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The Economic and environmental impact of mixed and boundary lubrication

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Introduction

It is well known that real engineering surfaces, at the micron scale, are rough (Figure 1). The roughness of surfaces depends on the type of surface finish used. Typical engineering steels have a root mean square surface roughness of between 0.5 to 1 μm, which is similar to the minimum oil film thickness that separates lubricated components [1]. The fact that surfaces are rough has some major implications for lubrication. Firstly, for a given load, engineers will usually use the simple geometrical area of a sample to work out the average pressure. So, for a sample of size 10 mm x 10 mm, an applied load of 100 N would give an average pressure of 10^6 N/m^2 (or about 10 atmospheres). However, for rough surfaces, the real contact area is much smaller than the geometrical area of the contact and so the actual pressure on each individual asperity is much higher – if the real contact area is just 1% of the geometrical area, the actual pressures will be 100 times greater. For brand new components, when they are first loaded, these high pressures will result in substantial plastic deformation of the highest asperities, and the process of "running-in" [2-4], over a few tens or hundreds of hours usually results in a smoother surface, and the change in surface shape of the component that can occur can also result in thicker oil films. It should be said that there are two possibilities for highly loaded rough surfaces, either they "run-in" and subsequently have a long life (many thousands of hours), or they fail quickly during the "running-in" process.

Once "running-in" has finished, there can still be metal-to-metal contact, during stop-starts for example, or at times when there is low speed and/or high load operation. However, any metal-to-metal contact that occurs after "running-in" generally involves elastic deformation of the asperities, rather than plastic deformation.

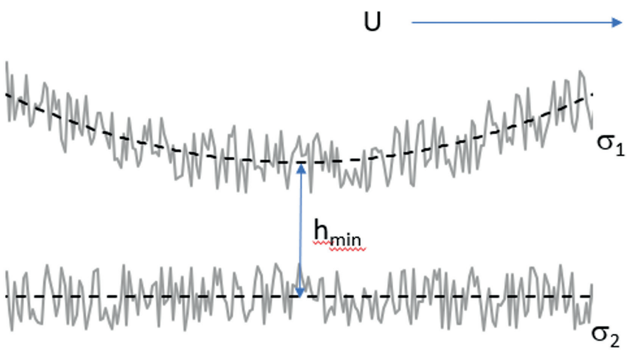

Figure 1: *Schematic showing rough surface lubrication – whether or not the rough asperities touch each other depends on the value of the minimum oil film thickness, h_{min} (dotted lines show the centre line average of each surface).*

Much useful information can be captured using the schematic curve of Figure 2, which shows how the friction coefficient typically varies with the λ ratio, defined as [5]:

$$\lambda = \frac{h_{min}}{\sqrt{\sigma_1^2 + \sigma_2^2}}$$

Where h_{min} (m) is the minimum oil film thickness separating the moving, lubricated surfaces, with σ_1 and σ_2 (m) being the root mean square surface roughnesses of each surface.

Boundary lubrication is where the applied load is entirely supported by the rough metal asperities and is generally accepted to occur if $\lambda < 1$. In mixed lubrication, some of the applied load is supported by the rough metal asperities and some is supported by the fluid film. Mixed lubrication tends to occur when $1 < \lambda < 3$. When the rough metal surfaces are completely separated by a fluid film (which is thought to occur when $\lambda > 3$) then all the applied load is carried by the fluid film.

Most machines are designed to have full fluid lubrication, so as to ensure a long machine lifetime. In an effort to improve energy efficiency, the viscosity of lubricants has been decreasing over the last 20-30 years (see Table 1), since a reduction in lubricant viscosity results in a decrease in friction losses. However, a decrease in lubricant viscosity also results in a thinner oil film separating the moving surfaces, and so there is an increased risk of mixed and boundary lubrication.

There is therefore increasing interest in being able to accurately estimate mixed/boundary friction losses, and in methods/materials that can help decrease mixed/boundary friction, as described in the following sections of the paper, and the costs of mixed/boundary friction are estimated, together with the expected CO_2 emissions that may arise from mixed/boundary friction.

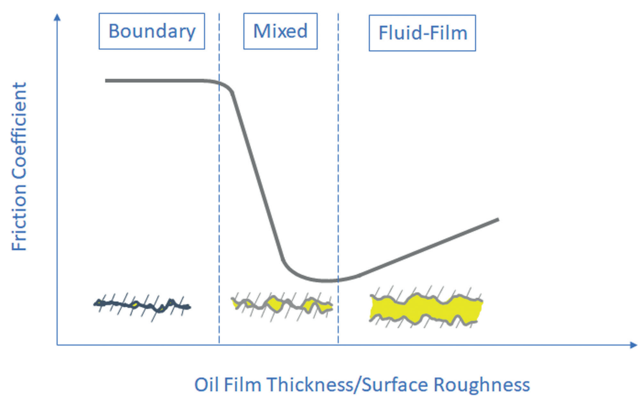

Figure 2: *Schematic curve showing how friction coefficient varies with λ.*

	Typical kinematic viscosity at 40°C (V_k40)	Typical kinematic viscosity at 100°C (V_k100)	Typical High Temperature High Shear Viscosity (mPa.s)	Year
SAE 15W-40	114.3	14.9	3.5	1990
SAE 5W-30	57.4	9.9	2.9	2000
SAE 0W-20	44.4	8.3	2.6	2015
SAE 0W-8	26.4	5.5	1.9	Future

Table 1: *Table showing how viscosities (for passenger car lubricants) have been decreasing over the last three decades. The High Temperature High Shear Viscosity is measured at a temperature of 150°C and a shear rate of 10^6 s^{-1}.*

Mixed/Boundary Lubrication in Real Machines

Although Figure 2 shows a useful schematic of how the friction coefficient should vary with λ, it is also useful to measure friction losses in real machines. Figure 3 shows measured total engine friction losses (measured as a Friction Mean Effective Pressure – FMEP – in bars) versus engine speed (revs/min) for a 1990's 2.0 litre gasoline engine, when lubricated with a 0W-8 lubricant (which does not contain a friction modifier), with a sump oil temperature of 93°C (representing a fully warmed up engine) [6]. Figure 3 also shows, separately, the fluid film friction and the mixed/boundary friction (in an engine, most of the mixed/boundary friction will come from the valve train and piston assembly).

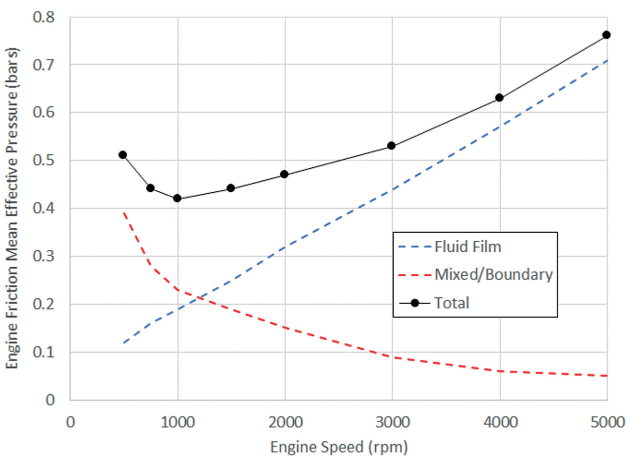

Figure 3: *Measured Friction Mean Effective Pressure (FMEP), versus engine speed (revs/min) of a 1990's 2.0 litre gasoline engine when lubricated with a 0W-8 oil, with a sump temperature of 93°C [6]. The effect of mixed/boundary lubrication can clearly be seen for engine speeds lower than 1000 rpm.*

Friction coefficient can also be measured in laboratory tribological test equipment. A typical example is shown in Figure 4, where the friction coefficient of a lubricant is measured in a Mini Traction Machine [7] as a function of the mean speed between the ball and disk. It should be noted that results of such experiments are typically plotted on a logarithmic scale, which gives the impression that boundary friction is much more important that it actually is. Figure 4 shows results plotted on both a logarithmic scale and on a linear scale.

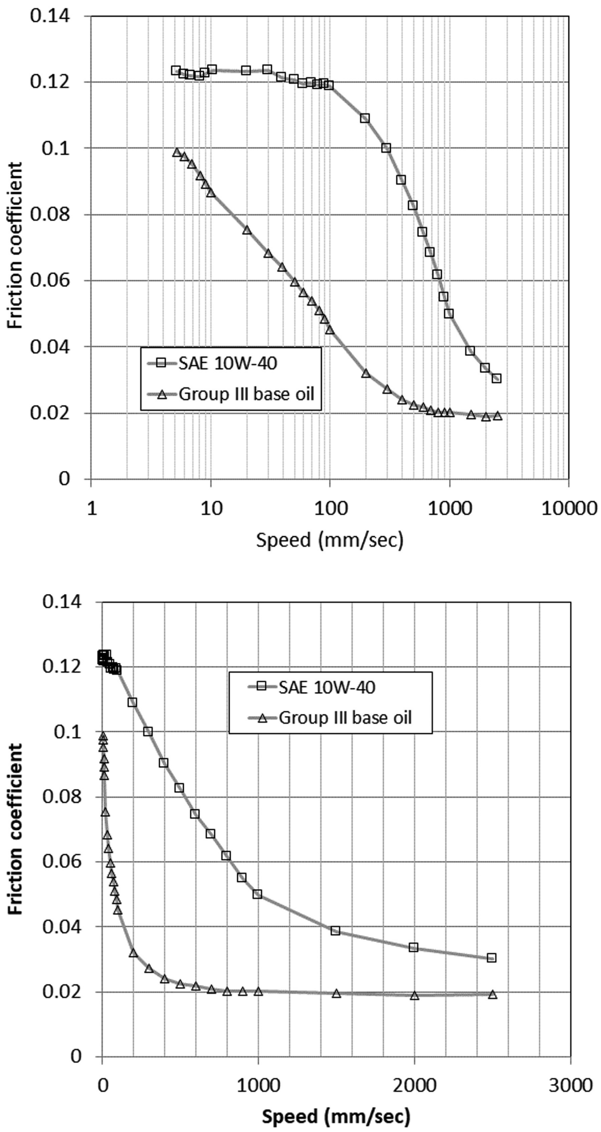

Figure 4: *Typical friction measurements from laboratory tribological test equipment (in this case a Mini Traction Machine [7]). The graphs show the same data, although the upper graph uses a logarithmic horizontal scale, whereas the lower graph uses a linear horizontal scale.*

As mentioned in the Introduction, the amount of mixed/boundary lubrication also depends critically on whether the components are brand new or have been allowed to "run-in". Figure 5 shows the substantially greater amount of measured mixed/boundary friction of a new piston ring, compared to a "run-in" ring [8]. It has also been reported that the friction power loss of a new piston ring pack, compared to a "run-in" piston ring pack, is approximately 10-15% higher [9].

Figure 5: *Measured friction force of new piston assembly compared to "run-in" piston assembly. Data replotted from reference [8].*

Models for Mixed/Boundary Lubrication

If W is the load applied to a lubricated contact, and it is assumed that a fraction of the load, WA, is carried by rough asperities, and the remaining fraction, WF, is carried by the fluid, then it is reasonable to assume that:

$$W = W_A + W_F \qquad [2]$$

If the friction coefficients for the asperities and the fluid are fA and fF, and the overall friction coefficient is f, then the following equations result:

$$f = \frac{F_{TOTAL}}{W} = f_A \frac{W_A}{W} + f_F \frac{W_F}{W} = f_A \frac{W_A}{W} + f_F \left(1 - \frac{W_A}{W}\right) \qquad [3]$$

If we write X = WA/W, then the above equation simplifies to become:

$$f = f_A X + f_F (1 - X) \qquad [4]$$

The above equation has been previously reported by Olver and Spikes [10].

The parameter X can be regarded as a useful measure of the amount of mixed/boundary lubrication in a contact, since when there is no fluid film separating the surfaces, λ = 0, and X = 1. On the other hand, for large values of λ, greater than 3, the rough surfaces are assumed to be completely separated, and in that case X = 0. In order to better predict mixed and boundary friction, it would be extremely useful to know how X varies with λ.

There have been many models that have attempted to predict mixed/boundary friction. Most of these models assume that asperities deform elastically, which is not as drastic an assumption as it looks, particularly after "running-in" has occurred, although such an assumption is likely to lead to incorrect results if such models are applied to brand new surfaces. A recent, thorough, review of mixed/boundary friction models is available for the interested reader [11]. The main analytical models worth mentioning are those due to Archard [12], Greenwood and Williamson [13], Greenwood and Tripp [14], Bush [15] and Persson [16]. In recent years, most modelling efforts have been focussed on numerical analysis of real rough surfaces (see for example reference [17]).

These models generally predict the load carried by the asperities as a function of the separation of the centre line average of the rough surfaces, d (m). If the load carried is WA(d), then the parameter X described above can be calculated by dividing WA(d) by WA(0). For example, the Greenwood and Williamson model [13] describes a rough surface, which can have different probability distribution functions, contacting a perfectly flat surface. In the case where there is an exponential probability distribution function, it is found that:

$$X = \exp(-\lambda) \qquad [5]$$

If the value of X for λ=1 is considered, then the Greenwood and Williamson model [13], with an exponential probability distribution function predicts a value for X of about 0.368 when λ=1. The Bush model [15] predicts a value for X of 0.317 for λ=1. The Greenwood and Tripp model [14], however, which considers two rough surfaces contacting each other, with each surface having a Gaussian probability distribution of roughness, only predicts a value of 0.131 when λ=1. It is clearly of interest to know which of these different models is the right one to use when predicting mixed/boundary friction.
Recent, useful, data has become available [18] in which the variation of X with λ has been reported from experimental measurements. Although there is some spread in the reported data, experiments indicate that for λ=1, X lies in the range 0.3 to 0.5, which suggests that the Greenwood and Tripp model [14] substantially underestimates the amount of mixed/boundary lubrication in contacts, and such a finding was also reported by Morris et al [19].

A mathematical fit to the experimental X versus λ data was reported [20], and it was found that a good fit to the data was provided by:

$$X = \frac{1}{(1+\lambda^k)^a} \qquad [6]$$

Where k ≈ 3/2 and a ≈ 4/3.

$$X = \frac{1}{(1+\lambda)^2} \qquad [7]$$

A similar equation (see below) had previously been reported by Olver and Spikes [10]:

Encouragingly, both equations (6) and (7) have the same asymptotic behaviour at high values of λ, such that $X \propto 1/\lambda^2$. For equation (6), the value of X when λ=1 is 0.397, whereas the value of X calculated using equation (7), when λ=1 is 0.25.

It is recommended that equation (6) should be used to calculate mixed/boundary friction, since (1) it is based on experimental data, (2) the equation takes the form of a "reverse S-curve", and this would be expected since the real contact area should increase (grow) with load (i.e. as 1/λ), and S-curves arise naturally in many growth processes [21].

Economic Impact of Mixed and Boundary Friction

For passenger car engines, experimental data is available on the amount of mixed/boundary friction in valve trains, piston assemblies and journal bearings. The amount of mixed/boundary friction depends strongly on engine speed, and an analysis of reported data [6] has found that up to 10% of fuel consumed on a fuel economy test cycle, which used a 1990's 2.0 litre gasoline engine, is used to overcome mixed/boundary friction. For more modern engines, fitted with stop-start systems, and with more advanced valve train designs using lower weight valves and softer springs, the figure will most likely be closer to 5%.

Therefore, it can be assumed that, for passenger cars, approximately 5-10% of total fuel consumption is used to overcome mixed and boundary friction. In the UK, assuming a typical vehicle mileage of 16,000 km, and an average fuel consumption of 6 litres/100 km, the annual fuel consumption of an average car will be around 1000 litres. Therefore, each passenger car in the UK will use approximately 50-100 litres of fuel per year, simply to overcome mixed/boundary friction.

Given that there are around 30 million passenger cars in the UK, this equates to 1.5 to 3 billion litres of fuel per year, with an associated cost of about £2.25 to £4.5 billion per year.

If these figures can be extrapolated worldwide, with 1 billion passenger cars, mixed/boundary friction results in approximately 50 to 100 billion litres of fuel being consumed per year, at a total cost of about $100 to $200 billion.

Note that these economic estimates do not include any costs for wear, breakdowns, repairs etc, that may occur due to mixed/boundary friction. Further work is needed to provide realistic estimates for these additional costs.

Environmental Impact

It is well known that CO_2 emissions result from the burning of fossil fuels. For gasoline, approximately 3 kg of CO_2 are emitted for each litre burnt, of which 2.4 kg is emitted directly through combustion whilst another 0.6-0.7 kg is emitted during the manufacture of gasoline from crude oil, and subsequent transportation of gasoline to fuel stations [22].

The previous section has suggested that annually, approximately 50 to 100 billion litres of fuel are consumed simply to overcome mixed/boundary friction in passenger cars. Therefore, this would result in annual CO_2 emissions of between 150 million to 300 million tonnes.

Some countries, or regions, set a "carbon tax", i.e. a price on each tonne of CO_2 emitted, to encourage producers to reduce their CO_2 emissions. Where such a price is set, at the time of writing, a typical figure is $50 per tonne of CO_2. Therefore, the cost of CO_2 emissions due to mixed/boundary friction, if carbon taxes were to be widely used, is potentially in the region of $7.5 to $15 billion, which, whilst far less than the direct fuel costs associated with mixed/boundary friction, is still a significant sum.

Mitigation Efforts to Reduce Impact of Mixed/Boundary Lubrication

Given the large costs and CO_2 emissions associated with mixed/boundary lubrication, much research is ongoing to attempt to reduce mixed/boundary friction in machines. The following measures, for example, are now widely used:

- Friction modifier (FM) additives are widely used in low viscosity passenger car engine lubricants (these are usually Molybdenum based FMs, or organic FMs, usually based on glycerol mono-oleate or oleyl-amide, or both [25]).
- Stop-start systems are now widely used to stop excessive idling in passenger cars and heavy-duty trucks. For city-type, stop-start driving, these systems can usually result in a 10-15% reduction in fuel consumption.
- Improved surface finishing – in some applications, such as motorsport, bearing surfaces for example are superfinished, to give a much lower surface roughness than normal engineering surfaces. However, the cost of improved surface finishing will add to the manufacturing cost of a machine, so may not be cost-effective for mass production.
- Hard coatings, either ceramic or based on diamond like coatings (DLC) are increasingly being used for high pressure contacts, such as valve tappets (in passenger cars), to reduce both friction and wear.
- In addition, interesting research is ongoing into the possibility of superlubricity [26-28], whereby the friction coefficient between moving surfaces could potentially be reduced by a factor of ten, or more. Such research is being pursued in tribology laboratories around the world, but has not yet been widely commercialised.

Conclusions

In conclusion, the improved prediction of mixed/boundary friction is becoming of increasing importance as energy efficiency of machines is being improved and lower viscosity lubricants are being used.

There is recent experimental data that suggests commonly used models significantly underestimate the amount of mixed/boundary friction in machines, and a new, simple-to-use, model has been proposed which is much better agreement with experimental data.

The cost to overcome mixed/boundary friction has been quantified for passenger car engines, and worldwide, it has been estimated that, annually, at least $100 billion is spent (mainly on energy) to overcome mixed/boundary friction, and that the associated CO_2 emissions are at least 150 million tonnes.

It would be very useful to extend this analysis to other sectors, such as heavy-duty transport (including mining vehicles, shipping, trains, buses, and heavy duty freight) and also to industry. Some studies have looked at the overall impact of tribology on these various sectors [23,24] but have not explicitly reported results for mixed/boundary friction separately.

It is anticipated that tribologists and lubrication engineers will be researching ways to reduce mixed/boundary friction in machines for many years to come, and significant reductions in costs and CO_2 emissions could potentially be possible if superlubricity research currently ongoing in laboratories can be more widely commercialied.

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