- ceramic ball/ceramic disc (kit 4): Slight abrasion of the ball and slight smoothing of the disc surface.

In Table 1 the mean values and standard deviations of friction and wear data for all material pairings investigated are compiled.

The results of the first round of the VAMAS Round Robin Comparison show that a good reproducibility of the numeric friction and wear data has been obtained which can be summarized in terms of the relative standard deviations (s_r and s_R according to ASTM Standard E 691 divided by the mean value) as follows:

- Reproducibility of the friction data:
  within laboratories: ± 9 to ±13 percent
  interlaboratory: ±18 to ±20 percent
- Reproducibility of specimen wear data
  within laboratories: ± 5 to ± 7 percent
  interlaboratory: ±15 to ±20 percent
- Reproducibility of system wear data
  within laboratories: ±14 percent
  interlaboratory: ±29 to ±38 percent

These results show that the overall reproducibility of systematic friction and wear measurements is comparable with that of other complex engineering quantities provided that the tests are performed under well controlled conditions.

APPENDIX

Participants of the VAMAS project on Wear Test Methods (Names of contact persons are added in brackets.)

Canada (3 participants)
- Ontario Hydro Research, Toronto, (P. E. Dale)
- National Research Council, Mechanical Research Department, Vancouver, (H. M. Hawthorne)
- National Research Council, Industrial Materials Research Institute, Canada, (J. Masounave)

Federal Republic of Germany (4 participants)
- Bundesanstalt für Materialforschung und -prüfung, Berlin, (H. Czichos)
- Technische Hochschule Darmstadt, (E. Broszeit)
- Technische Universität Berlin, (H. G. Feller)
- Robert Bosch GmbH, Stuttgart, (H. Schorr)

France (5 participants)
- Ecole Nationale Supérieure de Céramique Industrielle, Limoges, (P. Boch)
- Ecole Central de Lyon, (Ph. Kapsa)
- Hydromécanique et Frottement, Andrezieux-Bouthéon, (J. L. Polti)
- Institut National des Sciences Appliqués, Villeurbanne, (M. Godet)
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Great Britain (4 participants)
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- Brunel University, Uxbridge, (T. S. Eyre)
- National Physical Laboratory, Teddington, (E. A. Almond)
- University of Bradford, (T. H. C. Childs)

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- CISE, Milano, (R. Martinella)
- Istituto per le Ricerche di Tecnologia Meccanica, Vico Canavese, (G. Tipatti)

Japan (3 participants)
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- Government Industrial Research Institute, Osaka, (M. Iwasa)
- National Aerospace Laboratory, Tokyo, (M. Nishimura)

United States of America (9 participants)
- National Bureau of Standards, Gaithersburg, MD, (A. W. Ruff)
- National Bureau of Standards, Gaithersburg, MD, (S. Hsu)
- Eastman Kodak, Rochester, NY, (K. Budinski)
- EXXON Research and Engineering Co., Annandale, NJ, (T. E. Fischer)
- Sohio Engineered Materials Co., Niagara Falls, NY, (S. G. Seshadri)
- Corning Glass Works, Corning, NY, (J. W. Adams)
- GTE Laboratories Inc., Waltham, MA, (S. Wayne)
- Georgia Institute of Technology, Atlanta, GA, (W. Winer)
- Oak Ridge National Laboratory, Oak Ridge, TN, (C. Yust)

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Stability of Finite Orthogonally-Displaced Pressure Dam Bearings

N. P. Mehta[1] and Ajeet Singh[2]

Nomenclature

c = journal bearing radial major clearance given by a circle circumscribed on the bearing (L)
c_m = minor radial clearance given by inscribed circle on bearing (L)
d_h, d_v, δ_h, δ_v = horizontal, vertical presets of bearing lobes (L), d_h/c, dv/c
D = journal diameter (L)
e = eccentricity (L)
F = dimensionless shaft flexibility
L = bearing axial length (L)
L_d, L_t = pressure dam bearings dam width, relief-track width (L)
L_d, L_t = L_d/L, L_t/L
0_B, 0_J, 0_U, 0_L = bearing, journal, upper lobe and lower lobe centers
p = oil film pressure (FL^{-2})
$\bar{p}$ = dimensionless oil film pressure $p/\mu N(c/R)^2$
Rec = Reynolds number based on major radial clearance $\rho Vc/\mu$ (Dim)
S = Sommerfeld, $\mu NLD/W(R/c)^2$
S_d = step or dam depth
$\bar{\mathrm{S}}_d$ = S_d/c

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V = peripheral velocity (LT^{-1})
W = bearing external lead (F)
$\bar{W}$ = dimensionless load carrying capacity of bearing $\bar{W} = W/\mu NLD\ (c/R)^2$
X, Y = coordinate system for journal, X-horizontal, Y = vertically upwards
θ_s = location of step for pressure dam bearings measured in the direction of rotation from positive X-axis for anticlockwise rotation of the journal (degrees)
θg = oil supply groove angle at the leading and trailing edges of hydro-dynamic fluid film
ν = dimensionless threshold speed
ρ = fluid density (FT^2L^{-4})
ϕ = attitude angle of the bearing (degrees)
ϕ_1, ϕ_2 = angles (degrees)
ω = rotational speed (T^{-1})
$(\bullet)$ = first derivative with respect to time

During assembly, if the centers of the two halves of an elliptical pressure dam bearing are displaced horizontally, the bearing so obtained is termed orthogonally-displaced pressure dam bearing. The present paper describes the dynamic behavior of this bearing.

Introduction

References [1-4] give analytical dynamic analysis of the cylindrical pressure dam bearings. Their experimental stability analysis is available in [5-7], where it has been reported that analytical stability analysis provides the general trends in the experimental data. The analytical dynamic analysis of the noncylindrical pressure dam bearings is reported in [8-11]. In [11], it has been shown that an offset-halves pressure dam bearing is more stable than the other pressure dam bearings reported in the literature.

In this paper, the results of analytical dynamic analysis of the orthogonally-displaced pressure dam bearings are presented in the form of graphs. The graphs reported here include Sommerfeld number versus dimensionless threshold speed supporting a rigid rotor.

Bearing Geometry

Figure 1 shows the geometry of an orthogonally-displaced pressure dam bearing. This is a combination of a offset-halves [11] and an elliptical [9] pressure dam bearings. Center of each half is shifted along and perpendicular to the split-axis. d_h and d_v are the horizontal and vertical presets, respectively, which define the position of the lower lobe center, 0_L, and the upper lobe center, 0_U, of this bearing. 0_U is toward the right side of bearing center, 0_B, and below the split-axis, whereas 0_L is towards the left side of 0_B and above the split-axis for counterclockwise rotation of the journal. For concentric shaft position, there are thus two reference clearances. A minor clearance given by an inscribed circle denoted by c_m and a major clearance, c, given by a circle circumscribed on the bearing. For a given position of the journal center, 0_J, there are three eccentricities viz., being eccentricity, e, lower half eccentricity, e_1, and upper half eccentricity, e_2. The various eccentricities and presets are nondimensionalized by the major radial clearance.

The method of dynamic analysis of this bearing is described in references [8-11].

Results and Discussion

The values of horizontal and vertical presets are varied while the following design parameters are kept constant in the analysis

$$L/D = 1.0,\ \theta_s = 125\ \text{deg.},\ \theta_g = 10\ \text{deg.},\ \bar{s}_d = 2.0,\ L_d = 0.75\ \bar{L}_t = 0.25\ \text{and}\ R_{ec} = 210$$

Figure 2 shows the effect of presets on stability of the orthogonally displaced pressure dam bearing supporting a rigid rotor. The dimensionless threshold speed (maximum speed up to which a bearing is stable) has been plotted for different Sommerfeld numbers in this figure. The stability threshold curve divides any figure into two major zones. The zone above this curve is unstable, whereas, the zone below this is stable. The minimum value of this curve is termed as minimum threshold speed. The bearing is stable at all speeds towards the left side of this curve, which is called the zone of infinite stability. Very high stability threshold speeds are obtained as the values of the presets are increased, because of the very large hydrodynamic load generated in the upper half of this bearing.

Figure 3 compares the stability threshold speeds of various non-cylindrical pressure dam bearings having values of presets equal to 0.40. The stability threshold speeds of cylindrical pressure dam bearing having the same design parameters and plain journal bearing (L/D = 1.0) are also plotted. The orthogonally-displaced pressure dam bearing is the most stable one out of all the pressure dam bearings. The minimum threshold speeds of the orthogonally-displaced, offset-halves, elliptical, half-elliptical, and cylindrical pressure dam bearings are 12.0, 8.65, 6.95, 5.2, and 3.2, respectively. There is also a very large increase of the zone of infinite stability of the orthogonally-displaced pressure dam bearing as compared to the other bearings as is evident in Fig. 3.

Conclusion

The stability of two-axial groove cylindrical journal bearing is improved by cutting a step or pressure dam in the upper half surface and a deep groove or relief-track in the lower half surface. Furthermore improvement in stability of this bearing can be obtained by having noncylindrical bearing geometries as in the case of elliptical, half elliptical, offset-halves and or-

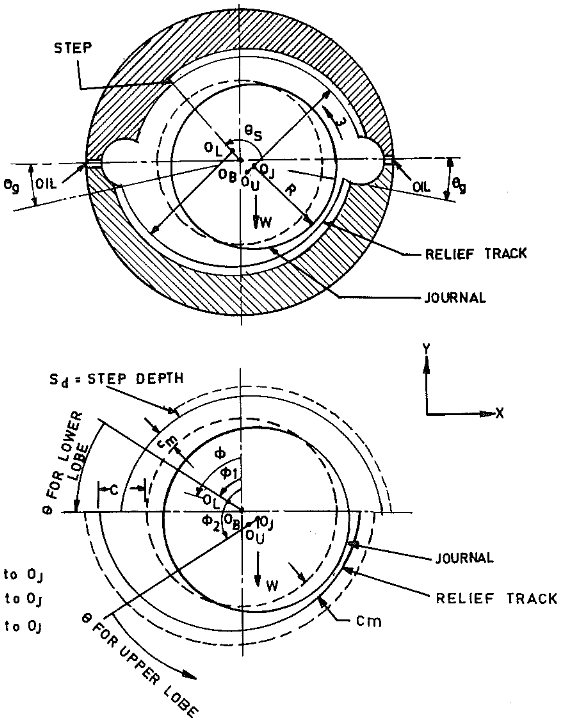
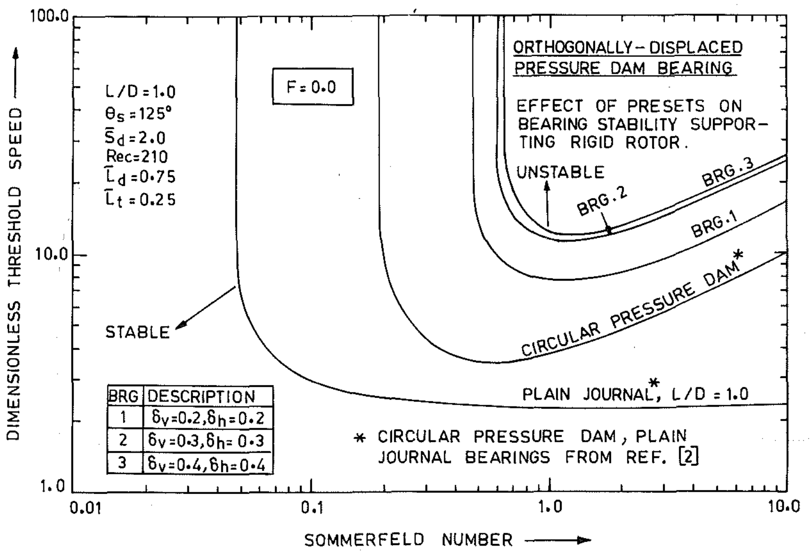
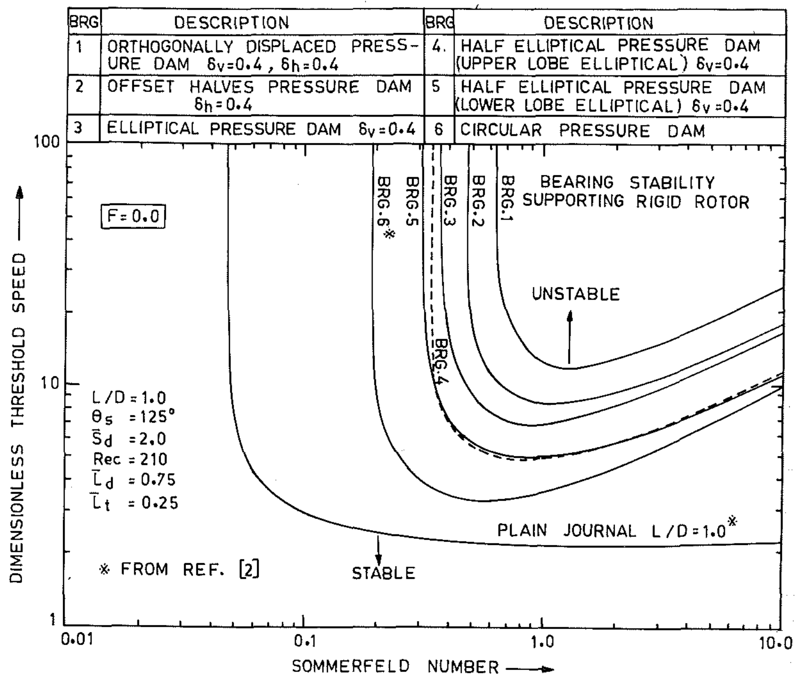


Fig. 1 Orthogonally-displaced pressure dam bearing geometry



BRG	DESCRIPTION
1	δ_v=0.2,δ_h=0.2
2	δ_v=0.3,δ_h=0.3
3	δ_v=0.4,δ_h=0.4

Fig. 2 Sommerfeld number versus dimensionless threshold speed for a rigid rotor



BRG	DESCRIPTION	BRG	DESCRIPTION
1	ORTHOGONALLY DISPLACED PRESSURE DAM δ_v=0.4, δ_h=0.4	4.	HALF ELLIPTICAL PRESSURE DAM (UPPER LOBE ELLIPTICAL) δ_v=0.4
2	OFFSET HALVES PRESSURE DAM δ_h=0.4	5	HALF ELLIPTICAL PRESSURE DAM (LOWER LOBE ELLIPTICAL) δ_v=0.4
3	ELLIPTICAL PRESSURE DAM δ_v=0.4	6	CIRCULAR PRESSURE DAM

Fig. 3 Sommerfeld number versus dimensionless threshold speed for a rigid rotor

thogonally displaced pressure dam bearings. Out of all these pressure dam bearings, orthogonally-displaced pressure dam bearing is the most stable having both very large minimum threshold speed and zone of infinite stability.

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