The Environmental and Economic Importance of Mixed and Boundary Lubrication

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ABSTRACT

One route to reducing CO_2 emissions is to improve the energy efficiency of machines. For example, conventional combustion engines are being downsized (and also down-speeded), and are now running on lower viscosity lubricants (such as 0W-20 or lower viscosity grade lubricants) and often also have stop-start systems fitted.

Some of these changes may result in higher levels of mixed and boundary friction, and so estimating the friction losses due to mixed/boundary friction, and the corresponding wear levels, is becoming of increasing importance. There is recent experimental evidence that traditional approaches (such as the Greenwood & Tripp model [1]) to predicting friction in mixed and boundary friction tend to underestimate these losses [2-5].

A new model is described, based on experimental data, that estimates the proportion of mixed/boundary lubrication, X, as a function of the λ value (where λ is the ratio of the oil film thickness separating the surfaces to the combined root mean square roughness of the surfaces). The precise equation that describes the way in which X varies with λ takes the form of a "reverse S-curve" which makes sense physically since S-curves arise naturally in growth processes and the real area of contact of rough lubricated surfaces grows as $1/\lambda$ increases.

Numerical estimates of the amount of mixed/boundary lubrication losses in internal combustion engines are made and compared with recently published experimental data [2, 6]. In addition, these improved calculations are used to estimate both the financial cost of mixed/boundary lubrication for today's vehicle fleet, and the CO_2 emissions associated with these losses.

1. INTRODUCTION

In most internal combustion engines, friction losses in engines and transmissions are primarily due to hydrodynamic (journal bearings, piston assembly), or elastohydrodynamic lubrication (valve train, gears). For hydrodynamic lubrication, if F (Newtons) is the friction force, h_{min} (m) is the minimum film thickness separating the surfaces, U (m/s) is the relative sliding speed of the surfaces, W (N) is the load and η (mPa.s) is the lubricant viscosity, then h_{min} and F should vary according to [7]:

$$h_{min} \propto \sqrt{\frac{\eta U}{W}}$$

$$F \propto \sqrt{\eta U W}$$

For fixed speeds and loads, clearly the above equations show that both friction and minimum oil film thickness should vary with the square root of viscosity, for hydrodynamic contacts. This is why friction losses increase for cold starts, when lubricant temperatures are low and lubricant viscosities are high, and also helps to explain the trend over the last 30-40 years towards lower viscosity lubricants, as shown in Table 1 below:

Table 1: Typical viscosities of different lubricant viscosity grades, showing the decrease in engine oil viscosity over time

	Typical V_k40 (cSt)	**Typical V_k100 (cSt)**	**Typical HTHS viscosity (mPa.s)**	**Approx viscosity (mPa.s) at -15°C**	**Approx Year**
SAE 20W-50	144.8	17.8	4.1	5,900	Before 1980
SAE 15W-40	114.3	14.9	3.5	2.900	1990
SAE 10W-30	72.3	10.8	3.2	1,900	1995
SAE 5W-30	57.4	9.9	2.9	1,100	2000
SAE 0W-20	44.4	8.3	2.6	700	2015
SAE 0W-8	26.4	5.5	1.9	≈ 250	Future

However, for hydrodynamic contacts, since the oil film thickness also varies with the square root of lubricant viscosity, oil films will get thinner as viscosities are decreased and so there is more chance of the contact entering the mixed and boundary friction regime. This is more likely, of course, when speeds are low and loads are high.

When mixed/boundary lubrication occurs, it can be assumed that a portion of the total load, W (N), is carried by the asperities, W_A, with the remainder of the load carried by the fluid, W_F. If the respective friction coefficients are written as f_A and f_F, the total friction force is F_{TOTAL} and the overall friction coefficient is f, then the following equations hold:

$$W = W_A + W_F$$

$$F_{TOTAL} = f_A W_A + f_F W_F$$

$$f = \frac{F_{TOTAL}}{W} = f_A \frac{W_A}{W} + f_F \frac{W_F}{W}$$

If X is defined to be equal to W_A/W, then X can be considered to be a measure of the amount of mixed/boundary lubrication (since X = 0 when hydrodynamic conditions hold, since the rough surfaces are completely separated by the fluid film and so W_A=0, and, in addition, X=1 when no fluid film separates the moving surfaces, since in that case W_F=0). The total friction coefficient can thus be written as:

$$f = f_A X + f_F (1 - X)$$

Such an equation has previously been reported by Olver and Spikes [8].

It is clearly of great interest to know how X varies with the λ ratio (where λ is the ratio of the oil film thickness separating the surfaces to the combined root mean square surface roughness). In the next section, experimental data is used to derive an equation for X in terms of λ, and this expression is then compared to other equations that have been reported in tribology journals.

This equation can be used to estimate the friction losses due to mixed/boundary lubrication in internal combustion engines. By considering the total number of engines in use, total friction losses due to mixed/boundary lubrication, and the equivalent CO_2 emissions due to these losses, can be estimated.

2. MODELS FOR PREDICTING FRICTION IN MIXED/BOUNDARY LUBRICATION

Numerous well-known models for mixed/boundary friction assume that asperities deform elastically [1, 9-11]. For example, the Greenwood-Williamson model [10] gives expressions for the way in which the load supported by the asperities varies with the separation, d, of the rough surfaces. If it is assumed that the asperities height distribution drops off exponentially, then it is reported [10] that:

$$W(d) \propto \exp\left(-\frac{d}{\sigma}\right)$$

Where σ is the root mean square roughness of the rough surface (in reference [10], only one surface was assumed to be rough, with the other surface being flat).
If the asperity height distribution varies according to a Gaussian distribution, then:

$$W(d) \propto F_{3/2}\left(\frac{d}{\sigma}\right)$$

Where:

$$F_n(u) = \frac{1}{\sqrt{2\pi}} \int_u^{\infty} (s-u)^n . \exp\left(-\frac{s^2}{2}\right) ds$$

A later model [1] which assumed both surfaces were rough found that:

$$W(d) \propto F_{5/2}\left(\frac{d}{\sigma}\right)$$

The above equations can be recast in terms of the proportion of mixed/boundary lubrication, X, by dividing W(d) by W(0), and so, the above equations become, respectively (where we have also written d/σ as λ):

$$X = \exp(-\lambda) \qquad X = \frac{F_{3/2}(\lambda)}{F_{3/2}(0)} \qquad X = \frac{F_{5/2}(\lambda)}{F_{5/2}(0)}$$

A later model due to Bush et al [11] gave the equation below for X:

$$X = erfc\left(\frac{\lambda}{\sqrt{2}}\right)$$

Rough surface contact models that assume elastic deformation of asperities strictly only apply when the real area of contact is small (less than a few %). However, rough surface contact models are also available in the other limit, when real areas of contact are high [13]. These models were originally motivated by the study of rubber surfaces.

However, in practice, the Greenwood and Tripp model is still widely used today for predicting the load carried by asperities in rough surface contacts. A comparison of the exponential Greenwood Williamson model, the Greenwood Tripp and Bush models is seen in Figure 1. For λ=1, the value of X is predicted to be about 0.368 for the exponential Greenwood and Williamson model [10], about 0.317 for the Bush model [11] and only about 0.131 for the Greenwood and Tripp model [1]. It is clearly of interest, in predicting mixed/boundary friction, to clarify which of these values of X is more reliable.

Recently, good quality experimental data became available for the proportion of mixed/boundary friction versus λ. Experimental details can be found in [3]. Figure 2 shows typical experimental data obtained from the Mini-Traction Machine [3], for a range of lubricants, also including Mini Traction Machine data from other sources [13,14]

One of the features of Figure 2, is that, despite the wide range of lubricants tested, the experimental data fits reasonably well on a single "master curve", and so it is reasonable that a single mathematical function should be able to fit all the data. The other feature of Figure 2 is that the data appears to follow a "reverse S-curve". In fact, if the data had been plotted against $1/\lambda$, rather than λ, then the data would follow a standard "S-curve". Since the real contact area is expected to grow as $1/\lambda$ increases, it is physically reasonable that the real contact area should grow as an "S-curve", as such curves are known to appear in many growth processes.

A suitable function, which takes the form of a "reverse S-curve" and provides a good fit to the experimental data is [4,5]:

$$X = \frac{1}{(1+\lambda^k)^a}$$

The values of k and a are given by $k \approx \frac{3}{2}$ and $a \approx \frac{4}{3}$. The above equation predicts that $X \approx 0.397$ when X=1, which suggests that the widely used Greenwood and Tripp model [1] substantially underestimates the amount of mixed and boundary friction in the range $1 < \lambda < 3$.

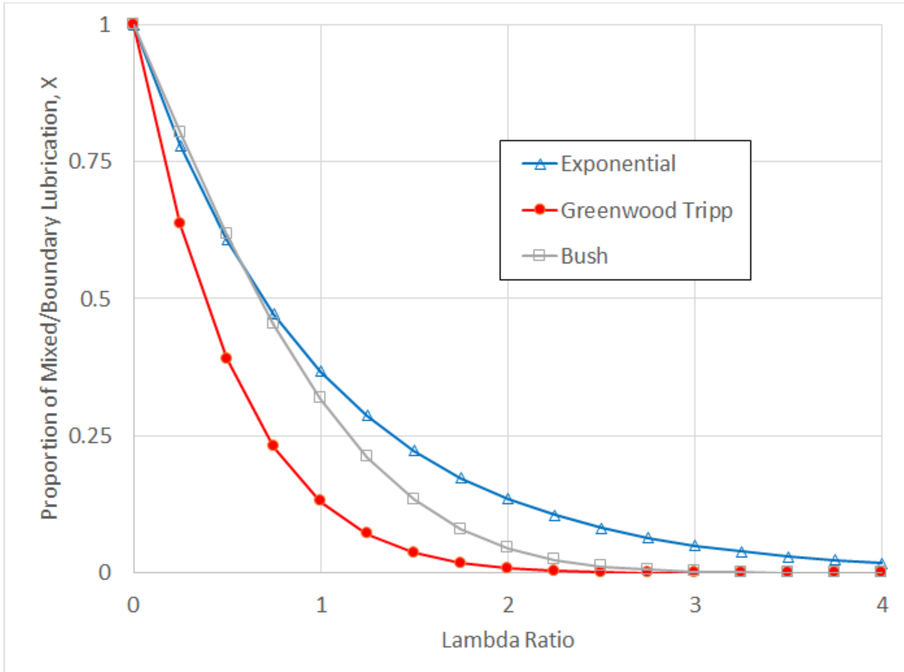

Figure 1. Proportion of mixed/boundary lubrication versus λ, for different rough surface models

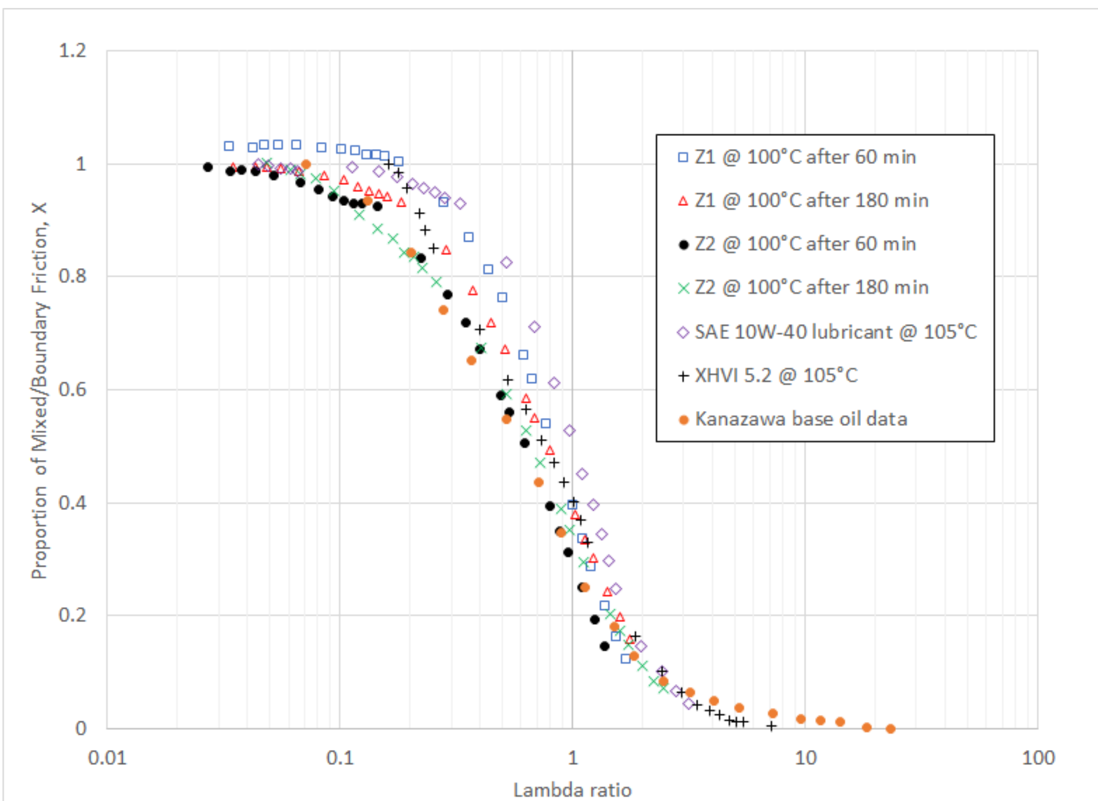

Figure 2. Experimental data showing the proportion of mixed/boundary lubrication versus λ, for different rough surface models, for a range of lubricants [5]

3. ESTIMATES OF MIXED/BOUNDARY LUBRICATION FOR INTERNAL COMBUSTION ENGINES

In internal combustion engines, there are a range of components that are in different lubrication regimes. For example, journal bearings are designed to operate hydrodynamically, and so it would be expected that the surfaces are fully separated by a lubricant, although even in these components, some mixed lubrication can occur at stop/start, and/or for high loads, low speeds with modern low viscosity lubricants, particularly if there is also some fuel dilution [15,16]. On the other hand, the valve train is designed to operate with very thin, elastohydrodynamic, oil films, and so for this component, the λ ratio will be well below 3 for the entire range of engine operation, when the engine is fully warmed up. The piston assembly predominantly operates in the hydrodynamic regime, but it is thought that the top piston ring (exposed to high combustion chamber pressures) will have the greatest amount of mixed/boundary lubrication, particularly at low speeds and high loads.

Published experimental data on piston assembly friction [6] for a fully warmed up engine shows that the amount of mixed/boundary friction drops off rapidly as engine speeds increase, as expected (since oil film thickness, and the λ ratio increase with speed). An example of this data [6] is shown in Figure 3, based on a 1990 2.0 litre gasoline engine. Clearly, the amount of mixed/boundary lubrication will depend on the driving cycle – city type driving, with many stop-starts and low speeds will tend to have more mixed/boundary lubrication than motorway type driving. However, one complication is that the actual fuel consumption is lower at low engine speeds compared to higher engine speeds. For example, in the older European NEDC driving cycle that was used until around 2020, the engine idled for 25% of the time, but this only contributed 10% to the overall fuel consumption (and this is a major reason why modern cars are equipped with stop-start systems).

For a "typical" driving pattern, with an average engine speed of 2000 rpm, the amount of mixed/boundary lubrication in the piston assembly, according to Figure 3 will be approximately 20%. Recall this data is for a 30-year-old 2.0 litre gasoline engine, and more modern engines will likely have less mixed/boundary friction than this (due to the use of lightweight valves with softer springs, and also due to the widespread use of "stop/start" systems that switch the engine off instead of keeping it running at idle speed). In total, at 2000 rpm, the overall engine power loss was measured to be 1570 Watts, with the proportion of mixed/boundary lubrication being about 32%. For an SAE 15W-40 lubricant, the measured friction was 1800 Watts, with the proportion of mixed/boundary lubrication being about 19%.

Holmberg et al [17] have estimated that the average amount of mixed/boundary lubrication in passenger cars was around 10%. For modern vehicles, with low viscosity lubricants, it is reasonable to assume the amount of mixed/boundary friction in modern passenger cars is in the range 10-20%, although in older vehicles, the amount could be higher. Given that a realistic average fuel consumption for modern gasoline cars is about 7 litres per 100 km, and that an average gasoline car is driven 16,000 km per year, then typically the amount of fuel used per car per year is about 1120 litres. The amount of fuel used to overcome mixed/boundary friction is thus in the range 112 to 224 litres for one car annually. In the UK, there are approximately 19 million gasoline fuelled cars (and 12 million diesel passenger cars), and so we estimate that, for gasoline cars, between 2 to 4 billion litres of gasoline are used to overcome mixed/boundary friction (out of a total annual gasoline usage in the UK of about 17 billion litres). When 1 litre of gasoline is combusted, 2.4 kg of CO_2 is emitted, so it is estimated that between 5 to 10 million tonnes of CO_2 are emitted per year, due to mixed/boundary friction from gasoline fuelled vehicles in the UK.

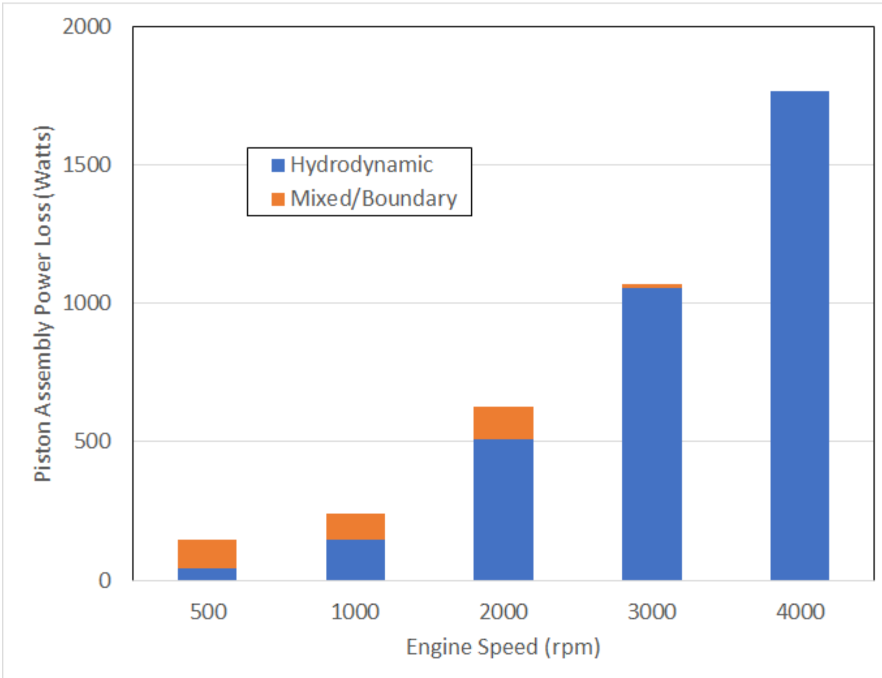

Figure 3. Experimental data [6] for the measured piston assembly power loss (Watts) versus engine speed, for an SAE 0W-8 lubricant in a 1990's 2.0 litre gasoline engine

4. DISCUSSION

A significant amount of fuel is used, with associated CO_2 emissions, for overcoming mixed and boundary lubrication in machines. It has been estimated that for a typical UK passenger car, under "average" driving conditions, between 10-20% of friction losses are due to mixed/boundary lubrication. It has also been reported [2] that when comparing measured friction losses with predictions, the commonly used Greenwood & Tripp model [1] tends to underpredict the amount of mixed/boundary lubrication. A new model [5], based on experimental data [3] can be used to make more accurate estimates of the amount of mixed/boundary lubrication in machine elements, as a function of the λ ratio.

Clearly, reducing mixed/boundary friction, by making surfaces smoother, using different materials (such as DLC coatings), and/or by using friction modifiers, can result in significant friction savings. It is anticipated that as the pressure on energy efficiency increases and the move to lower viscosity lubricants intensifies, the need to better predict mixed/boundary friction will become more and more important.

5. CONCLUSIONS

One route to reducing CO_2 emissions and energy usage is to improve the energy efficiency of machines, such as internal combustion engines. Lubricant viscosities have been decreasing since the 1990's, in order to reduce the overall friction in engines, and this approach has been effective in substantially reducing hydrodynamic friction. However, a reduction in lubricant viscosity has also led to thinner oil films separating the moving surfaces and increased the chances of mixed/boundary lubrication occurring.

It is thus becoming more important to be able to accurately predict mixed/boundary friction losses. Recent experimental data [3] has been used to develop an easy to use, and more accurate equation for predicting the amount of mixed/boundary lubrication in a tribological contact.

In addition, the contribution of mixed/boundary lubrication to energy usage, and CO_2 emissions of passenger car gasoline engines in the UK has been quantified. Tribology can help to reduce these losses by optimizing the surface finish, the materials used, and by incorporating friction modifiers into lubricants.

6. ACKNOWLEDGEMENTS

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