$$A_{i,n+2} = \int_\Omega \Big\{ 3H^2 \nabla P \cdot \nabla \psi_i \sin\theta + 6\left[U\cos\theta + V\frac{\sin\theta}{\Delta T} \right] \psi_i \Big\} d\Omega,\ i = 1,n$$

$$A_{n+1,i} = \int_\Omega \psi_i \cos\theta d\Omega,\ i = 1,n$$

$$A_{n+2,i} = \int_\Omega \psi_i \sin\theta d\Omega,\ i = 1,n$$

$$R_{n+1} = W_x - \int_\Omega P\cos\theta d\Omega$$

$$R_{n+2} = W_y - \int_\Omega P\sin\theta d\Omega$$

$$A_{n+1,n+1} = A_{n+1,n+2}, A_{n+2,n+1}, A_{n+2,n+2} = 0$$

$$L\psi_i \doteq \sum_{k=1}^{3} L_{ki}\psi_i$$

L $((P-Q)/(\Delta T))$ is the dimensionless rate of change of deformation. It can be approximated by first premultiplying the vector $\{(P-Q)/(\Delta T)\}$ by the matrix $[L]$. This yields a vector of nodal values, and the values at interior points can then be obtained by interpolation.

In the foregoing expressions, upper case symbols are dimensionless quantities. Spatial quantities are non-dimensionalized by the bearing radius, while pressure is nondimensionalized by the Young's modulus E. Time is nondimensionalized by μ_0/E. ψ_i is the basis function corresponding to the ith node, such that

$$\psi_i = 1 \text{ at node } i$$
$$= 0 \text{ at all other nodes.}$$

ψ_i varies linearly in triangles which contain node i.

$G = -\alpha/E$

$U = u\mu_0/(ER)$

$V = 2$

Q = dimensionless pressure for the preceeding time step

DISCUSSION

J. Frene[4] and B. Fantino[5]

The authors have presented a very comprehensive elastohydrodynamic study of a connecting-rod bearing and should be congratulated. In our work (ASME Paper No. 84-Trib-43) which was presented at the open forum, we found that the elastic displacements could either increase or decrease the minimum film thickness depending on the load direction. This could be due to the fact that when the load is directed toward the small end the bearing is a very rigid structure and only local deformations appear as in rolling EHD problem. But when the load is directed toward the cap end of the connecting-rod the bearing structure is very deformable and the entire shape of the bearing is modified. In this case we found that the minimum film thickness is decreased. Does the author have any references on this question?

Also, could the authors give the locus of the shaft center for both rigid and elastic bearings?

F. A. Martin[6]

The authors are to be commended for successfully predicting the performance of a connecting rod bearing under dynamic loads using an elastohydrodynamic solution, albiet it taking 35 hours of CPU time on the computer.

Figures 6 and 9 are of particular interest to the discusser as these are the first available predicted film thickness results which can be transformed into a special format[7] showing limiting distorted clearance shapes. The method of construction is as follows:

(*a*) choose an origin representing the journal center

(*b*) draw radial lines at small angular intervals of the circumferential coordinate θ

(*c*) mark film thickness along these lines at the relevant angle θ

(*d*) draw limiting lines at right angles

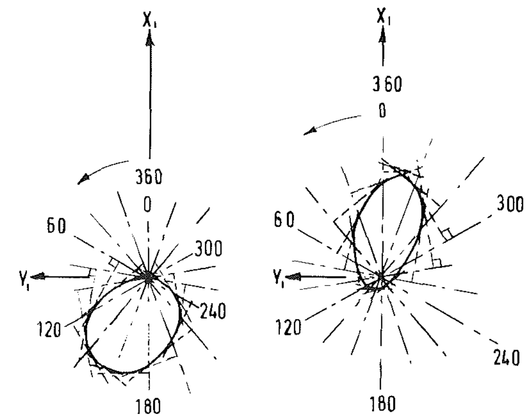

Fig. D1 Limiting clearance shapes

(*e*) draw curves generated by (*d*) resulting in the limiting clearance shape

The authors results have been transformed from Fig. 6 (load on rod) and Fig. 9 (load on cap) to the limiting clearance shapes shown in Figs. D1 (*a*), and (*b*), respectively. Such diagrams are not intended to replace the linear plots but rather to supplement them. They show the position of the minimum film thickness and the limiting clearance shape. Physically this shape can be interpreted by considering the bearing "frozen" at that instance and the shape can then be described by the trace of the journal center if the journal where rolled around the bearing surface (in the same way as a clearance circle applies to a circular bearing). Figure D1(*b*) shows the stretching effect when the load is the cap half of the bearing. It is believed that the "fish tail" effect at the top of that diagram indicates that the bearing radius, locally, is less than the journal radius and hence it is outside the limiting "clearance" (to journal movement).

It is hoped that limiting clearance shapes such as shown in Fig. D1 relating to the authors work will give insight into bearing behavior. The discusser would be pleased to hear the authors views on such diagrams for future use.

Additional References

(D1) Campbell, J., Love, P. P., Martin, F. A., and Rafique, S. O., "Bearings for Reciprocating Machinery: A Review of the Present State of Theoretical, Experimental and Service Knowledge," *Proc. I. Mech E 1967*, Vol. 182, Part 3A, pp. 51–74.

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[7] This special format was initiated by Martin in 1967 [D1] when presenting experimental data from the Ruston and Hornsby 6VEB III engine.

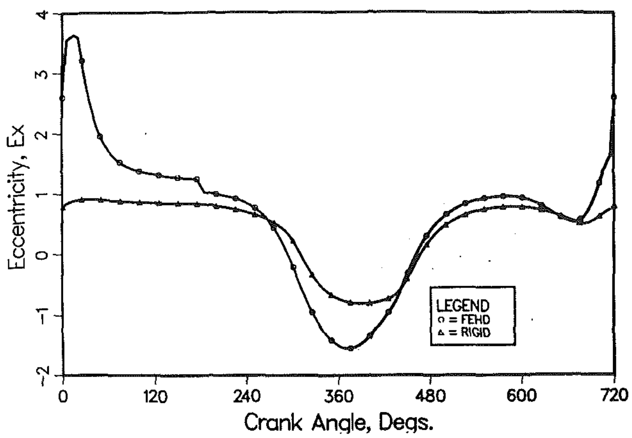

Fig. R1(a)

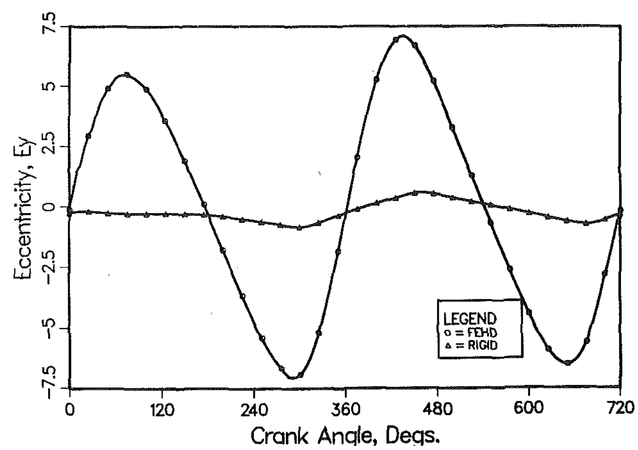

Fig. R1(b)

J. F. Booker[8]

In just six pages the authors have given us convincing evidence that their program for EHD journal bearing analysis is the current "state-of-the-art." One only hopes that more details of applications will appear in their Authors' Closure or in future publications in which the space constraints are not so severe. Particularly welcome would be detailed studies of the (harmful and/or beneficial) effects of elasticity in both analysis and design.

The authors have evidently combined Newton-Raphson and Murty algorithms in a robust and efficient procedure. Still, they are clearly correct in their call for faster (and possibly approximate) methods more suited to the design process. Possibilities which come to mind include the "direct" method of reference [14] as well as the "short bearing" approach of reference [5]; hybrid schemes which limit the degrees of freedom are also possible. Perhaps the authors would care to comment on their own progress in these or other directions.

The authors' achievement is especially impressive when viewed in the context of two other closely related papers [15, 16] fortuitously presented at the same Joint Conference. Perhaps someday soon the authors' Sample Problem (or its equivalent) can serve as the basis of a comparison for these three calculation schemes and any others which may appear subsequently. Meanwhile, theirs appears to be the benchmark by which other work must be judged.

Additional References

15 La Bouff, G. A., and Booker, J. F., "Dynamically Loaded Journal Bearings: A Finite Element Treatment for Rigid and Elastic Surfaces," ASME/ASLE Joint Lubrication Conference, San Diego, California, October 22-24, 1984, ASME Paper No. 84-Trib-11.

16 Fantino, B., and Frêne, J., "Comparison of Dynamic Behavior of Elastic Connecting-Rod Bearing in Both Petrol and Diesel Engines," ASME/ASLE Joint Lubrication Conference, San Diego, California, October 22-24, 1984, ASME Paper No. 84-Trib-43.

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Author's Closure

The authors are grateful for the discussions by Messrs. F. A. Martin, and B. Fantino and J. Frene. The comments are very constructive, and the authors plan to take advantage of them in their future work. Specific responses to the discussors' comments are given in the following:

F. A. Martin

The diagrams which result from the transformation method in Martin's reference [1] do add to the physical understanding of the journal lubrication problem. The authors will try to incorporate this method in future graphics output produced by the program. The process of drawing the limiting clearance shapes may be somewhat difficult, however, to implement in a computer program.

B. Fantino and J. Frene

The authors have performed the calculations described in the paper for many connecting rod designs, some drastically different in the way the load is supported by the structure underneath the journal bearing. Based on those results, the authors agree that the minimum film thickness around the bearing at a specific crank angle can indeed increase or decrease with the bearing structure compliance. The observation the discussors made about the connecting rod structure "pinching in" is certainly possible, and is very likely when the engine is running at a high speed.

The loci of the shaft center are shown in Fig. R1 as requested by the discussors. The large eccentricities in the y-direction in the EHD case are due to the bending of the connecting rod at about the cut-off point. Changing the location of the cut-off point will significantly affect the EHD loci, but will not affect the pressure and film thickness solutions.