Discussion on a paper by H. Maes, E.G. Sikkes and R. Bosma "Mobility and impedance tensor methods for full and partial-arc journal bearings"

Where ξ' and η' are defined by

$$\xi' = x'/C = \epsilon c\alpha$$

$$\eta' = y'/C = \epsilon s\alpha$$

For arbitrary cartesian coordinates follows

$$\xi' = (XV_X + YV_Y)/(VC)$$

$$\eta' = (YV_X - XV_Y)/(VC)$$

$$V_X = \dot{X} + \bar{\omega}Y$$

$$V_Y = \dot{Y} - \bar{\omega}X$$

$$V = (V_X^2 + V_Y^2)^{1/2}$$

Solutions for the mobility

Exact closed-form solutions:

Complete-film short-bearing mobility

$$M_x^{2\pi,0} = (1 - \xi^2 + 2\eta^2)H$$

$$M_y^{2\pi,0} = -3\xi\eta H$$

$$H = (2\pi\lambda^2)^{-1}(1-\epsilon^2)^{3/2}(1+2\epsilon^2)^{-1}$$

Complete-film long-bearing mobility

$$M_x^{2\pi,\infty} = (1 - \xi^2 + \tfrac{1}{2}\eta^2)H$$

$$M_y^{2\pi,\infty} = -3\xi\eta H/2$$

$$H = (6\pi)^{-1}(1-\epsilon^2)^{1/2}$$

Curve-fit solutions (exact along the centerline, i.e. for $\eta = 0$):

Ruptured-film short-bearing mobility

$$M_x^{\pi,0} \cong H$$

$$M_y^{\pi,0} \cong -3\eta(1-\xi)^{-1}H/2$$

$$H = (2\lambda^2)^{-1}(1-\xi^2)^2\{(1+2\xi^2)(1-\xi^2)^{-1/2}\bar{c}(-\xi) + 3\xi\}^{-1}$$

Ruptured-film long-bearing mobility

$$M_x^{\pi,\infty} \cong H$$

$$M_y^{\pi,\infty} \cong -3\eta(1-\xi)^{-1}H/4$$

$$H = 6^{-1}(1-\xi^2)\{(1-\xi^2)^{-1/2}\bar{c}(-\xi) + \xi\}^{-1}$$

Where ξ and η are defined by

$$\xi = x/C = \epsilon c\gamma$$

$$\eta = y/C = \epsilon s\gamma$$

For arbitrary cartesian coordinates follows

$$\xi = (XF_X + YF_Y)/(FC)$$

$$\eta = (YF_X - XF_Y)/(FC)$$

$$F = (F_X^2 + F_Y^2)^{1/2}$$

References

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

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

3 Booker, J. F., "Dynamically Loaded Journal Bearings: Numerical Application of the Mobility Method," ASME Journal of Lubrication Technology, Jan. 1971, pp. 168–176.

4 Blok, H., "Topological Aspect and the Inpulse/Whirl Angle Method in the Orbital Hydrodynamics of Dynamically Loaded Journal Bearings," Lecture notes (condensed English version), Delft, The Netherlands, Aug. 1965.

5 Childs, D., Moes, H., and Van Leeuwen, H., "Journal Bearing Impedance Descriptions for Rotordynamic Applications," ASME Journal of Lubrication Technology, Apr. 1977, pp. 198–215.

6 Moes, H., Sikkes, E. G., and Bosma, R., "Dynamic Analysis of Diesel Engine Main Bearings," 10th Leeds-Lyon Symposium on Tribology, Lyon, 6–9 Sept. 1983.

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

8 Tondl, A., "*Some Problems of Rotor Dynamics*," London, 1965, pp. 122–130.

9 Moes, H., Discussion, I. Mech. E., 1969 Tribology Convention, Gothenburg, *Proceedings of the Institute of Mechanical Engineers*, Vol. 183, Part 3P. M sub y in this discussion has to be multiplied by y.

10 Ten Napel, W. E., Moes, H., and Bosma, R., "Dynamically Loaded Pivoted Pad Journal Bearings: Mobility Method of Solution," ASME Journal of Lubrication Technology, April 1976, pp. 196–205.

11 Booker, J. F., A Table of the Journal Bearing Integral," ASME *Journal of Basic Engineering*, June 1965, pp. 533–535.

DISCUSSION

Harry van Leeuwen[1]

In their paper the authors present a very elegant and attractive generalization of the well-known mobility and impedance vector methods for dynamically loaded journal bearings. One of the inherent limitations of a *vector* notation is the bearing circumferential symmetry or the fixed load direction, thus confining mobility and impedance methods to full journal bearings, or a special type of pivoting pad bearings, respectively. As many bearings in rotating machinery are unsymmetric or do not have a fixed load direction, this is a really severe limitation. By adopting a *tensor* notation, the authors show how to avoid this restriction, while at the same time keeping the advantage of the mobility and impedance vector methods, viz. low computer time. Hence, it is an ideal method for the analysis of large complicated systems with many fluid film bearings, seals, and squeeze film dampers, and for many repeated calculations. Can the authors give some idea of possible computational time savings, compared to conventional numerical techniques, as used in [10]?

The similarity between Figs. 2 and 3 is impressive, especially when bearing in mind that in Fig. 2 the simple Ocvirk cavitation boundaries are used. Presumably the dashed lines in Fig. 2 are thresholds marking cavitation occurring at the outside of the pad. If this is true, the cavitation regions under these conditions are much larger than shown in [10], Fig. 8, where a Reynolds cavitation boundary condition is used. But the shape of the squeeze paths and the magnitudes of the iso mobility lines are quite similar. Comparison of this Fig. 2 with Fig. 2 from [10], where cavitation could not be simulated, shows that the iso mobility lines are quite different in the cavitation area. So the conclusion is near that the mobility tensor curve fit (with Ocvirk cavitation boundaries) yields better results than the mobility vector full numerical solution (without cavitation simulations), which is remarkable.

The authors hypothesize that a good approximation of finite length mobility tensors can be obtained from an addition of short and long bearing mobility tensors, as is evidenced by their Fig. 2 and by full journal bearings. Finite length impedance tensors can be obtained from finite length mobility tensors by inversion. Could the authors comment on the general validity of their hypothesis, e.g., in the case of multibore bearings?

At first sight this discusser was puzzled by equation (13*a*), which shows that the impedance tensor for long bearings is symmetrical. This seems to be denied by [5], equation (20),

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

where an unsymmetrical notation appears. It took some time before it became clear that no cavitation or film periodicity conditions are present in equation (13a). If these conditions are substituted, the impedance tensor elements can be transformed into equation (20) of [5], so the long bearing impedance tensor according to I5I is seemingly unsymmetrical!

Although inertia has been neglected in the example, it does not present a limitation to the tensor approach. Have the authors treated the inertia case? Of course there are still some restrictions, as constant viscosity and rigid surfaces. Can the authors suggest possible generalizations of the tensor approach, thus avoiding other (inherent?) limitations, or minimizing their impact?

The removal of a severe limitation in the mobility and impedance method for dynamically loaded journal bearings is a big step forward to a better description of their behavior. The authors are to be commended for their ingenious generalization of existing methods.

Authors' Closure

The authors are grateful for the discussion by Mr. H. J. van Leeuwen. In particular since his comment answers some of the questions that may be raised when reading this concise paper. The following specific responses apply to the open questions of the discusser:

1. The question about the time savings to be expected is a rather difficult one. How to compare a micro-computer with a mainframe? By applying the methods proposed both the CPU time and the storage area needed are substantially reduced. Therefore some of the authors' simulations may even be executed on a $50 programmable hand calculator!
2. The hypothesis made by the authors, stating that the addition of the long and short bearing mobility tensors leads to a good finite length bearing approximation, is meant to be valid for all fluid film bearings, provided the surfaces are smooth and have a uniform curvature. However, for most bearing types the hypothesis still has to be confirmed by comparing the approximative solution with accurate numerical data. Yet for full journal bearings and partial-arc bearings, and consequently multibore bearings, adequate evidence exists.
3. The authors did not yet simulate the inertia effects in pivoted pads. Though it may quite simply be performed by applying the tensor methods (instead of the vector methods applied by way of an example) in combination with a balancing of the inertia forces with the force component perpendicular to the pivot position, i.e., F_x in Fig. 1.
4. The restrictions to constant lubricant viscosity and rigid surfaces seem to be absolute. Maybe this is the penalty to be paid for efficiency!