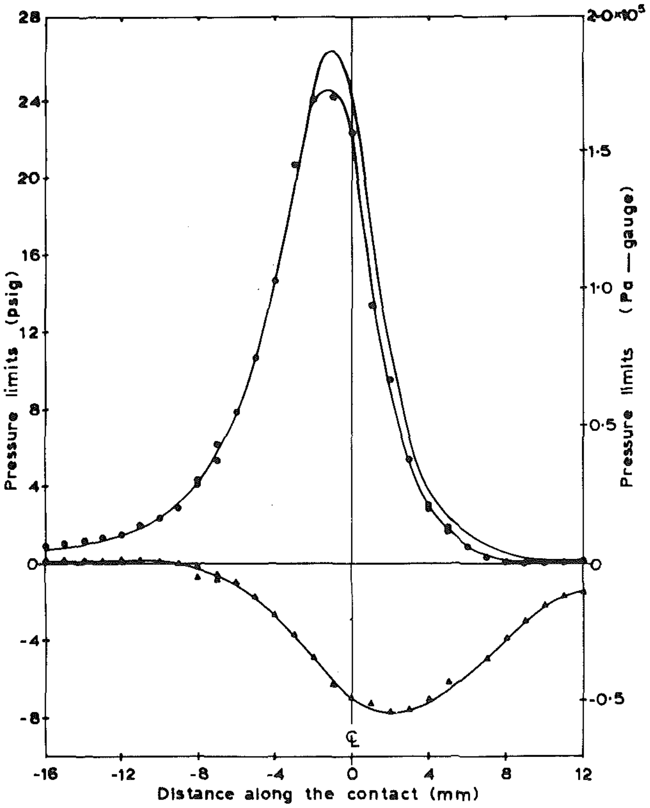

**Fig. 9 Upper and lower limits of pressure in the conjunction for a rotating drum under the conditions of Fig. 8. Average dynamic viscosity = 0.138 Pa s. (— Theoretical; -●- Experimental, upper limit, -▲- Experimental, lower limit.)**

ported in this paper is demonstrated in Fig. 9. The theoretical lower limit of pressure is the zero gauge pressure line which is compatible with the cavitation boundary condition used in the analysis. Consequently, the two lower limits of pressure are far apart. The most interesting feature of Fig. 9 is the agreement between the two upper limits, particularly over most of the inlet (converging) part of the conjunction.

**Inertia Effects.** A solution taking into account the effect of inertia forces upon fluid flow for the type of the conjunction under investigation is not yet available. It is instructive, however, to refer to Kuhn and Yates [6] who have shown that for squeeze motion between two circular plates, the ratio of maximum pressure allowing for inertia forces to maximum pressure neglecting inertia forces is

$$\mathcal{R} = 1 + \frac{1}{24}(2 + 3\alpha).\text{Re} \qquad (1)$$

where the Reynolds number $\text{Re} = h_0\dot{h}_0/\nu$ and the correction factor $\alpha$ has a value between 1 (for constant velocity profiles) and 1.54 (for a laminar pure Poiseuille flow). For a coupled Poiseuille and Couette flow, it is expected that the range of variation of $\alpha$ will be less important [6]. Using an alternative method, Kuhn and Yates came up with a value of $\alpha = 1.5$.

Thus with this value of $\alpha$,

$$\mathcal{R} = 1 + \frac{13}{48}\text{Re} \qquad (2)$$

Applied to the conditions of the present investigation, $(h_0\dot{h}_0)_{\max} \simeq 11.6 \times 10^{-6}\ \text{m}^2/\text{s}$ and $\nu_{\min} \simeq 1.35 \times 10^{-4}\ \text{m}^2/\text{s}$, gives $\text{Re}_{\max} \simeq 0.086$, hence $\mathcal{R}_{\max} \simeq 1.023$. This indicates that the maximum error which may result from ignoring the effect of inertia forces acting on the fluid is 2.3 percent.

Although the conditions leading to equation (2) are rather removed from those described in the present investigation, the above estimate of the maximum error is thought to be acceptable.

## Conclusions

The experimental results described in this paper are in good agreement with the theoretical predictions of the analysis presented in Part I despite the shortcomings of the outlet boundary condition used in that analysis.

Under conditions of pure "normal" motion, the lubricant is unable to sustain the full subambient pressures predicted by theory for the "normal" separation part of the motion of the plate. Cavitation sets in leading to pressures which are intermediate between ambient and those predicted by theory. With rolling superimposed, pressures are measured which, in front of the cavitation region at outlet, may be more than 0.5 bar (0.05 MPa) below ambient during part of the 'normal' separation phase. Despite this, there is remarkable agreement with theory during the remainder of the cycle of the plate vibration involving maximum oil film pressures just below two bar (0.2 MPa) gage. These observations regarding the magnitude of subambient pressures in the oil film are in agreement with earlier observations by Dowson and others.

Finally it is believed that the inertia forces, ignored in the analysis, are unlikely to have a significant effect on the experimental results reported in this paper.

## Acknowledgment

The authors greatly appreciate the award of a Research Studentship to Dr. P. H. Markho by the Esso Petroleum Company to enable him to undertake the investigation described in this paper in the Institute of Tribology at the University of Leeds.

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## DISCUSSION

### *James J. Kauzlarich*[2]

The authors are to be congratulated for their interesting paper on *the dynamic performance of line contacts. Their experimental results* help to establish the importance of cavitation on hydrodynamic lubrication involving squeeze film motion.

In their analysis for the dynamic squeeze film behavior of the line contact, the usual assumption of negligible inertia effects is taken in the theory, and justified for their test conditions. The following

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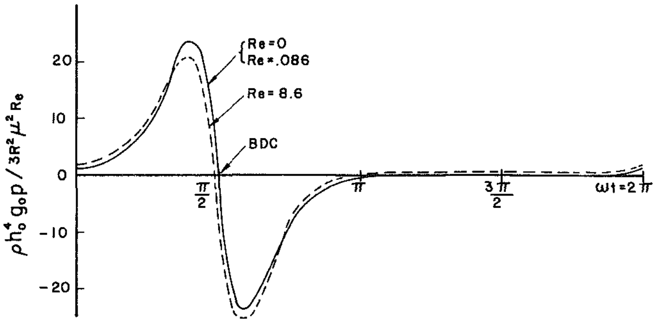

**Fig. 10 Squeeze film pressure at low Re**

analysis offers a bit more information concerning squeeze film inertia effects.

The theory for parallel, circular, flat plate, hydraulic squeeze film bearings was presented by Kauzlarich[3] (equation (18)), where the gage pressure at the center of the plate is given by

$$\frac{p}{3R^2\mu \mathrm{Re}/\rho g_0 h_0{}^4} = \frac{\cos \omega t}{(1 - e \sin \omega t)^3} - \left[\frac{\sin \omega t}{10e(1 - e \sin \omega t)} - \frac{5 \cos^2 \omega t}{28(1 - e \sin \omega t)^2}\right] \mathrm{Re} \qquad (3)$$

The first term is due to viscous effects and the other terms are due to inertia effects. The squeeze Reynolds number is

$$\mathrm{Re} = \frac{\rho h_0 \dot{h}_{\mathrm{max.}}}{\mu} \qquad (4)$$

and the "squeeze eccentricity" for the bearing is given by the ratio of amplitude to mean film thickness as

$$e = \frac{a}{h_0} \qquad (5)$$

Several plots of the dimensionless pressure at the authors' eccentricity of $e = 0.803$ are shown in Fig. 10. Bottom dead center (BDC) is noted and represents 0 rad on the authors' plots. It is evident that inertia effects are negligible for the test Re = 0.086, and not very great for 100× the test Re. The maximum pressure occurs at $\omega t = 0.8\pi/2 = 72°$, where the pressure ratio based on equation (3) with and without inertia terms is

$$\frac{p'}{p} = 1 - \frac{0.308}{e}(1 - 0.95\mathrm{e})^2\mathrm{Re} + 0.055(1 - 0.95\mathrm{e})\mathrm{Re} \qquad (6)$$

Fig. 10 suggests that the squeeze film played no role during a great part of the squeeze cycle. Keeping the amplitude smaller and closing the minimum gap may produce more interesting results. It would be of interest to test the equipment with degassed lubricant to see if the cavitation pressure will become lower as expected.

## Authors' Closure

The authors are grateful to Dr. Kauzlarich for his discussion of their paper. It is interesting to note that the flat-plate analysis presented earlier by Dr. Kauzlarich confirms the negligible effect of fluid-inertia for the conditions of the authors' experiments. This is entirely consistent with the conclusion reached by the authors on the basis of the calculations referred to equations (1) and (2) which are summarized in the paper. There appear to be some transcription errors in the discussers equations (3) and (6) but these do not materially affect the conclusions.

The illustration in Dr. Kauzlarich's Fig. 10 that the pressure remains fairly constant over much of the cycle is also evident in Figs. 5 of the authors' paper. This is expected for the conditions of the test and if squeeze-film action alone had been the subject under investigation, it would indeed have been appropriate to consider the behavior of the film at different amplitudes and smaller values of minimum film thickness. However, the main purpose of the study was to explore the film characteristics under combined rolling, sliding and normal approach and the experimental conditions were selected with this in mind.

The authors agree that tests on a degassed lubricant could be instructive.

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[3] Kauzlarich, James J., "Hydraulic Squeeze Bearing," *ASLE Trans.*, Vol. 15, No. 1, 1972, pp. 37–44.