# DISCUSSION

## J. H. Vohr[2]

I would like to compliment the authors on their worthwhile contribution to design theory for spiral-grooved bearings. I tend to agree, however, with the observation made by Mr. Wildmann in his discussion that the paper, as presented, would have been more valuable had it included some comparisons with previous analyses, such as those in references [1, 6, and 7].[3] These previous analyses have all employed the simplifying assumption that the local pressure profiles within groove and land regions can be approximated by straight lines. Since most of the design data now available for spiral-grooved bearing is from these analyses, it is important to determine the conditions under which the linear assumption leads to significant error. This the present authors should be able to do with their numerical analysis.

Concerning details of the present paper, I believe equation (6) as it is now written is in error. The groove-to-land "jump" equation is obtained from the condition that the mass flow normal to a groove-to-land boundary is continuous across the boundary. When the mass flows in the land and groove regions are expressed in terms of the local land and groove pressure profiles, this condition of continuity of normal mass flow gives the following "jump" equation which should replace equation (6).

$$ {}_L\Delta_G \left\{ h^3 \frac{\partial P}{\partial \theta} \right\} - r^2 f'(r) \left[ {}_L\Delta_G \left\{ h^3 \frac{\partial P}{\partial r} \right\} \right] = 6\mu\omega r^2 (h_L - h_G) $$

where

$$ {}_L\Delta_G \left\{ h^3 \frac{\partial P}{\partial r} \right\} = h_L{}^3 \left[ \begin{matrix} \text{limit} \\ \theta - f(r) \end{matrix} \right] \rightarrow 0^+ \frac{\partial P}{\partial r} - h_G{}^3 \left[ \begin{matrix} \text{limit} \\ \theta - f(r) - \frac{2\pi}{N} \end{matrix} \right] \rightarrow 0^- \frac{\partial P}{\partial r} $$

Note that both $\frac{\partial P}{\partial \theta}$ and $\frac{\partial P}{\partial r}$ are discontinuous at groove-to-land boundaries. The reader can satisfy himself that the change of variables $\xi = r$ and $\eta = \theta - f(r)$, when substituted in the foregoing equation, will yield equation (13) in the text.

### Additional References


6 E. A. Muijderman, "Spiral Groove Bearings," Philips Research Reports Supplement, No. 2, Philips Research Laboratories, Kastjelaan, Eindhoven, The Netherlands, 1964.

7 M. Wildmann, "Grooved Plate, Gas-Lubricated Thrust Bearings With Special Reference to the Spiral Groove Bearing," ASME Paper No. 64—Lub-25.


[2] Mechanical Technology Inc., Latham, N. Y. Assoc. Mem. ASME.

[3] Numbers 6 and 7 in brackets designate Additional References at end of discussion.

## M. Wildmann[4]

The authors have obtained finite difference numerical solutions to Reynolds equation for a gap geometry usually known as "spiral groove." Whether this spiral groove is a bearing, a pump, or a seal does not really enter into the solution to Reynolds equation. The spiral groove can be either a bearing, a pump, or a seal, depending on the user's viewpoint. Like all hydrodynamic bearings, it is really all of these things at once.

In addition to the numerical solution, the authors have also presented some experimental data. Very few finite difference solutions and very little experimental data on spiral groove bearings with a compressible lubricant have been published. The authors' contributions are therefore welcome.

An approximate spiral groove bearing solution was first obtained by Whipple [8].[5] This solution gives a transcendental equation (not an infinite series) which relates all the bearing parameters to the performance of the bearing as a bearing or a pump. The only major assumption made by Whipple is that the number of grooves is large so that the pressure change across the groove is linear. The limitations of this solution have been examined in detail by Wildmann [9], and improvements to the solution have been made by Muijderman [10]. The great advantage of the Whipple solution is, of course, that it gives the bearing performance in algebraic form, which is much easier to use than a finite difference numerical solution.

The authors' contribution would have been much more valuable if they had compared their numerical solution to the one obtained by Whipple and used their solution mainly to determine what are the limitations of Whipple's solution. The more accurate–and more cumbersome—numerical solution would then be needed only when the simpler Whipple solution does not give adequate results. By not even referring to the Whipple solution, the authors give the misleading impression that only numerical solutions can be used for this particular geometry.

### Additional References


8 R. T. P. Whipple, "Theory of Spiral Grooved Thrust Bearing With Liquid or Gas Lubricant," Atomic Energy Research Establishment, Harwell, Berkshire, England, T/R 622, 1951.

9 M. Wildmann, "Grooved Plate, Gas-Lubricated Thrust Bearings With Special Reference to the Spiral Groove Bearing," ASME Paper No. 64—Lub-25.

10 E. A. Muijderman, "Spiral Groove Bearings," Thesis, Technological University, Delft, The Netherlands, March, 1964; also, Philips Research Reports Supplement, No. 2, Philips Research Laboratories, Kastjelaan, Eindhoven, The Netherlands, 1964.


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[5] Numbers in brackets designate Additional References at end of discussion.