GEOMETRY AND SPEED EFFECTS ON THE PERFORMANCE OF THE HERRINGBONE GROOVED GAS LUBRICATED JOURNAL BEARING

Foster, D.J.

Member ASME

ABSTRACT
The plain gas lubricated journal bearing is dynamically unstable. The addition of grooving on the bearing surface has been found to overcome this instability. In particular, the herringbone groove pattern has been found to provide damping with increase of static stiffness. The effect of the available geometry characteristics on performance is computed from solution of the compressible Reynolds equation. The geometry features examined are groove angle, and the ratios:- groove depth to clearance, groove-to-land width, axial groove length to bearing length and bearing eccentricity. The performance is determined over a range of compressibility numbers.

Keywords: Compressible Hydrodynamic Grooved Bearing

INTRODUCTION
A commercially available program [1] uses the Galerkin Finite Element method of weighted residuals with quadratic basis to convert a non-linear, two-dimensional partial differential equation into discrete nodal equations. An initial coarse grid of triangular patches is used. The initial computed solution is used to determine the error in each patch. Where the error exceeds a user-specified value, the cell is subdivided and the solution process is repeated.

The program cannot be used to solve the lubrication equation for a journal bearing directly, since it assumes a plane domain with explicit Neumann or Dirichlet boundary conditions. Although the journal-bearing surface can be cut along a generator and unwrapped onto a plane, the boundary conditions at the cut ends are not known. By applying a conformal transformation to one half of the unwrapped surface, the problem domain is converted to a hollow disc, where the outside diameter represents the end of the bearing, at ambient pressure, and the inside diameter represents the bearing centerline with a zero normal-flow condition. The cut ends of the unwrapped surface are joined naturally at the zero and two pi angular coordinates of the disc. The integrals of pressure to obtain resultant force and attitude angle are also transformed to the solution cylindrical coordinates. The conformal transformation is described under Nomenclature.

NOMENCLATURE
Physical variables
- $c$: bearing nominal clearance
- $e$: bearing center offset.
- $gh$, $gw$, $gl$: groove depth, width, axial length
- $h = c + [gh] + e \cos(y/r)$: local bearing clearance,
- $lt$: bearing total length
- $p_a, p$: ambient pressure, local pressure
- $r$: journal radius
- $x$: distance from journal end, toward center
- $y$: circumferential co-ordinate on journal
- $\mu$: Gas viscosity
- $\omega$: Bearing angular velocity

Dimensionless variables
- $N$: Number of grooves
- $X = x/r$, $Y = y/r$, $L = l/2r$
- $\varepsilon = e/c$, $GH = gh/c$, $H = h/c$
- $GW = gwN/2\pi r$, $GL = 2gl/lt$
- $\Lambda = 6\mu\omega r^2/p_a c^2$: compressibility number, Lambda

Conformal Transformation
The dimensionless, unwrapped journal surface is the domain $Z = X + iY$ and the solution domain is $W = R\exp(i\theta)$. The conformal transformation $W = \exp(-Z)$ infers the geometric relationship $R = \exp(-X)$; $\theta = Y$. Axial boundaries $X_e = 0$; $X_e = L$ have corresponding radial boundaries $R = 1$; $R = 1/\exp L$. The normalized compressible Reynolds equation in solution co-ordinates, but in terms of the cylindrical compressibility number, becomes
$$\text{div} \left[ H^3 P \text{grad} (P) + \Lambda H \text{P grad} (\theta) \right] = 0$$
This is the preferred format for the program PDEASE2.
Resultant loads and stiffness were computed in dimensionless form as follows, to compare to published results [2].

\[
W_1 = \frac{1}{p_r r^2} \int_{0}^{\pi/2} \int_{0}^{2\pi} p \cos \frac{\theta}{r} r dx dy = \int_{0}^{\pi} \int_{0}^{2\pi} \frac{P}{R^2} \cos \theta RdRd\theta
\]

\[
W_2 = \frac{1}{p_r r^2} \int_{0}^{\pi/2} \int_{0}^{2\pi} p \sin \frac{\theta}{r} r dx dy = \int_{0}^{\pi} \int_{0}^{2\pi} \frac{P}{R^2} \sin \theta RdRd\theta
\]

\[
St = \frac{\sqrt{W_1^2 + W_2^2}}{\pi L \varepsilon} = W/\left(\pi p_r rl \varepsilon\right) \text{ (Ausman’s stiffness)}
\]

SOLUTION SPECIFICS
Separate solution regions were defined for the grooves; the program guarantees continuity of pressure and normal flow at their boundaries with the ungrooved surface. Because of the presence of pressure spikes at the inside corners of the grooves, the default accuracy tolerances for the program had to be tightened; the individual cell tolerances were specified to be the same as the overall solution tolerance of 0.001

PARAMETER VARIATIONS STUDIED
The maximum pressure, dimensionless stiffness and attitude angle were determined for a series of analyses. A full permutation of all parameters was not feasible in the time available. The parameter ranges used were:
- Compressibility number (Lambda) \( \Lambda = 1,2,..,10,20,..,100 \)
- Groove angle \( \phi = 15,30,45,50,55,60,65,70 \) degrees
- Groove depth/clearance ratio \( GH = 0.0,0.5,1.0,1.5,2.0,2.5,3.0 \)
- Length/diameter ratio \( L = 0.4 \) to 2 in 8 steps for \( GL = 0.49 \)
- Groove length ratio \( GL = 0.42 \) to 1.0 at \( L = 2 \)
- Number of grooves \( N = 4,8,12,16,24 \)
- Groove width/Groove spacing ratios \( GW = 0.25 \) and 0.5

RESULTS
Unless otherwise stated, the conditions were

\( L = 1, \varepsilon = 0.1, \Lambda = 50, GL = 0.69, \phi = 60 \) and \( N = 16. \)

1. The method was validated by comparison to Ausman’s linear-ph results [2], for \( \varepsilon = 0.2, L = 1 \) and \( L = 4.5 \). The values for stiffness and attitude angle agreed within five percent across the range of compressibility numbers from 0.1 to 100.
2. The best groove angle for stiffness is shown in Fig.1 to be between 55 and 65 degrees, for length-to-diameter ratios \( L \) of 1,1.5 and 2. An angle of 60 degrees also gave the smallest attitude angle, 2 degrees, c.f. 8 degrees for zero angle.

3. Fig. 2 shows that 16 grooves had the best stiffness, for \( GH = 0.5 \) to 3; \( \Lambda = 1 \) to 100; \( N = 8,12,16,24 \) at \( GW = 0.5 \) and \( N = 4,8,12,16 \) at \( GW = 0.25 \). The attitude angle was also least for 16 grooves. Note that plain journal is stiffer for \( 0 \leq \Lambda \leq 12 \)

4. The effect of groove depth on Stiffness and Attitude angle is shown in Fig.3. A value of \( GH = 1.6 \) gave best stiffness, but a value of 1.0 resulted in smallest attitude angle.

5. Change of length to diameter ratio from \( L = 0.42 \) to 2.0, at \( GL = 0.49 \) with \( GH = 0.5 \) to 3., caused an increase of stiffness with \( L \), but best attitude angle was obtained at \( L=1, \) GH=1, at Lambda of 50.

CONCLUSIONS
The transformation to polar coordinates has enabled a detailed examination of the performance of the herring bone journal bearing, for a range of possible geometries. An outline of the results from the large amount of data has been presented. The geometry for least attitude angle can be different than that for highest stiffness, which may be a stability consideration.

REFERENCES
1. PDEASE2. Finite element analysis for partial differential equations. Distributed by Macsyma,Inc