DISCUSSION

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I last published in this area in the early 60's (refs. [D1-D3]) in what may have been the first paper on non-Newtonian lubricants in journal bearings. I am impressed by how much progress has been made in the mathematical ability to solve such problems of increasing complexity, although the results are pretty much the same: the shear thinned oil performs as if it had a viscosity somewhere between the low shear value and the high shear (base stock) value. Our old work was at steady state only, whereas the present work includes dynamic loading, viscoelasticity and compressibility and probably could include pressure and temperature effects.

One interesting effect that we noted was that while there is only one local viscosity at every point in the lubricant film, defined by the resultant of the shear rates at that point, just as in the present paper, the effective apparent viscosity in the circumferential direction is lower than in the axial or side leakage direction. This shows up more dramatically with greases, which shear thin much more than lube oils [D3]. Perhaps this shows up in the factors called F_2 and F_3 in the present paper.

As shown in Fig. 4 of the paper, the constitutive equation chosen to relate viscosity to shear rate makes the transition from the high to the low viscosity more rapidly than the experimental data. Our measurements with polyisobutylene and methacrylate polymers showed that it took about four decades of shear stress for this transition, rather than the ~ 1.5 decades of Fig. 4. (Shear stress is a convenient parameter, because the degree of shear thinning of the polymer contribution to viscosity versus shear stress is virtually independent of temperature (4) and pressure (5).) Why not use the more accurate data as long as the computer program is so versatile? Along this same line I would urge the author to apply his mathematics to 5W-30 oils, which have a higher ratio of the low to the high shear viscosity than the ones shown in Fig. 4, and might show larger effects.

There are not many terms dealing with time effects and so I wonder whether the "squeeze film" effect has been simulated adequately or whether the calculations come closer to generating a succession of steady-state solutions. Has the mathematics ever been applied to the case of zero rotation speed to determine whether a reasonable "squeeze film" solution is obtained?

It seems to me that one of the difficulties in dynamically loaded bearing calculations is that there are configurations where a sudden change in the size and direction of the load could produce a large change in eccentricity.

In the case of non-Newtonian lubricants a large number of trial-and-error solutions would appear to be necessary to establish the correct eccentricity. The author has circumvented this problem by introducing an admittedly minor term having to do with the inertia of the journal. He then uses this term to calculate the eccentricity when the load changes. Is this not a case of the tail wagging the dog? Inclusion of this term may help the robustness of the solution, but is it also possible that it helps to damp out the occasional extreme changes in eccentricity that cause the metal-metal contact often observed? Figures 5(a) and 6(b), and 8 all show hefty minimum oil film thicknesses, for example.

In the end the bearing performs as if it were operating on a lubricant of lower viscosity than the low shear value. The power loss is reduced "despite the fact that the film thickness is reduced." There should be no surprise here. Reducing the film thickness under hydrodynamic conditions always gives a friction reduction, but at a sacrifice in the margin of safety against wear. In real bearings capacitance measurements show that when the film thickness falls to values within the r.m.s. roughness values of the surfaces, metallic contact is made. Most auto drivers would feel that the danger of this situation outweighs the benefit of the power loss reduction, which is small in journal bearings, in general.

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References

(1) Horowitz, H. H., and Steidler, F. E., 1960, "The Calculated Journal-Bearing Performance of Polymer-Thickened Lubricants," *ASLE Transactions*, Vol. 3, pp. 124–132.

(2) Same Authors, 1961, "Calculated Performance of Non-Newtonian Lubricants in Finite Width Journal Bearings," ibid., Vol. 4, pp. 275–281.

(3) Same Authors, 1963, "Calculated Performance of Greases in Journal Bearings," ibid., Vol. 6, pp. 239–248.

(4) Horowitz, H. H., 1958, "Predicting the Effects of Temperature and Shear Rate on the Viscosity of Viscosity Index-Improved Lubricants," *Ind. Eng. Chem.*, Vol. 50, pp. 1089–1093.

(5) Hamilton, G. M., and Bottomley, L., May 1987, "Measurement of Viscosity Loss of Polymer-Containing Oils at High Shear Stresses," *Tribology International*, pp. 60–67.

Author's Closure

I would like to thank the discusser for his insightful comments and will try to respond to the questions raised in the discussion.

It is not clear how "effective apparent viscosity in the circumferential/axial direction" is defined by the discusser. We treat viscosity as a scalar which is considered mathematically sound. The factors F_0, F_2 and F_3 are cross-film integrals of viscosity and have no direction associated with them.

In response to the discusser's call for application of this analysis to a 5W-30 oil with more accurate data we present additional results for a 5W-30 oil where the Cross equation was used for representing the shear-rate dependent behavior of the oil (Cryoff et al.). The Cross equation parameters used are shown in Table A.1. Figure A.1 (a) shows the film thickness and Fig. A.1 (b) the power loss as a function of crank angle for a 5W-30 non-Newtonian oil. Curves for two Newtonian oils with viscosities evaluated at zero shear (Newtonian 1) and at infinite shear (Newtonian 2) are also included for comparison. The results are similar to the cases discussed in the paper. The only difference is that in this case the non-Newtonian oil is very close in its response to a Newtonian oil at infinite shear.

Table A.1

Cross equation parameters for 5W-30 oil
(Oil No. 22 Cryoff et al, 1990)

$$\mu_1(\mu_0) = 8.822 \times 10-3 \text{ Pa.s}$$
$$\mu_2(\mu\infty) = 4.829 \times 10-3 \text{ Pa.s}$$
$$A = 2 \times 10-11$$
$$C = 2207.24$$
$$N = 0.92346$$

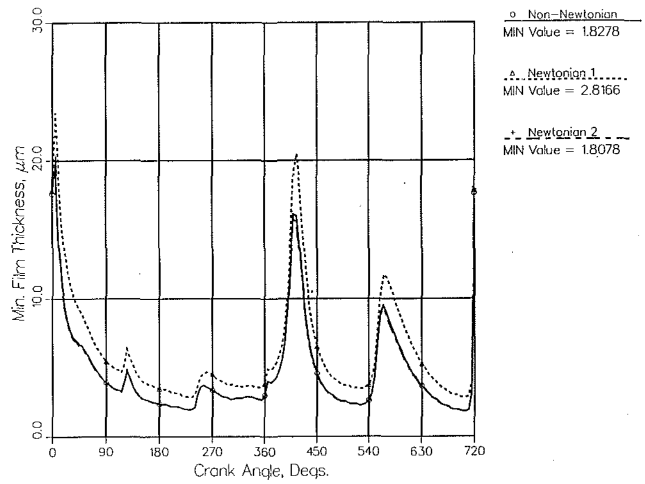

Fig. A.1 *(a)* **Minimum film thickness**

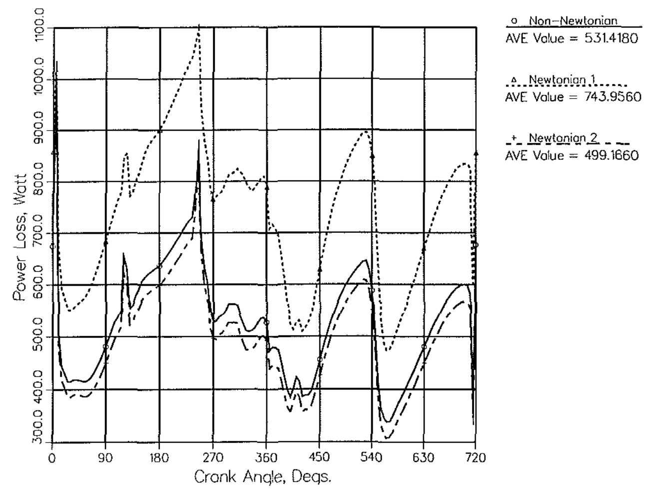

Fig. A.1 *(b)* **Power loss**

Fig. A.1 Results for a 5W-30 oil with the shear-rate dependent behavior of the oil represented by the Cross equation for a main bearing of a typical automotive engine running at 5000 r/min.

It is felt that all the physics necessary for simulation of dynamically loaded bearings, viz. squeeze film and journal inertia are correctly included. The case of zero rotational speed is analyzed. A sinusoidal load given by:

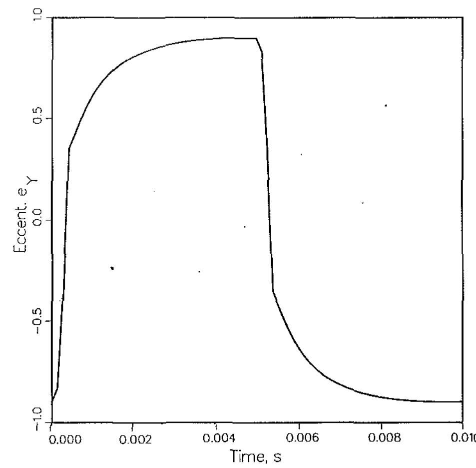

Fig. A.2 The case of zero rotational speed (pure squeeze) for a bearing subjected to sinusoidal loading

$$F_X = 0.0$$

$$F_Y = \text{A} \sin(\omega t)$$

where

$$A = 20.0 \text{ kN}$$

$$\omega = 628.3 \text{ rad/s}$$

is applied to the main bearing described in Table 1, with the exception that the half groove is replaced by end feeding arrangement to preserve circular symmetry. The algorithm rapidly converges to a periodic solution. The calculated Y-direction eccentricity, $e_Y(e_X = 0)$ is shown in Fig. A.2. It is apparent that a large load is supported by squeeze film alone.

I agree with the discusser that it is not very surprising that the power loss is reduced for a non-Newtonian oil as compared with a Newtonian oil. However, it is not true that reducing film thickness always reduces the power loss. For instance if film thickness is reduced by simply forcing the journal to a greater eccentricity (implying higher load), the power loss would increase.

Reference

1 Cryoff, S. A., Spearot, J. A., and Bates, T. W., 1990, "Engine Bearing Oil Film Thickness Measurement and Oil Rheology-an ASTM Task Force Report," SAE paper 902064.