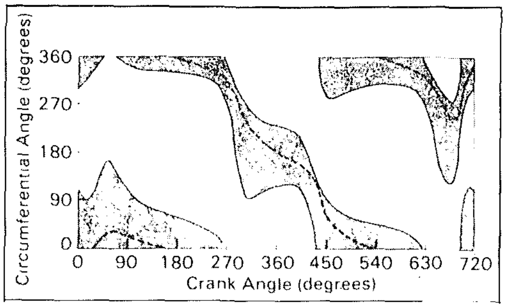


**Fig. 12 Pressure curve extent plot for the bearing as obtained from the new curve fit (partial extent).**
**□ $P < 0.01\, P_{max}$, ■ $P \geq 0.01\, P_{max}$, · · · · Location of $P_{max}$.**

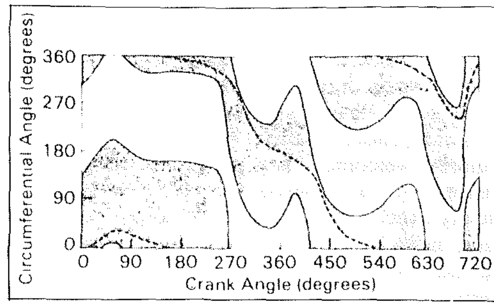


**Fig. 13 Pressure curve extent plot for the bearing as obtained from the FEM analysis (full extent).**
**□ $P = 0$, ■ $P > 0$, · · · · Location of $P_{max}$.**

12. The pressure curve extent plot that includes all the areas of positive pressure, as obtained from the FEM analysis, is shown in Fig. 13. The effect on the pressure curve extent, of removing the pressures that are less than 1% of the maximum value, is dramatic. This small modification in the pressure curve extent definition frees up a large area of the bearing for the location of oil holes.

## Closure

The mobility method with the new curve fit, Map 4, gives solutions which are almost identical to the FEM solutions. The mobility method solution is faster than the FEM solution by a factor of about 2000. Though conceptually the new curve fit is applicable only to ideal bearings, it has been used to solve centered and off-center half-grooves, and eccentric (offset) bearings with satisfactory results. These applications greatly enhance the value of the new curve fits. Although only one example is presented in this paper, the results from the new curve fits compare very well with the FEM result for all other cases investigated to date.

## Acknowledgment

The author would like to thank Mr. Michael Adam of the Adam Open Division of General Motors for his assistance in data acquisition.

## References


1 Goenka, P. K., "Dynamically Loaded Journal Bearings: Finite Element Method Analysis," GM Research Publication GMR-4308, ASME Journal of Lubrication Technology, Vol. 106, No. 4, Oct. 1984, pp. 429–439.

2 Booker, J. F., "Dynamically Loaded Journal Bearings: Mobility Method of Solution," ASME *Journal of Basic Engineeering*, Sept. 1965, pp. 537–546.

3 Booker, J. F., "Dynamically Loaded Journal Bearings: Maximum Film Pressure," ASME Journal of Lubrication Technology, Vol. 91, July 1969, pp. 534–543.

4 Booker, J. F., "Dynamically-Loaded Journal Bearings: Numerical Application of the Mobility Method," ASME Journal of Lubrication Technology, Vol. 93, January 1971, pp. 168–176. Errata: No. 2, April 1971, p. 315.

5 Childs, D., Moes, H., and vanLeeuwen, H., "Journal Bearing Impedance Descriptions for Rotordynamic Applications," ASME Journal of Lubrication Technology, Vol. 99, Apr. 1977, pp. 198–214.

6 Moes, H., Discussion, I. Mech. E. 1969 Tribology Convention, Gothenburg, Proceedings of the Institution of Mechanical Engineers, Vol. 183, Part 3P, 1968–1969, pp. 205–206.

7 Moes, H., and Bosma, R., "Mobility and Impedance Definitions for Plain Journal Bearing," ASME Journal of Lubrication Technology, Vol. 103, July 1981, pp. 468–470.

8 Barr, A. J., et al., *A User's Guide to SAS 76*, SAS Institute, Inc., Raleigh, N. C., 1976, pp. 127–144.

9 Graybill, F. A., *An Introduction to Linear Statistical Models*, Vol. I, McGraw-Hill, New York, 1961.

10 Goenka, P. K., "Analytical Curve Fits for Solution Parameters of Journal Bearings," GM Research Publication GMR-4315, Mar. 1983.

11 Blok, H., and Herrebrugh, K., "Discussion of Reference [3]."

12 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," *Proceedings of the Institute of Mechanical Engineers*, 1967–1968, Vol. 182, Part 3A.

13 Martin, F. A., "Developments in Engine Bearings," *9th Leeds-Lyon Symposium on Tribology*, "Tribology of Reciprocating Engines," Sept. 1982, *Tribology International*, Vol. 16, No. 3, June 1983, pp. 147–164.


# DISCUSSION

**Harry van Leeuwen**[3]

In his paper the author shows that his new curve fits for the mobility vector components are much more accurate than any other existing curve fit, and approach the very accurate finite element solution very well. In addition, he develops curve fits for the maximum pressure vector, and for the location of the boundaries of the pressure distribution. I am very grateful that the author has spent so much energy in finding these approximations, in order that the bearing designers can save tremendous amounts of computer calculation time in future.

This remark gives rise to the question: is a very accurate curve fit worth all effort to obtain it, knowing that in practice most bearings show oil feed holes and groves, and elastic distortion, which ruins the symmetry and is in conflict with the basic assumptions of the mobility theory? If, for example, the most accurate theoretical (FEM) results are compared with the measurements reported in [12], where $h_{min} = 3.30$ $\mu$m on the Glacier dynamic similarity machine, and $h_{min} = 2.79$ $\mu$m in the real engine, the curve fit from [4] does not appear to be too bad.

It is interesting to find that the position of the author's curve $P_{max} = 1.6$ in his Fig. 5 shows some deviation of the curve obtained by Blok and Herrebrugh in their Fig. 5 of [3]. To the discusser's knowledge, the Blok and Herrebrugh map is the only possibility in the literature for a check. The former solution is based on a FEM method, while the latter is based on a FDM method. In both cases, boundary conditions are Reynolds. Maybe the author has some idea where the difference comes from?

The Moes and Bosma [7] curve fit is compared to the accurate FEM solution in Table 3, where it can be found that it has an overshoot of +9.5 percent, while the older Moes curve fit [4] shows an undershoot of −6.9 percent. Probably, the author has performed calculations using both solutions [4] and [7]. It is true that, in general, solution [4] has to be preferred over solution [7]?

[3]Eindhoven University of Technology, Department of Mechanical Engineering, 5600 MB Eindhoven, Netherlands

**J. F. Booker[4] and S. Govindachar[4]**

The mobility method has now survived two decades. Those who hovered nearby during its infancy and early development can take a certain vicarious satisfaction in observing from a distance its independent maturation.

Particularly welcome, then, is word of Dr. Goenka's development of a consistent and complete set of "numerically-exact" data for mobility and pressure "maps." These data, in the form of their associated curve fits, promise to extend indefinitely the useful life of the mobility method. With them, it forms a perfect complement of the more elaborate (and enormously more costly) analysis schemes [1] now becoming available to study effects of geometric irregularities.

Thus, *in the long run*, we would expect the presently pre-eminent "short bearing" curve fits to fall into disuse. However, *in the short run*, the existing data base of field experience is expressed almost entirely in terms of "short bearing" results which bear no simple relationship to "finite bearing" results (such as generated with the present curve-fits). Until a comparable body of field experience is correlated with the latter, prudent designers will presumably follow the easiest (and soundest) path, using results of *both* "short bearing" *and* "finite bearing" analyses in their comparisons.

It is worth noting that the "short-bearing" approximation is widely assumed to be *qualitatively* correct, whatever its *quantitative* inaccuracies. Even these are often difficult to assess. For example, Table 2 would suggest (unfairly) that the short bearing approximation represents maximum pressure poorly, while Table 3 suggests (fairly) that it is the minimum film thickness (and thus the orbit itself) which is unreliable!

In summary, then, we heartily (and paternally) approve the major contribution made by these new curve-fits to the permanent literature of the mobility method. We also applaud provision of Table 1 as a check on the typographic elves which have plagued most previously-published curve-fits, including our own. (We can testify from our own experience that coding the present curve fits in a few dozen FORTRAN statements does indeed reproduce the numerical data of Table 1 – at least prior to final publication!)

[4] School of Mechanical and Aerospace Engineering, Cornell University, Ithaca, N.Y. 14853

## Authors' Closure

Professor J. F. Booker and Dr. H. vanLeeuwen have raised some interesting points. The mobility method, no doubt, has its limitations. It can be used only for ideal journal bearing configurations and by assuming rigid bearing surfaces. It is true that elastic deformations may significantly effect the results, but to the best of my knowledge, EHD analysis has not yet become a part of the bearing design process. For most bearing designers, rigid bearing analysis is still the state-of-the-art. I agree that the new curve fits cannot do all that the FEM program described in [1] can do. But the results obtained using the curves are almost as accurate as those obtained through the use of the FEM program and at a small fraction of the cost. This one attribute makes the new curve fits "worth all the effort to obtain them." They are sufficient for many routine bearing designs, a good starting place for solving more complex design problems, and indispensable for parametric studies.

The deviation observed by Dr. vanLeeuwen in Fig. 5 of this paper, as compared to the corresponding figure of [3], is not in the data. Figure 5 is plotted from the pressure curve fit. The map from the actual FEM data, if plotted, will look very similar to the map in [3]. The pressure data in the region in question was very difficult to curve fit. Since it is not a particularly significant region, the data was simplified. Hence the curve fit is approximate.

My experience with solution [7] is limited to what is described in the paper. Based on that, it is difficult to make a general statement comparing the solution in [4] and [7].

Professor Booker's comments about existing databases are very well taken. These curve fits have been available at General Motors for over two years and we have been able to generate enough of a database to rely on the new curve fits. I am hoping that over the years the same will be done by other bearing users/manufacturers and then the finite bearing analysis can be used in routine design work.