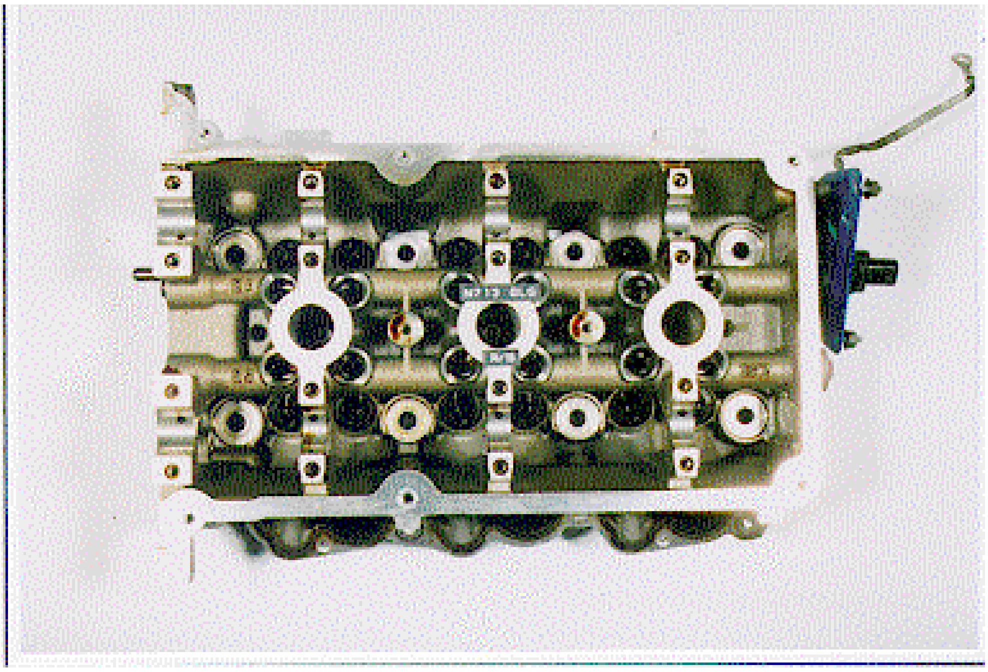

Figure 24 : Good engine cleanliness observed after 60,000 km with an SAE-5W/20 oil

Other workers have also published field trial data with SAE-5W/20 lubricants[38].

Durability in Heavy Duty Diesel Engines

The issue of durability is perhaps of more concern for heavy duty diesel engines. As mentioned above, however, although thinner lubricants have been demonstrated to give fuel economy benefits, there is no major pressure at present from the OEMs to move to lubricants whose HTHS viscosity is lower than 3.5 mPa.s. This is true not just in Europe but also in the USA and Japan. In addition, there are no heavy duty diesel engine fuel economy engine tests in place at present.

The key issues at the moment are (1) how to lubricate heavy duty diesel engines satisfactorily when the lubricants have high soot loadings, and (2) how to ensure longer oil drain intervals.

The second factor is an issue in Europe where OEMs such as Mercedes Benz have introduced heavy duty diesel engines with recommended service intervals of 120,000 km.

The traditional European approach to cope with lubricant soot loading is to use high dispersancy oils, which helps ensure that the soot particles do not agglomerate. Engine tests, such as the Cummins M11 Cross-Head wear test, try to discriminate between heavy duty diesel engine lubricants that cause high wear with a soot loading of around 5%. The new CH-4 specification is aimed at improving the performance of heavy duty diesel engine lubricants containing relatively high amounts of soot. It is worth pointing out that not all markets like high dispersancy oils, because of concerns about seal compatibility. In Japan, there is a preference for removing soot by using centrifugal filters in the engine. In the USA, there are also issues surrounding the use of high ash oils, since these do not perform particularly well in Caterpillar single cylinder piston assembly deposit tests.

OEMs such as Volvo and Scania prefer to qualify oils using long duration field trials. The trials typically take 2 years, which is becoming an issue since that is typically the timescale between new lubricant specifications! However, this approach does at least ensure that the lubricant performs adequately in the field, under realistic operating conditions, and the trials involve engine strip downs to ensure that component durability is acceptable.

This brief discussion shows that the focus on durability in heavy duty diesel engines is quite different to that in passenger car engines. It is expected that lubricant viscosities will still generally be greater than 3.5 mPa.s, but that the lubricants will be formulated to cope with higher soot loadings and viscosity grade may play a part in improving performance.. The issue is how to ensure durability with the high soot loadings envisaged when EGR engines emerge onto the market.

Conclusions

This review has attempted to give a snapshot of some of the issues surrounding fuel consumption in passenger car and heavy duty diesel engines, and how judicious lubricant design can give observable fuel economy advantages. A brief review was given of the different focus in European, Japanese and US markets. Then, the lubricant factors that affect fuel consumption were discussed. A substantial section of the review was devoted to engine friction modelling, both from first principles, and also from an empirical viewpoint. This enabled insight to be obtained into how different engines and operating conditions would be expected to respond to viscometry and the presence of friction modifiers. These insights were reinforced to some extent by the limited engine test data presented on two quite different gasoline engines, the Sequence VI-A engine test, and the Mercedes Benz M111 engine test.

The differences between gasoline and heavy duty diesel engine fuel consumption appetites were discussed, and these differences were demonstrated using engine friction models and field data.

A general discussion of engine durability was then undertaken, which again demonstrated the quite substantial difference in focus between passenger car oils and heavy duty diesel engine oils.

It is true to say that the vast effort put into increasing fuel economy has been driven by legislation (mainly US-based) aimed at improving the efficiency of passenger car gasoline engines. It is also fair to say that emissions legislation is driving the heavy duty diesel engine manufacturer to consider fuel consumption as a key selling factor in the future. Lubricant marketeers now typically offer a range of fuel economy lubricants, with demonstrated benefits in industry standard fuel economy engine tests and/or field trials. However, to achieve good fuel consumption, whilst still retaining low deposit forming tendency, good oxidation stability, good durability control, etc., still requires careful lubricant formulation, and requires a judicious choice of base oil, additive package and Viscosity Index Improver. Although not much has been said in this review about base oils, this choice can have a key effect on fuel economy[39], and the advent of Group II base oils (hydrotreated base oils, now being manufactured predominantly in the USA) is expected to be significant for formulating future fuel economy lubricants. The specifications that are now being proposed are pushing up the quality of base oils required, in particular in the case of reduced volatility to meet extended drain requirements. This raises issues over whether consumers will be prepared to pay for such products, and whether or not there is enough of the required base oil to supply the demand. These issues are somewhat peripheral to the main topic of the paper and have not therefore been discussed in great detail.

In future years, the Sequence VI-B engine test will be the first gasoline fuel economy engine test that tries to ensure the lubricant gives a fuel economy benefit throughout the oil drain interval (although the drain interval represented by the aging cycle is still short by European standards). It is also expected that there will be pressure for a heavy duty diesel engine fuel economy engine test. This might, however, have to test both engine oil and transmission oil to see a large enough benefit to be observed repeatably.

Continuing pressure on emissions and fuel efficiency will lead to hybrid vehicle designs such as Zero Emission Vehicles (ZEVs), Ultra Low Emission Vehicles (ULEVs,) and vehicles with continuously variable transmissions (CVTs) becoming more commonplace. These and other developments in vehicle technology are expected to continue to challenge the lubricant formulator.

Key references

1. "Engine Oil Performance Requirements & Reformulation for Future Engines & Systems", S.Korcek & M. Nakada, Proceedings of the International Tribology Conference, Yokohama 1995, p 783
2. "Fuel Economy Engine Oils : Present & Future", M. Yamada, Jap. J. Trib., **41**, No. 8, p 783, 1996
3. "Fuel Economy Factors in Lubricants", J.G. Damrath & A.G. Papay, SAE 821226
4. "Tribology of Reciprocating Internal Combustion Engines", S. Furuhama, Jap. Soc. Mech. Engrs Int. J., **30**, No. 266, p 1189, 1987
5. "Fuel Economy Improvement by Engine & Gear Oils", W.J. Bartz, 5th CEC International Symposium : Performance Evaluation of Automotive Fuels & Lubricants, CEC97-EL19
6. "Some Relationships Between the Viscometric Properties of Motor Oils & Performance in European Engines", J.C. Bell & M.A. Voisey, SAE 770378
7. "Engine Friction Reduction for Improved Fuel Economy", J.T. Kovach, E.A. Tsakiris & L.T. Wong, SAE 820085
8. "Engine Friction - A Change in Emphasis", M.L. Monaghan, Proc. Instn. Mech. Engrs, 202, No. D4, pp 215-226
9. "Engine Friction : The Influence of Lubricant Rheology", R.I. Taylor, Proc. Instn. Mech, Engrs, **211**, Part J, p 235, 1997
10. "Prediction of the Influence of Lubricant Formulations on Fuel Economy, from Laboratory Bench Tests", C. Bovington & H. Spikes, Proceedings of the International Tribology Conference, p 817, 1995
11. "Lubricant Related Fuel Savings in Short Trip, Cold Weather Service", T.J. Sheahan & W.S. Romig, SAE 750676
12. "A Cold-Start Track Test for Evaluating Fuel-Efficient Oils", G.B. Toft, I.Mech.E/SAE Joint Int. Conf. *on Fuel Efficient Power Trains and Vehicles*, London, October 1984
13. "FLARE : An Integrated Software Package for Friction & Lubrication Analysis of Automotive Engines - Part I : Overview & Applications", P.K. Goenka, R.S. Paranjpe & Y-R. Jeng, SAE 920487
14. "FLARE : An Integrated Software Package for Friction & Lubrication Analysis of Automotive Engines - Part II : Experimental Validation", R.S. Paranjpe & A. Cusenza, SAE 920488
15. "Sequence VIB Engine Test for Evaluation of Fuel Efficiency of Engine Oils - Part II. Stage Selection and Time Factor Determination", J. Sorab, S. Korcek, C.B. McCollum & K.W. Schriewer, SAE 982624
16. "Practical Applications of Lubrication Models in Engines", R.C. Coy, p 197, in *New Directions in Tribology* (edited by I.M. Hutchins, published by MEP, 1997)
17. "Friction Modelling for Internal Combustion Engines", Li Sheng Yang, PhD Thesis, University of Leeds, 1992
18. "Development of a Friction Prediction Model for High Performance Engines", K. Hamai, T. Goto & S. Kai, J. Soc. Trib. & Lub. Engrs., **47**, No. 7, p 567-573, 1990
19. "Trends in Engine Technology and Tribology", M. Nakada, Trib. Int., **27**, p 3-8, 1994

20. “Engine Friction Modelling”, R.H. Thring, SAE 920482

21. “Prediction of ASTM Sequence VI and VIA Fuel Economy Based on Laboratory Bench Tests”, A.K. Gangopadhyay, J. Sorab, P.A. Willermet, K. Schriewer, K. Fyfe & P.K.S. Lai, SAE 961140

22. “Influences of Lubricant Properties on ASTM Sequence VI and Sequence VI-A Fuel Efficiency Performance”, A.J. Moore, SAE 961138

23. “Critical Oil Physical Properties that Control the Fuel Economy Performance of General Motors Vehicles”, M.T. Devlin, W.Y. Lam & T.F. McDonnell, SAE 982503

24. “Reibungsverluste in Verbrennugsmotoren (Friction Losses in Combustion Engines)”, O.R. Lang, Schmiertechnic Tribologie, **29**, p 90-92, 1982

25. “Reducing Friction Losses in Automobile Engines”, M. Hoshi, Trib. Int., **17**, No.4, p 185-189, 1984

26. “Engine Oil Effects on Fuel Economy in GM Vehicles - Comparison with the ASTM Sequence VI-A Engine Dynamometer Test”, S.I. Tseregounis & M.L. McMillan, SAE 952347

27. “Sequence VIB Engine Test for Evaluation of Fuel Efficiency of Engine Oils - Part I. Aging Procedure for Determination of Fuel Efficiency Retention”, M.D. Johnson, C.B. McCollum, S. Korcek, R.K. Jensen, K.W. Schriewer, P.H. Neal & P.K.S. Lai, SAE 982623

28. “European Requirements for a Super High-Performance Diesel Oil”, D.C. Colbourne, Truck Technology International, 1988, pp 108-111

29. “Engine Friction Lubricant Sensitivities : A Comparison of Modern Diesel and Gasoline Engines”, R.I. Taylor, Esslingen 1998

30. “Fuel Savings with Multigraded Engine Oils in Medium-Speed Diesel Engines”, Stauffer, Zahalka & Kornmann, Lubrication Engineering, **40**, No. 12, pp 744-751

31. “Fuel Economy Improvement by Engine and Gear Oils”, W.J. Bartz, Proceedings of the 24th Leeds-Lyon Symposium on Tribology (published in Tribology for Energy Conservation, Tribology Series, pp 13-24, 34, 1998, Elsevier, Editor : D.Dowson)

32. “Quantifying the Potential Fuel Economy Benefit of Transmission Lubricants”, D. Simner, Esslingen 1998

33. “Effects of Valve Train Design Evolution on Motor Oil Anti-Wear Requirements”, J.C. Bell, paper CEC97-EL02, 5th CEC International Symposium on the Performance Evaluation of Automotive Fuels and Lubricants, 1997

34. “Engine Friction and Bearing Wear. III. The Role of Elasticity in Bearing Performance”, E.H. Okrent, ASLE Transactions, **7**, 1964, pp 147-152

35. “The Viscoelastic Properties of Multigrade Oils and Their Effect on Journal-Bearing Characteristics”, B.P. Williamson, K. Walters, T.W. Bates, R.C. Coy & A.L. Milton, J. Non-Newtonian Fluid Mech., **73**, 1997, pp 115-126

36. “Piston Ring Oil Film Thickness - The Effect of Viscosity”, S.L. Moore, SAE 850439

37. “Influence of Piston Ring Gaps on Lubricating Oil Flow into the Combustion Chamber”, K. Nakashima, S. Ishihara & K. Urano, SAE 952546

38. “Development and Field Test Performance of Fuel Efficient SAE 5W-20 Oils”, A. Yaguchi & K. Inoue, SAE 952341

39. “Fuel Efficient Lubricants and the Effect of Special Base Oils”, T.E. Kiovsky, N.C. Yates & J.R. Bales, J.STLE, **50**, No. 4, p 307, 1993