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Lubrication, Tribology & Motorsport

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ABSTRACT

We review some of the key tribological issues of relevance to motorsport applications. Tribology is the science of friction and wear, and in a high performance engine, friction and wear are controlled by good component design (e.g. the engine and the transmission) and also by the use of high performance lubricants with the correct physical (and chemical) properties, *matched to the machine they are used in*. In other words, design of a specific lubricant for specific hardware can lead to optimised performance. (Tribology is also important in the tire-road contact but is not considered here.)

The importance of key physical properties of a lubricant is demonstrated with an emphasis on how the choice of the correct lubricant can help to minimize engine friction (and thus increase available power output) whilst protecting against engine wear. Key lubricant parameters discussed in the paper are the viscosity variation of a lubricant with temperature, shear rate and pressure. The relevance of these parameters in controlling oil film thickness and friction in bearings, the piston assembly, and the valve train is demonstrated with examples showing how a high performance engine differs from more conventional engines.

INTRODUCTION

The variation of lubricant viscosity with temperature, shear rate and pressure is crucial to how the lubricant performs in an engine. Unfortunately, these effects are typically ignored in models that are used to design engine components. This paper describes the important ways in which lubricant viscosity may vary, and then demonstrates, by using models which take into account these lubricant properties, how the lubricant can have a large impact on both the minimum oil film thickness in key engine components, and on the friction loss of the engine. A conventional engine is compared with a high performance engine, and it is shown that by matching the correct lubricant to the engine, improved performance may be achieved.

KEY LUBRICANT PARAMETERS THAT AFFECT PERFORMANCE

The most significant physical property of a lubricant is viscosity. Lubricant viscosity is strongly dependent on temperature, shear rate, and pressure. These effects are often neglected in many simulations of engine components (it is often assumed that the viscosity of the lubricant is constant at the temperature of interest). However, if a realistic assessment of friction and oil film thickness in the contact is required, these effects need to be taken into account. Figure 1 shows the typical way in which lubricant viscosity varies with temperature and shear rate.

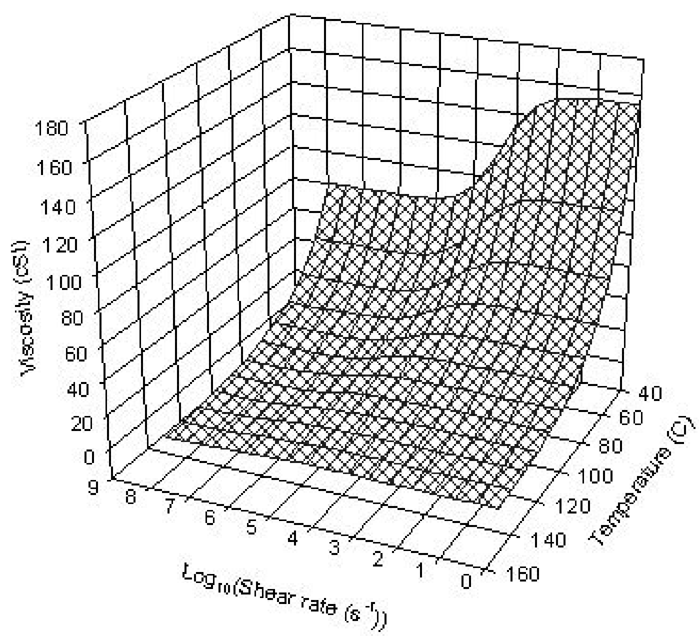

Figure 1: Variation of viscosity with shear rate for a SAE-10W/50 lubricant

Table 1 shows how the viscosity of different SAE viscosity grades differs.

SAE grade	V_k40 (cSt)	V_k100 (cSt)	Vd (mPa.s) at -20°C
20W/50	144.8	17.8	10,200
15W/40	114.3	14.9	4,800
10W/30	72.3	10.8	3,100
5W/30	57.4	9.9	1,900
0W/20	44.4	8.3	1,100
30	91.3	10.8	6,800

Table 1: Typical viscosities of common SAE grades

The variation of viscosity with temperature is quite accurately represented by the Vogel equation[1]:

$$\eta = \kappa \cdot \exp\left(\frac{\theta_1}{\theta_2 + T}\right)$$

where η is the viscosity (mPa.s) at temperature T (°C), and κ (mPa.s), θ_1(°C) and θ_2(°C) are constants.

A simple expression, the Cross equation[2], is useful for describing the variation of viscosity with shear rate:

$$\eta = \eta_\infty + \frac{\eta_o - \eta_\infty}{1 + \dfrac{\gamma}{\gamma_c}}$$

where η is the viscosity (mPa.s) at shear rate γ (s^{-1}), η_o is the viscosity (mPa.s) at zero shear rate and η_∞ is the viscosity (mPa.s) at infinitely high shear rate. γ_c (s^{-1}) is the shear rate at which the viscosity lies precisely halfway between η_o and η_∞. It is a good approximation to assume that η_∞/η_o is a constant, independent of temperature[3]. For the above expression to apply to realistic lubricants, it is found that we have to make γ_c dependent on temperature. A good description of the required temperature dependence is:

$$\log_{10} \gamma_c = A + B \cdot T$$

where A and B (°C^{-1}) are constants for a given lubricant.

The variation of viscosity with pressure is simply described by the Barus equation:

$$\eta(P) = \eta(0) \cdot \exp(\alpha P)$$

However, for this simple expression to apply to real lubricants, α must be assumed to depend on both temperature and pressure.

Table 2 summarises values of α for various different base fluids, as originally reported by Larsson[4]. Similarly, Table 3 reports results from Infineum for base fluids[5]. Generally it may be assumed that α decreases with both temperature and pressure. This has quite dramatic consequences for the prediction of friction losses in elastohydrodynamically (EHD) lubricated contacts. Alternative expressions for the variation of lubricant viscosity with pressure and temperature include the Roelands' equation[6] and the Sorab-van Arsdale equations[7].

Base oil	α(GPa^{-1}), p=0, T=20°C)	α(GPa^{-1}), p=0, T=80°C)	α(GPa^{-1}), p=400 MPa, T=20°C)	α(GPa^{-1}), p=400 MPa, T=80°C)
Naphthenic	40	19	35	16
Paraffinic	26	18	19	14
PAO A	21	14	15	11
PAO B	22	15	16	11
Polyglycol	24	15	18	11

Table 2: Values of α (GPa^{-1}) at different temperatures and pressures for different base fluids, according to Larsson

Base stock	α (GPa^{-1}) at 60°C	α (GPa^{-1}) at 80°C	α (GPa^{-1}) at 100°C
PAO 4	10.6	9.9	9.2
PAO 6	11.9	10.5	9.1
Group III, 4 cSt	11.9	10.9	9.8
Group III, 5 cSt	13.4	11.7	10.0
Group I, 100N	15.5	13.6	11.7
Group I, 150N	16.7	15.2	13.7

Table 3: Values of α (GPa^{-1}) at different temperatures for a range of basestocks, according to Infineum

Simply using the correct value of α (i.e. the value of α corresponding to the temperature and pressure in the contact) may still lead to errors in the prediction of friction losses in an EHD contact. This is because it is thought that lubricants have a limiting shear stress which cannot be exceeded[8-10]. In other words, there is a maximum friction force (per unit area) of the EHD contact, which occurs when the limiting shear stress is reached. If the Barus equation is used, then at high enough pressures, the viscosity will be high enough that the friction force would exceed this maximum value. In such circumstances, the Barus equation will give friction predictions that are too high, and estimates based on the limiting shear stress of the lubricant should be used.

We should finally mention that lubricants are not simple fluids. They are dilute polymer solutions, typically consisting of around 80% base oil (typically with molecular weights in the range 300-600), and 20%

additives. Additives such as Viscosity Modifiers are very large polymers with molecular weights of the order of 10,000 or greater. These additives can impart some elastic properties to the lubricant. It is thought that under certain conditions, these elastic properties can be beneficial in reducing both wear and friction in engine bearings[11-14]. These effects cannot be predicted from Reynolds' equation, since Reynolds' equation does not allow for viscoelastic behaviour.

Needless to say, a lubricant that is suitable for motorsports applications must have a viscosity that separates all the key moving surfaces in the engine, whilst at the same time minimizing total engine friction.

In addition, to these physical requirements on the oil, the lubricant must also have good anti-foaming properties, because any air that can be entrained in the oil can cause problems in hydraulically actuated valves, and may also cause cavitation damage in bearings.

COMPARISON OF HIGH PERFORMANCE AND CONVENTIONAL ENGINE PARAMETERS

There are clearly many differences between a conventional gasoline engine and a high performance engine. For most conventional European gasoline engines, the maximum rpm is around 7500 rpm, typical mid-range displacements are around 2.0 litres, with a bore-stroke ratio close to 1. The most common engine configuration would be an in-line 4 cylinder engine, often with 4 valves per cylinder.

In comparison, a Formula 1 engine has a displacement of 3 litres, with a V10 engine configuration, a bore-stroke ratio in the range 2.0-2.5, and a maximum rpm of 17000-18000 (these latter figures were reported by Wright[15]). Table 4 shows a comparison of some of the key engine variables (based on the above figures) for a typical conventional European engine and a Formula 1 engine.

In Table 4, the parameters for the Formula 1 engine were estimated as follows :

Since the displacement per cylinder is 0.3 litres, and we know that is the area swept out by the cylinder multiplied by the stroke, then if we know that the bore-stroke ratio is in the range 2-2.5, we can calculate the range within which the bore diameter and the stroke must lie. For the conventional engine, we know that the maximum angle that the con-rod makes with the vertical is around 15°. By assuming that the maximum angle that the con-rod makes with the vertical for the Formula 1 engine lies in the range 10-15°, we obtained a likely range for the con-rod length (however, the precise value of the con-rod length is not essential for carrying out preliminary sensitivity studies into the tribology of the key engine components).

Note that for both types of engine, the maximum linear speed is roughly the same.

	Formula 1 engine	European 2.0 litre engine
Engine type	V10	Inline 4
Displacement (litres)	3.0	2.0
Displacement per cylinder (litres)	0.3	0.5
Bore diameter (mm)	91.4-98.5	89.8
Stroke (mm)	39.4-45.7	78.7
Bore-Stroke ratio	2.0-2.5	1.14
Con-rod length (mm)	76-132	154.0
Max piston speed (m/s)	37-48 at max rpm	32 at 7500 rpm
Max rpm	17000-18000	7500

Table 4: Key engine parameters for a Formula 1 engine and a conventional European engine

Note also that if a higher maximum rpm is desired, then the stroke would need to be reduced, and since the displacement per cylinder is constant at 0.3 litres, the bore diameter would have to be increased. There is clearly a limit to how far this can go, since the larger the bore-stroke ratio, the more sensitive the piston will be to secondary motion (piston tilt). Note also that a racing piston will differ substantially from a conventional piston in that the piston skirt area will be much smaller (so that friction is reduced), again making the racing piston more sensitive to secondary motion

PISTON ASSEMBLY TRIBOLOGY

The calculation of oil film thickness under the piston rings involves solving Reynolds' equation, using the appropriate piston ring profile, and taking into account the variable speed of the piston ring as the piston moves from bottom dead center to top dead center. It is also necessary to know the gas pressures on either side of the piston ring, the piston ring temperature, and the liner temperature at the piston ring position (so that the lubricant viscosity may be estimated). If all these parameters are known, then the oil film thickness and friction loss of the piston ring may be calculated[17-22]. A further complication then arises since there is interaction between the various rings in the piston assembly. In general, the oil control ring is designed to restrict the oil supply to the upper rings. In effect, this means that the upper piston rings are often “starved” of oil (this is particularly true at mid-stroke positions). Therefore, some further analysis is required to determine when the rings are starved, and to recalculate the oil film thickness and friction accordingly. It is usually found that the effect of oil starvation is to reduce the oil film thickness on the liner, and to increase the friction loss of the ring-pack.

In a high performance engine, when there are often only two piston rings (the oil control ring and the top ring) a compromise must be made. If the oil control ring tension is too low, then too much oil will be available to the top ring, and although friction may be reduced, oil consumption may be too high. On the other hand, if the oil control ring tension is too high, oil consumption may be reduced, but there will be excessive oil starvation for the top ring, which would lead to higher friction (and possibly high wear – leading to failure). Clearly, bore distortion will also play a part in controlling oil consumption and friction loss.

For a conventional engine, there are usually three piston rings, and oil consumption can be controlled more effectively.

Figure 2 shows a typical combustion chamber pressure curve for a conventional European 2.0 litre gasoline engine.

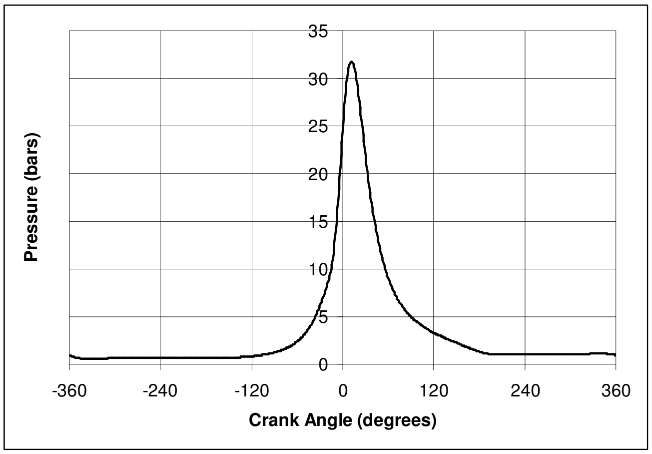

Figure 2: Combustion chamber pressure versus crank angle for a conventional European 2.0 litre gasoline engine operating at 2500 rpm and medium load

Note that the brake mean effective pressure (BMEP) corresponding to Figure 2 is approximately 4 bars.

For this engine, it is assumed that the top piston ring is 1.5 mm in height, with a symmetric, parabolic ring profile, with a ring tension of 200 kPa. Under these conditions, Figure 3 shows the predicted oil film thickness versus crank angle, assuming fully flooded conditions, for a speed of 7500 rpm, for an SAE-15W/40 lubricant, for different ring profile radii of curvature.

Figure 3 show that piston rings that have a larger radius of curvature (i.e. are flatter) have a larger oil film thickness in those parts of the engine cycle where the "squeeze" effect dominates (e.g. around dead center positions) and smaller oil film thicknesses in those parts of the engine cycle where the "wedge" effect dominates (e.g. in mid-strokes), compared to a more curved piston ring. Clearly, the piston ring shape also has an impact on the ring friction. Table 5 shows the effect that the ring radius of curvature has on oil film thickness and friction loss.

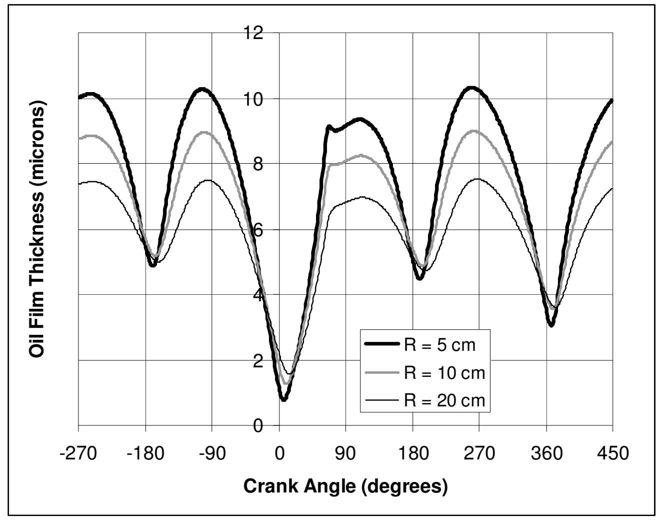

Figure 3: Oil film thickness versus crank angle for the top piston ring of a conventional European 2.0 litre gasoline engine, at 7500 rpm, assuming fully flooded lubrication

	R=5 cm	R=10 cm	R=20 cm
OFT (μm) at -270°	10.01	8.76	7.38
OFT (μm) at -180°	5.51	5.73	5.46
OFT (μm) at -90°	10.08	8.85	7.46
OFT (μm) at 0°	1.17	1.84	2.23
OFT (μm) at 90°	9.15	8.07	6.81
OFT (μm) at 180°	5.09	5.42	5.17
OFT (μm) at 270°	10.22	8.93	7.51
OFT (μm) at 360°	3.55	4.00	4.02
Friction Loss (W)	109.3	137.0	171.3

Table 5: Effect of ring radius of curvature on oil film thickness and friction. Lubricant is assumed to be SAE-15W/40

We can carry out a similar analysis for a Formula 1 engine top ring. For such an engine, we assume that the top ring has a height of 1mm, and is a symmetric parabolic ring. We assume a ring tension of 1000 kPa. Figure 4 shows the predicted oil film thickness versus crank angle for different ring radii of curvature. (For these simulations we assumed an engine speed of 18000 rpm, a bore diameter of 94.7 mm, a stroke of 42.6 mm and a con-rod length of 104 mm – these are the mid values of the parameters estimated in Table 4. We also assumed that the lubricant was an SAE-15W/40 grade, with a bottom dead center liner temperature of 100°C and a top dead center liner temperature of 150°C. In addition we assumed a peak combustion chamber pressure of around 110 bars – the BMEP of the conventional engine was 4 bars, with a peak combustion chamber pressure of just over 30 bars, and Wright reports that the BMEP of a modern Formula 1 engine is between 13 and 14 bar. Therefore we have scaled the combustion chamber pressure accordingly).

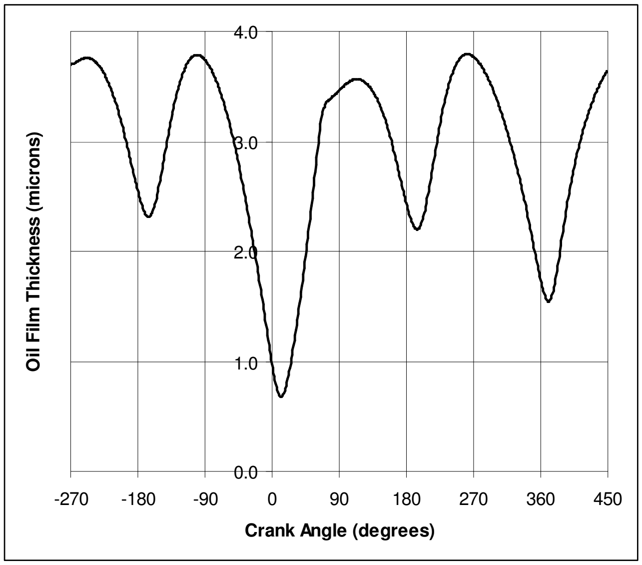

Figure 4: Estimated oil film thickness under the top ring of a Formula 1 engine, at 18000 rpm, assuming fully flooded conditions

The friction loss for the Formula 1 top piston ring, under fully flooded conditions, at 18000 rpm, was predicted to be 300 W for a ring radius of curvature of 5 cm, 375 W for the ring with a radius of 10 cm, and 470 W for the ring with a radius of 20 cm.

In practice we know that the interaction of the oil control ring and the top ring leads to lower top ring oil film thicknesses at mid-stroke, and correspondingly higher friction losses. Figure 5 shows the effect of oil starvation on predicted top ring oil film thickness for the conventional European 2.0 litre gasoline engine at 7500 rpm. Also in this complete ring-pack simulation, the squeeze effect has been neglected, although this is not expected to impact significantly on the predicted friction loss (since the squeeze effect dominates at dead centers where the piston speed is low). For the conventional engine, the effect of oil starvation is to increase the predicted ring frictional loss from approx 110 W to 258 W. Although the top piston ring is starved of oil, the assumption that the oil control ring is fully flooded, is considered to be fairly reliable. For a conventional engine, the friction mean effective pressure (FMEP) of the top piston ring, under starved conditions, is estimated to be around 8.2 kPa, and the FMEP of the oil control ring is estimated to be approximately 6.4 kPa. In addition, the total ring power loss is thought to be around 60% of the total friction loss of the piston[23], with the remainder coming from piston skirt friction. Therefore, for a conventional engine, we would expect an FMEP in the range 25-30 kPa. If we assume similar numbers for the FMEP of a Formula 1 piston assembly, then a ball-park figure for the total piston assembly friction loss would be 15 kW. If the piston ring tensions used in the Formula 1 engine are far higher than those used in conventional engines, then this estimate will be too low.

Using the friction estimates for the Formula 1 ring pack, (from the calculations for Figure 4), and taking into account the effects of starvation in a similar way to that for the conventional engine, the estimated top ring friction is 880 W (assuming a ring radius of curvature of 10 cm). If the oil control ring has a similar friction loss, then the total ring loss is 1.76 kW. If we then assume that the ring friction is 60% of the total piston assembly friction, then the total friction for one cylinder is approximately 3 kW, so that the total piston assembly friction is equal to around 30 kW. This equates to an FMEP of approximately 70 kPa.

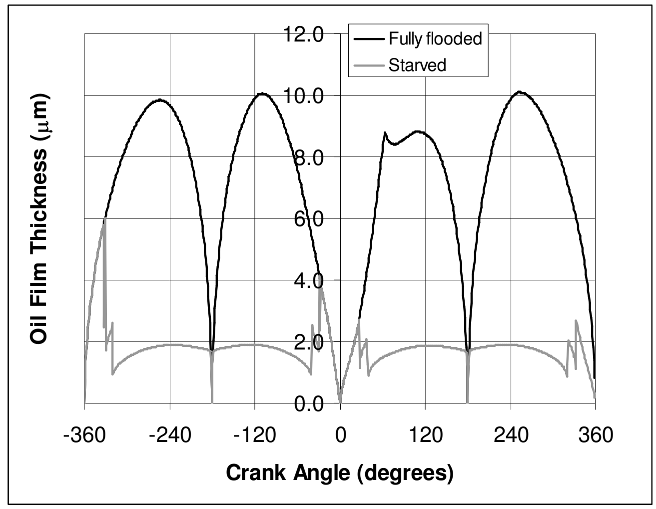

Figure 5: Predicted oil film thickness under the top ring of a conventional European engine, operating at 7500 rpm with an SAE-15W/40 lubricant

Finally, we discuss the effect of lubricant viscometry on piston assembly friction. For the Formula 1 engine, using the same data as for Figure 4, and assuming a top ring radius of curvature of 10 cm, Table 6 summarises the predicted top ring friction power loss, and the predicted minimum oil film thickness, for different lubricants

Lubricant	Power loss (W)	Minimum oil film thickness (μm)
SAE-20W/50	402	0.76
SAE-15W/40	375	0.73
SAE-10W/30	313	0.62
SAE-0W/20	259	0.51

Table 6: Effect of lubricant viscosity grade on top ring power loss and minimum oil film thickness (assuming fully flooded conditions)

Clearly, Table 6 shows that reductions in piston ring friction can be achieved by moving to a lower viscosity lubricant. However, this is at the expense of lower minimum oil film thicknesses. There is a trade-off between reduced friction (and greater power available to the wheels) and engine durability. In Table 6, we can see that moving from an SAE-15W/40 lubricant to an SAE-

0W/20 lubricant will lead to a 30% decrease in top ring minimum oil film thickness, and will also give a decrease in top ring friction of approximately 30%. The benefits of moving to lower viscosity lubricants to obtain better fuel efficiency via lower engine friction have been well documented[24-29].

BEARING TRIBOLOGY

Con-rod bearing loads may be estimated by knowing the combustion chamber pressure and the engine speed. For low speed, conventional engines, the con-rod bearing load is often dominated by the combustion gas pressure. However, as engine speed increases, inertial effects begin to dominate.

For the main bearing, the load needs to be calculated from the adjacent con-rod bearing loads, and in particular, for the Formula 1 engine it would be necessary to know the angle of the “vee” in the V10 configuration.

In the absence of data for Formula 1 con-rod and main bearings, we try to illustrate how lubricant viscosity can influence oil film thickness and friction by analyzing bearings from a conventional engine, the Mercedes-Benz M111 2.0 litre gasoline engine. Figure 6 shows the calculated con-rod load of the M111 engine at two speeds, 2500 rpm and 7500 rpm. At the lower speed the con-rod load is dominated by the gas pressure in the combustion chamber, whereas at the higher speed, inertial effects clearly start to become more important.

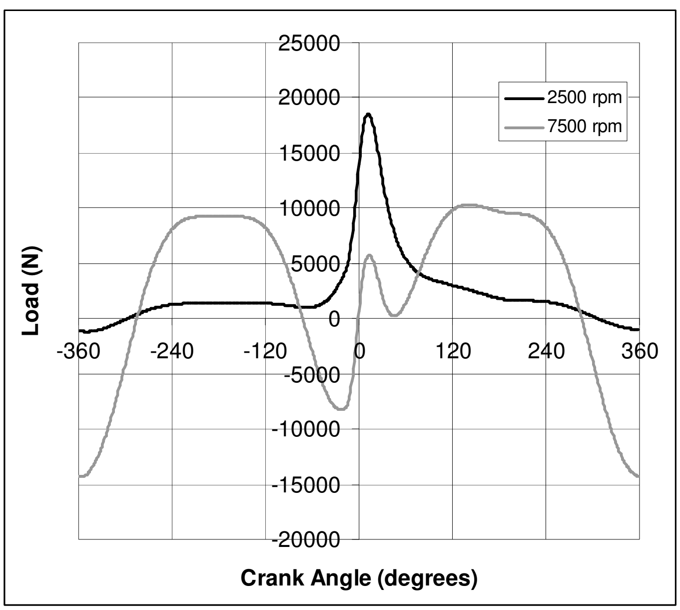

Figure 6: Con-rod bearing loads for conventional European 2.0 litre gasoline engine, at 2500 rpm and 7500 rpm

Figure 7 shows the predicted oil film thickness for both load curves using a modified version of the Short Bearing Approximation, which allows for lubricant shear thinning[30]. The “squeeze” effect is also included. (The standard Short Bearing Approximation often assumes that it is only the “wedge” effect that is important in the lubrication of engine bearings). The simulations were carried out for an SAE-15W/40 lubricant. Two important conclusions can be drawn from the results. Firstly, for bearing loads that are dominated by gas pressure, the minimum oil film thickness in the bearing will occur at a position corresponding to the peak combustion chamber pressure. For inertially dominated loads, the minimum oil film thickness position will occur elsewhere, and there may be two or three positions around the bearing where the oil film thickness is low. In the calculations carried out above, the predicted oil film thicknesses for the higher speed condition were greater than those for the lower speed condition because the oil temperature was assumed to be the same in both simulations. In practice, it is likely that the oil temperature at the higher speed condition is likely to be higher, so the minimum oil film thickness is likely, in practice, to be lower than the values shown in Figure 7.

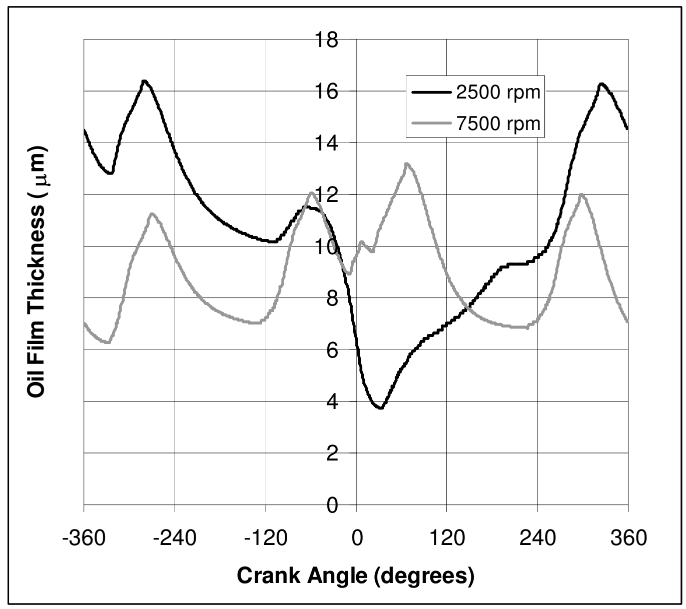

Figure 7: Predicted con-rod bearing oil film thickness based on load data from Figure 6.

In addition to oil film thickness, it is also important to know what flow rate of oil is required to lubricate the bearings. If the flow rate is not sufficient, lubricant starvation can occur which may lead to catastrophic damage to the bearings. Figure 8 shows the flow rates predicted by the modified Short Bearing Approximation.

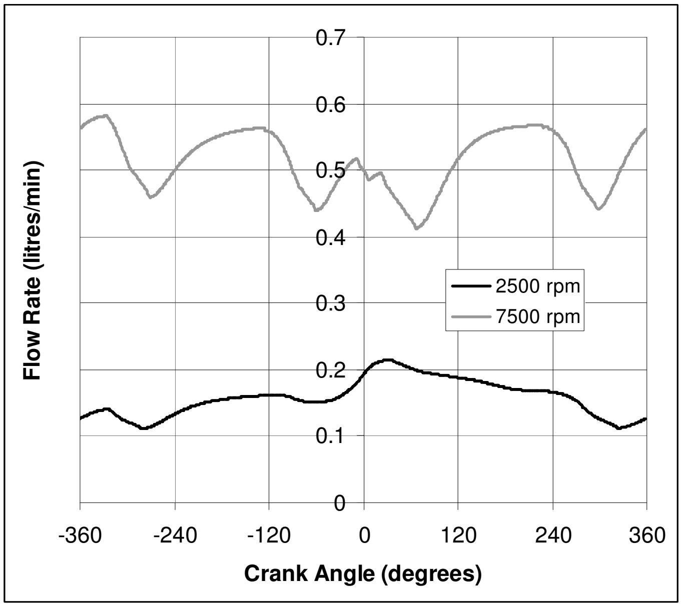

Figure 8: Flow rate (litres/min) required to avoid lubricant starvation of con-rod bearing, based on load data of Figure 6

Table 7 summarises the results for minimum oil film thickness, friction loss, and average flow rate, for different lubricants (an SAE-20W/50, an SAE-15W/40, an SAE-10W/30 and an SAE-0W/20) for the conventional European 2.0 litre gasoline engine at 2500 rpm, and Table 8 summarises the results at 7500 rpm.

Lubricant	Minimum OFT (μm)	Average Friction Power Loss (W)	Average Flow Rate (litres/min)
SAE-20W/50	4.13	73.5	0.152
SAE-15W/40	3.72	63.4	0.158
SAE-10W/30	3.01	46.7	0.171
SAE-0W/20	2.63	37.4	0.179

Table 7: Sensitivity of con-rod bearing results to lubricant viscosity grade at 2500 rpm

Lubricant	Minimum OFT (μm)	Average Friction Power Loss (W)	Average Flow Rate (litres/min)
SAE-20W/50	6.89	671.5	0.496
SAE-15W/40	6.27	577.7	0.514
SAE-10W/30	5.15	425.1	0.548
SAE-0W/20	4.55	348.4	0.568

Table 8: Sensitivity of con-rod bearing results to lubricant viscosity grade at 7500 rpm

It is recognized that the model used here is too simplistic for use as a detailed design tool of bearings. A useful model would need to take into account differences in temperature around the bearing, and also bearing deformation. The lubricant viscosity may be expected to change around a "real" bearing for a number of reasons. The viscosity will vary because (i) the temperature of the oil changes around the bearing (cool at the inlet oil hole, hotter as it shears during flow around the bearing), (ii) the shear rate varies around the bearing, and (iii) the pressure varies around the bearing. Clearly models that assume the viscosity of the lubricant is constant around the bearing will have limited usefulness.

Although the short bearing approximation has it's limitations, it is useful for identifying trends. In particular, the model can be used to predict the variation of key parameters such as the minimum oil film thickness, h_{min} (μm), the friction power loss of the bearing, F_B(W), the maximum pressure in the bearing, P_{MAX} (MPa), and the lubricant flow rate, Q, required to supply the bearing :

$$h_{\min} \propto \sqrt{\frac{\eta \omega R L^3}{W}}$$

$$F_B \propto \frac{\eta^{0.75} \omega^{1.75} L^{0.25} W^{0.25} R^{2.75}}{c^{0.5}}$$

$$P_{MAX} \propto \frac{c^{0.5} W^{1.25}}{\eta^{0.25} \omega^{0.25} L^{1.75} R^{1.25}}$$

$$Q \propto \omega R L c$$

where η is the lubricant viscosity, R is the bearing radius, L is the bearing length, c is the radial clearance, ω is the angular speed of the bearing, and W is the load on the bearing. Note that the expressions above are only valid if the minimum oil film thickness is small compared to the bearing radial clearance. The derivation of the above expressions is detailed in Appendix 1.

Although a Formula 1 engine will have a much greater engine speed than a conventional engine, the bearings will be proportionately smaller. If we simply assume that all the bearings dimensions of a Formula 1 engine are 0.8 times those of a conventional engine (although the displacement per cylinder for the Formula 1 engine is 0.3 litres, whereas that for the conventional engine we are considering is 0.5 litres, the loads in a Formula 1 engine are far greater – therefore we use 0.8 rather than 0.6), then for a given viscosity, we can estimate the power loss of a typical Formula 1 bearing by comparing with the results on a conventional bearing. For the con-rod bearing considered in Table 8, for an SAE-15W/40 lubricant, the conventional bearing had a power loss of approximately 580 W at a speed of 7500 rpm. Therefore for the same lubricant, assuming that the temperature of the oil is the same, the Formula 1 con-rod bearing power loss would be expected to be a factor of $(17500/7500)^{1.75} * 0.8^3$ higher. This means that a ball-park estimate of the con-rod bearing friction is 1.3 kW. Since there are 10 con-rod bearings, a Formula 1 engine may be expected to have a power loss of 13 kW from these bearings. There are 6 main bearings for the V10 engine. These are probably of larger dimension than the con-rod bearings. So we could assume that each main bearing may contribute 2 kW. Therefore a ball-park figure of the engine bearing power loss is 25 kW. This figure may have to be reduced if oil temperatures are substantially higher than for the conventional engine. (It should also be noted that the peak pressure in the Formula 1 con-rod bearing is expected to be approximately 1.6 times higher than the peak pressure in the conventional con-rod bearing, ignoring any change in the bearing load).

The simple equations above are very approximate, and should be used for identifying trends only. The expressions are not recommended for use in the design of bearings. For the detailed design of bearings, based on rigid bearing models, there are many fast, robust methods available[31-34].

For high pressures (those greater than about 50 MPa) it is likely that a rigid bearing model will start to become inaccurate. The reason is that the pressures become so high that elastohydrodynamic lubrication occurs, and it is necessary to solve both the Reynolds' equation and the equation that describes the elastic deformation of the bearing surfaces. This is generally a time consuming calculation that requires a numerical treatment either by the finite element or finite difference techniques[35-38].

The paper on the elastohydrodynamic lubrication of journal bearings by Fenner et al[36] is particularly illuminating. The authors find that under high loads, there is a central portion of the bearing where the oil film thickness is essentially constant, and where the pressure distribution is effectively that of a Hertzian contact. In essence, once the pressure in the bearing exceeds a certain value (in the case of the bearing analysed by Fenner et al this pressure was 200 MPa), it effectively became constant. Therefore the main conclusion was that the peak pressure in the con-rod bearing was much lower than would have been predicted from a rigid bearing model (the rigid bearing model predicted a peak pressure of 1200 MPa, whereas the model which took elastic deformation into account predicted a peak pressure of only 200 MPa). The effects of elastic deformation on friction were not reported.

Concluding this section, it is suggested that although there are many sophisticated models of journal bearings in widespread use, not many take full account of the way in which the lubricant viscosity may vary around the bearing. The variation of lubricant viscosity around the bearing, due to temperature variations, due to the changing shear rate, and due to the pressure variation around the bearing, are expected to have a significant effect on predictions of oil film thickness, and friction, for both hydrodynamic and elastohydrodynamic lubrication conditions.

VALVE TRAIN TRIBOLOGY

Lubrication in the valve train differs from that in the bearings and the piston assembly, since it is generally accepted that the valve train is in the mixed or boundary regime for most of its operating conditions. This is certainly true as far as conventional engines are concerned. Measurements of valve train friction torque frequently show that, under normal operating conditions, the valve train friction torque increases as the viscosity of the oil decreases. It is also in the valve train that friction modifier additives are thought to have the largest effect.

In a valve train lubrication model, the kinematics of the cam-follower system need to described, and from knowing the cam and follower profiles, the oil film thickness may be calculated from elastohydrodynamic lubrication theory[1]. Friction losses may then also be calculated.

Total friction torque in valve train is Γ_{TOTAL}, where an approximate expression is :

$$\Gamma_{TOTAL} = \Gamma_B + C\eta\omega$$

where Γ_B is the friction torque due to boundary friction, and the second term represents hydrodynamic friction in the valve train system (both from the cam-follower contact and from the camshaft bearings). In the term for hydrodynamic friction, C is a constant, η is the lubricant viscosity (at the appropriate temperature) and ω is the camshaft angular speed.

The boundary friction torque may be calculated by estimating the oil film thickness using elastohydrodynamic theory, and then comparing this value to the combined surface roughness of cam and follower, and assigning an effective friction coefficient. This approach has been used successfully by researchers from Shell[39], Ford[40] and Nissan[41], and is found to give good agreement with experimental measurements.

A lower viscosity lubricant will give a higher boundary friction torque, whereas the hydrodynamic friction torque will be lower. Friction modifier additives can be added to the lubricant, and these are effective at reducing the boundary friction torque.

Figure 9 shows the valve lift curve for a typical European 2.0 litre gasoline engine (with a direct acting bucket tappet valve train system).

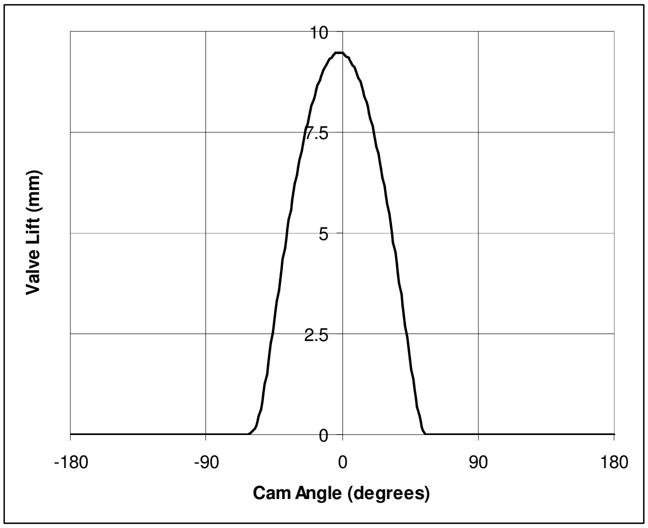

Figure 9: Valve lift for typical European 2.0 litre gasoline engine

Figure 10 shows the predicted oil film thickness for this engine, at an engine speed of 2500 rpm, assuming an SAE-15W/40 lubricant, with a temperature of 100°C.

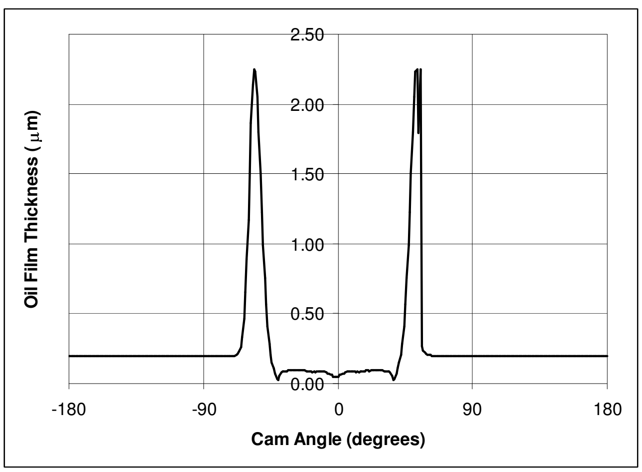

Figure 10: Predicted valve train oil film thickness in conventional European 2.0 litre gasoline engine

Figure 11 shows the predicted friction torque for this valve train.

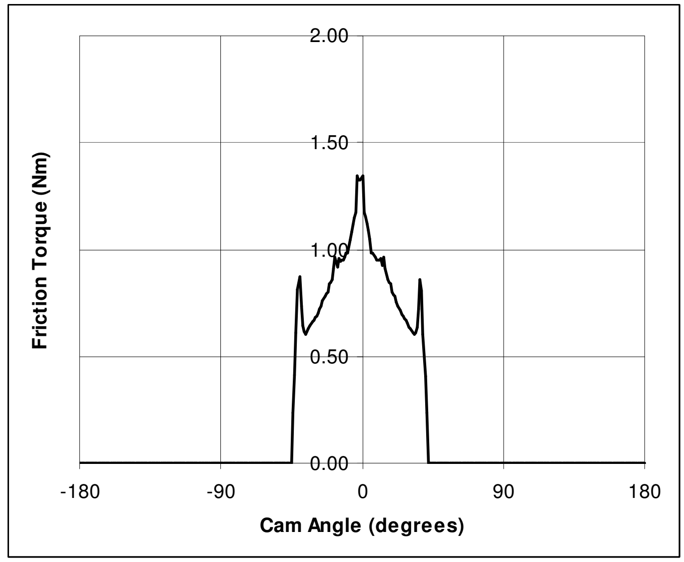

Figure 11: Predicted boundary friction torque for one cam of the conventional European 2.0 litre gasoline engine

For the conventional European 2.0 litre gasoline engine, at 2500 rpm, the valve train friction power loss will be approximately 599.8 W (of which 481.8 W is due to boundary friction. Each cam has an approximate power loss of 37.5 W, of which 30 W is due to boundary friction).

For a high performance engine, the valve lift curve will differ, the spring loads are likely to be far higher, and the components making up the valve train will be lighter.

TOTAL ENGINE FRICTION

By counting up the total number of pistons in an engine, the total number of con-rod and main bearings, and the total number of cams, total friction for the engine can be calculated, given the engine operating conditions, and the type of lubricant used.

For the typical European engine, Figure 12 details the engine friction breakdown at idling, under urban conditions (2500 rpm, medium load), and under motorway conditions (7500 rpm ,high load). These calculations were performed assuming a standard SAE-15W/40 oil. Table 9 shows how the results differ when different lubricants are used in the engine (for urban conditions).

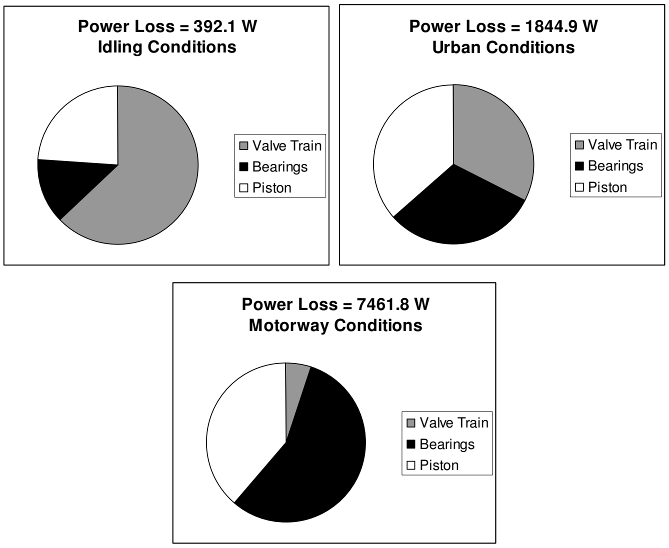

Figure 12: Engine friction breakdown for various operating conditions for European 2.0 litre engine

Oil grade	Valve train	Bearings	Piston assembly	Total
SAE-20W/50	588.5	610.1	743.9	1942.5
SAE-15W/40	599.8	567.5	677.6	1844.9
SAE-10W/30	625.4	467.5	554.2	1647.1
SAE-0W/20*	410.2	380.0	492.2	1282.4

Table 9: Sensitivity of engine and component friction to lubricant viscosity grade, for European 2.0 litre engine under urban operating conditions (* Note that the SAE-0W/20 oil is assumed to be friction modified)

In Table 9, it can be seen that reducing the lubricant viscosity is effective at reducing friction in the bearings and piston assembly. However, reducing the lubricant viscosity causes an increase in the valve train friction. The reason why the SAE-0W/20 oil has a sharp decrease in valve train friction compared to the other oils is that it is assumed that it contains a friction modifier (if it didn't contain a friction modifier the valve train friction power loss would have been 641.2 W).

Unfortunately, we do not have sufficient details of a Formula 1 valve trains system to enable us to calculate the total engine friction of a Formula 1 engine. However, we know that at the maximum engine speed, 18000 rpm, the valve train friction is likely to make a smaller contribution than at lower speeds. We estimated piston assembly losses of around 30 kW, and bearing friction losses of 25 kW. At 18000 rpm, we would expect valve train friction losses to be less than 10 kW (and that this would mainly be hydrodynamic friction losses in the valve train). Figure 13 summarises this very rough analysis for a Formula 1 engine at 18000 rpm.

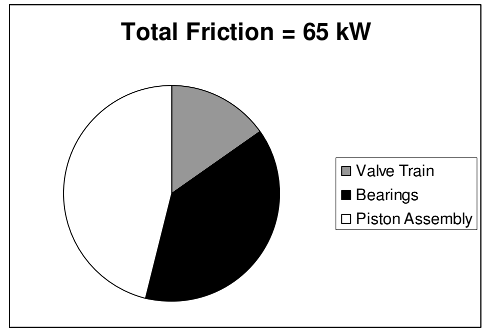

Figure 13: Approximate estimate of engine friction in a Formula 1 engine at 18000 rpm (valve train friction estimate is uncertain due to lack of information)

The strategy for minimizing engine friction in a Formula 1 engine is fairly simple. If the engine has a lot of boundary friction, you would use a higher viscosity lubricant, with a friction modifier, whereas if the engine has less boundary friction, you would use a lower viscosity lubricant (again with a friction modifier). By looking at engine friction results for the conventional engine, it is possible to decrease engine friction by 10 or 20% by using the correct lubricant. There are a number of papers[41-47] which the reader can refer to for more information on total engine friction calculations.

To optimize power output from a high performance engine, it is necessary to choose a lubricant which gives the lowest possible friction. This entails choosing the lubricant viscosity which gives the lowest friction over the range of operating conditions appropriate for the engine, and choosing an optimum friction modifier for reducing friction in boundary lubricated contacts.

CONCLUSION

In this paper I have attempted to demonstrate that the physical properties of a lubricant are extremely important in determining both the minimum oil film thickness in key lubricated contacts, and the associated friction losses. It is important to use models for the piston assembly, the engine bearings, and the valve train, that include accurate information about how the lubricant viscosity varies with temperature, shear rate, and pressure.

We have compared a conventional gasoline engine with a high performance Formula 1 engine, and shown that matching the lubricant to the engine can lead to significant reductions in engine friction, with consequent improvements in the available power to the wheels.

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APPENDIX 1

In this Appendix we derive the relationships quoted in the paper for oil film thickness, friction power loss, maximum pressure, and lubricant flow rate, from the Short Bearing Approximation.

In the standard Short Bearing Approximation (see for example reference A1), we often find the following relationship between load W and eccentricity ratio ε:

$$W = \frac{\eta\omega\varepsilon RL^3}{c^2(1-\varepsilon^2)^2}\frac{\pi}{4}\sqrt{\left(\frac{16}{\pi^2}-1\right)\varepsilon^2+1}$$

…(A1.1)

where W is the load (N), η is the lubricant viscosity (Pa.s), ω is the angular speed (rad/s), R is the bearing radius (m), L is the bearing length (m), c is the radial clearance (m), and ε is the eccentricity ratio. The eccentricity ratio is defined as:

$$\varepsilon = \left(1-\frac{h_{\min}}{c}\right)$$

…(A1.2)

where h_{min} is the minimum oil film thickness in the bearing.

Frequently, equation A1.1 is solved numerically. Although this is efficient and fast, the numerical solution has the disadvantage that it does not explicitly reveal how the minimum oil film thickness in the bearing varies with the key variables (W, η, ω, L, R, c). In this Appendix we start with equation A1.1 and derive analytical expressions for the minimum oil film thickness, the friction power loss, the lubricant flow rate and the maximum pressure, in terms of the key problem variables. This expressions are valid when $h_{min} \ll c$. In this limit, ε ≈ 1, and (1-ε) = h_{min}/c. Applying these approximations to equation A1.1, we obtain the simplified equation below:

$$W \approx \frac{\eta\omega RL^3}{4{h_{\min}}^2}$$

...(A1.3)

and so:

$$h_{\min} \approx 0.5\left(\frac{\eta^{0.5}\omega^{0.5}R^{0.5}L^{1.5}}{W^{0.5}}\right)$$

...(A1.4)

We also know that the friction power loss, $F_B(W)$, is given, in the standard short bearing approximation by:

$$F_B = 2\pi \frac{\eta\omega^2 LR^3}{c(1-\varepsilon^2)^{0.5}}$$

...(A1.5)

By using the expression for hmin derived in A1.4, we find that F_B (W) is given by :

$$F_B \approx 2\pi \frac{\eta^{0.75}\omega^{1.75}L^{0.25}R^{2.75}W^{0.25}}{c^{0.5}}$$

…(A1.6)

We also know that in the Short Bearing Approximation, the lubricant flow rate required to avoid lubricant starvation is given by:

$$Q = \omega RLc\varepsilon$$

...(A1.7)

In the approximation we are using here, $\varepsilon \approx 1$, so:

$$Q \approx \omega RLc$$

...(A1.8)

Finally, we consider the maximum pressure in the bearing. The standard Short Bearing Approximation gives the following expression for the pressure, P, in the bearing:

$$P = \frac{3\omega\eta\varepsilon\sin\theta}{c^2(1+\varepsilon\cos\theta)^3}\left(\frac{L^2}{4} - y^2\right)$$

...(A1.9)

Clearly, the maximum pressure in this model will occur on the bearing center line, where y = 0. The maximum pressure will occur when the derivative of the above expression (with respect to θ) is equal to zero. Detailed calculations show that the angle at which this occurs, θ_m, is given by:

$$\cos\theta_m \approx -1 + \frac{1}{5}\frac{h_{\min}}{c} + O\left(\frac{h_{\min}}{c}\right)^2$$

...(A1.10)

and

$$\sin^2\theta_m \approx \frac{2}{5}\left(\frac{h_{\min}}{c}\right) + O\left(\frac{h_{\min}}{c}\right)^2$$

...(A1.11)

Therefore, since we are assuming that $h_{min} << c$, the expression for the maximum pressure in the bearing becomes:

$$P_{MAX} \approx \frac{125}{36\sqrt{5}}\frac{c^{0.5}W^{1.25}}{\omega^{0.25}\eta^{0.25}R^{1.25}L^{1.75}}$$

...(A1.12)

This completes the derivation of the relationships quoted in the section on bearing modeling.

As an example of the use of these relationships, consider the case where : L = 21 mm, R = 25 mm, c = 30 μm, η = 10 mPa.s, W = 20000 N, and assume the engine speed is 1700 rpm. A numerical solution of the Short Bearing Approximation gives : h_{min} = 2.22 μm; F_B = 125.5 W; P_{MAX} = 227.7 MPa. The relations derived in this Appendix give : h_{min} = 2.25 μm; F_B = 121.7 W; P_{MAX} = 229.2 MPa.

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