DISCUSSION

J. F. Booker

The recent Leeds-Lyon Symposium included related papers by the present authors [22] and by a group from the University of Nottingham [23] which seems to be following the general route previously established by Oh & Huebner [9]. The authors might wish to comment on similarities and differences in the numerical and experimental approaches of the two groups. In particular, it would be interesting to know the reasons(s) behind the present authors' choice of a finite difference (rather than finite element) scheme for the fluid film solution. Or was it simply a matter of taste?

Two rather interesting simplifying constraints are employed to reduce the dimensions of the present analysis. Firstly, the cavitation boundary condition is made one-dimensional. Secondly, the elastic deformation is made two-dimensional. Both constraints have been employed by others, both are appealing, and both are probably justifiable here. Could the authors amplify a bit their investigation of the effect of the elastic constraint?

The authors are to be congratulated on apparently achieving convergence over a whole range of physically-realistic loads and stiffnesses; importantly, they have done it for the practical problem of specified journal loads (rather than displacements).

As might be hoped, this ambitious work has not just quantified the qualitative predictions of intuition; it has also produced a number of quite unexpected results as well. In particular, the "lumpiness" of some of the pressure distributions comes as a surprise. So does the high value of the pressures found in unusually flexible bearings. Amplification and clarification of the discussions(s) of classical elastic models [4, 5, 7] might put the new results in better perspective.

It is not surprising to learn that piezoviscosity has little effect on minimum film thickness (though the direction of the small effect on maximum film pressure is expected). It is surprising, however, to find that the effect of elasticity on minimum film thickness is also essentially negligible. Together these two (non-) results suggest why classical rigid and isoviscous predictions of minimum film thickness seem to work so well as the basis for design.

The proposed dimensional equation (19) for minimum film thickness is worthy of careful study. Could the authors comment on its applicability for situations in which the speed and load vary independently? (In this regard they might wish to expand their interpretation of Fig. 9 and the resulting relation (14).)

As its title accurately represents, the present work reports the steady-state analysis of connecting-rod bearings—an obvious contradiction in terms. Could the authors comment on the possibilities for extending their approach (or similar ones) to transient analysis? (Progress in this direction along the lines of the previous work by Rohde, et al., is reported in another ASME-ASLE Conference paper [24].)

Clearly the authors have presented an extremely significant and comprehensive study. Quite simply, I wish I had done it; consequently any unintentional negativity in previous remarks should be interpreted as simple jealousy.

Additional References

S. M. Rohde

The authors are to be congratulated on having attacked an interesting and important problem. Certainly elastohydrodynamic effects associated with the connecting rod are intimately related not only to bearing performance, but also to the durability of the rod itself.

The one point which disturbs this discussier is the presence of many oscillations in the computed pressure distributions. It has been our experience that these oscillations are due to the numerical scheme (iteration) adopted. One can easily prove that so many oscillations cannot exist by considering the one dimensional problem. In that case, we have:

\[
\frac{d}{dx} \frac{1}{h^2} \frac{dp}{dx} = \frac{dh}{dx}
\]

Integrating once we get

\[
\frac{dp}{dx} = \mu \left( \frac{1}{h^2} - \frac{C}{h^3} \right)
\]

where \( C \) is a constant.

Hence, at those points where \( dp/dx = 0, h \) must equal \( C \). From Fig. 6, we see that a horizontal line (constant \( h \)) will intersect the computed film thickness at no more than two (or possibly three) points between \( \phi = 90 \, \text{deg} \) and \( 270 \, \text{deg} \). These points thus represent the only possible local extrema.

Film thickness distributions, however, represent a somewhat different situation. It has been our experience that the film thickness distributions (and hence the minimum film thicknesses) converge much more rapidly than the corresponding pressure distributions.

The authors are encouraged to use methods such as in [10-11]. Convergence problems, using those methods, are almost entirely eliminated. Typically 3-5 iterations are sufficient for convergence using those methods. A more complete discussion of convergence in EHD problems can be found in [25].

Finally, have the authors computed the coefficient of friction for the compliant journal bearing? Judging by the film thickness distributions, it appears that the coefficient of friction may go up appreciably in the elastic case.

Additional Reference

F. A. Martin, C. S. Lee, and D. R. Adams

The authors are to be congratulated for showing in a concise way how related variables, in an "elastic" connecting rod bearing, affect both the minimum oil film thickness and developed film pressures. Although their geometric model is relatively simple, they nevertheless give a useful guide to trends in performance.

We were interested to see the graph (Fig. 6) of predicted film thickness against bearing angle for the connecting rod bearing (with a load of 25000 N). This showed a large region around the bearing where the oil film thickness was relatively constant and brought to a load of 600 rev/min diesel test engine. With the load on the cap half

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6 Cylinder, 600 bhp, 600 rev/min Diesel Engine.
Fig. 14 Film thickness around bearing at instant B when maximum inertia load occurs

<table>
<thead>
<tr>
<th>CASE</th>
<th>A</th>
<th>B</th>
<th>C</th>
</tr>
</thead>
<tbody>
<tr>
<td>steady load (in cap half)</td>
<td>max dynamic load (in cap half)</td>
<td>dynamic load to give min. film thickness (in rod half)</td>
<td></td>
</tr>
<tr>
<td>steady load eccentricity ratio $e_A$</td>
<td>max. eccentricity ratio $e_B$</td>
<td>min. eccentricity ratio $e_C$</td>
<td></td>
</tr>
</tbody>
</table>

$W_1 = 2, \quad W_2 = 0.42, \quad \frac{R}{L} = 0.42$

Fig. 15 Maximum eccentricity ratio for various load conditions

C considers the load (in rod half) which is associated with the smallest film thickness throughout the load cycle.

In the lower graph of Fig. 15 we have compared the eccentricity ratio for the various load conditions A, B, and C plotted against the eccentricity ratio which would occur for the steady load case A. For the authors load condition of 25000 N it can be seen that the film thickness with the dynamic load (B) is about 1.75 times that for the steady load case A, both using a rigid bearing model. It is possible that similar trends may also apply using an elasto-hydrodynamic model. It is also of interest to note that the smallest predicted film thickness (for the rigid bearing with inertia loads) occurs with a moderate load in the rod half as shown by case C.

At AE Developments my colleague, D. R. Adams, has been using finite element methods to calculate connecting rod distortion. The calculations are being used to provide information on a number of factors important in rod design, e.g., tendency of the rod to wrap itself around the crankpin, bearing stresses induced by rod distortion, and the interfacial slip between bearing shell and rod as an indicator of fretting at the interface. In this work the effects of the oil film, believed to be of secondary importance, were neglected. Fig. 16 shows the displaced shape, calculated by finite element analysis, of an oblique split connecting rod when subjected to peak inertia load. The variation of connecting rod/pin gap is also shown— together with the variation of stress normal to the bearing surface over the contact region. The analysis shows that the load is in fact distributed over an arc of contact of about 150°, which corresponds to the almost parallel oil film shown at the highest load condition in Fig. 6 of the authors paper. The normal stress distribution also shown in Fig. 16 resembles aspects of the hydrodynamic pressure distribution, particularly in respect of the peak at the ends of the contact (see authors Fig. 5). While this suggests that to a good approximation the gross distortion of the connecting rod can be calculated, under high loads, without the necessity of considering the hydrodynamics. Conversely in order to predict oil film conditions it would appear feasible to use the gross distorted bearing shape (with no oil film as
in Fig. 16) as a starting point in the iterative process of the elastohydrodynamic model.

The authors have tackled a difficult task and have produced a useful first guide to the overall general effect of the many variables involved in the connecting rod bearing model. However, the discussers suggest that the precise geometry of the rod, including the location of the bolts, split lines, local changes in cross section together with more appropriate dynamic load conditions will have a substantial effect on the hydrodynamic analysis and this will eventually need to be taken into account when applying this type of analysis to an actual connecting rod.

Authors' Closure

The authors wish to thank the discussors for their kind and constructive comments. In answer to Messrs. Martin et al., the authors agree that the exact shape of the rod must be taken into account and that apparently minor changes such as variations in the bolt position can induce significant variations in the deformed rod shape.

Further, the calculation of rod deformations does, in our mind, depend on the presence of the oil film. That point is illustrated in figure 17 which shows the variation of film thickness along the contact arc. Clearly that variation is small as its maximum value is 1.5 μm but it is significant if compared to the minimum film thickness.

In answer to Professor Booker's Comments:
1) There is no particular reason for the choice of finite difference and finite element methods in the case tested. In fact both solutions existed in the laboratory, respectively for the fluid and solid part of the problem, and were combined in this particular study. We tend to believe that for lubrication as opposed to structure problems, finite differences are if anything somewhat more efficient and easier to use.

2) Concerning the two dimensional boundary condition, the solution used could have been more clearly stated by writing \( p = \partial p/\partial n = 0 \) for \( \theta = \psi (z) \) which shows that the unknown boundary \( \psi \) solved for in this case is a function of \( z \).

3) The two dimensional elastic solution had to be chosen to keep within reason the time needed for the calculation. At high loads the simplified method presented here requires already more than 30 minutes on a CDC 6600 computer to yield the full deformation.

4) In connecting-rod bearings the inertia load is known to vary with the square of the rotational speed; the load decreases therefore faster than the speed. In our calculations in which the effect of the applied load is studied the load varies independently of the speed which is fixed at 5700 rpm. Calculations showed that it was not possible to decrease the speed for the maximum load tested of 25000 N because the film thickness dropped below 1 μm. This explains why in our study speed and load were decreased concurrently. Further the plot of \( \text{Ln}(h_{\text{min}} \times W^{0.85}) \) against the speed shows that all the data points are on a straight line (fig. 9) such that:

\[
h_{\text{min}} \times W^{0.85} \sim N^{0.49}
\]

thus, for a constant load one obtains:

\[
h_{\text{min}} \sim N^{0.49}
\]

5) The authors think that the technique used in this study is not fast enough to be extended to transient effects. Other techniques such as those developed by Rohde and Oh [10] might indeed prove to be more efficient.

In answer to Dr. Rohde:
1) We were also concerned with the many oscillations of the pressure but finally explained them in the following manner. The full two dimensional Reynolds equation presents both terms in \( \partial^2 p/\partial z^2 \) and in \( \partial p/\partial x \). Both these terms have to be nil to justify a one dimensional approach. While it is certain that \( \partial^2 p/\partial z^2 \) is indeed equal to zero on the center line, \( \partial^2 p/\partial x^2 \) is not nil at that section as the pressure curve does not show an inflexion point there. Therefore it

does not seem to us that the conclusion concerning the number of pressure peaks drawn from the one dimensional approach can be extended to our study. In the case shown in figure 17 in which a quasi continuously convergent film is calculated, four pressure peaks were obtained which result from large changes in slope (\( \partial h/\partial x \)) associated with very thin film thickness (h).

Calculations performed more recently with stricter convergence criteria decrease the amplitude of the oscillations which are observed in the center of the pressure curve but maintain without modification the two peaks noted at entry and exit. Film thickness was however not at all modified in the same proportion which agrees with the discussor's statement concerning the more rapid convergence for film thickness than for pressure. Further the existence of more than one pressure peak is confirmed experimentally by Bozaci et al. [26]
2) The techniques presented in this paper were initiated before the authors knew the method developed by the discussors. This method has been tested since in another application and does indeed converge more rapidly.

3) The answer to the last question is given in figure 18. The torque presented in this figure was calculated assuming that the inactive zone of the bearing is partially flooded. The increase in torque varies from 6% for small load to 8% for the maximum load tested.

Additional Reference