EFFECT OF JOURNAL MISALIGNMENT ON THE OIL FILM PRESSURE AND TEMPERATURE DISTRIBUTION OF 3-LOBE OFFSET JOURNAL BEARING

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ABSTRACT
The design of turbines and compressors operating at the high rotational speeds applies the 3-lobe journal bearings. In many cases the classic 3-lobe journal bearings supporting the rotors, are showing the problem of rotor stability. This problem can be avoided by the application of 3-lobe Offset bearings. This type of bearing fulfills the conditions of reliable bearing design and good stability in the case of high speed rotating machines.

INTRODUCTION
The design of turbines and compressors operating at the high rotational speeds applies the 3-lobe journal bearings [1,2]. In case of the classic 3-lobe journal bearings supporting the rotors, there are the problems of operation that concern the stability of rotor. The stability can be improved by the application of 3-lobe Offset bearings. This type of bearing fulfills the conditions of reliable bearing design and good stability in the case of high speed rotating machines.

Recently the offset type journal bearings are extensively investigated and applied in the rotating machines. The classic 3-lobe bearing (Fig. 1) can operate in both directions of journal rotation but the 3-lobe offset one operates only in one direction of rotation [1,2]. Generally, the single lobes of the multilobe bearing are designed as the arc of the circle with the centre points placed on the symmetry line of the single lobe. In the symmetric multilobe bearing the circle inscribed in the bearing profile is tangent to the lobe exactly at the middle point of each lobe. In case of bearing designed for only one direction of revolution as, e.g. 3-lobe offset one, this circle touches the end of the convex gap of the bearing.

The design and calculation procedure of both static and dynamic characteristics requires more investigation. The misalignment of journal and bearing axis is caused by the deformation of shaft or displacements of bearing supports [3]. The oil film pressure and temperature distributions are affected by the of misalignment of journal and bearing axis.

The paper introduces the results of calculation of oil film pressure and temperature distributions of symmetrical 3-lobe offset journal bearing at different operating parameters. For comparison purposes some results for the classic 3-lobe journal baring were given too. The Reynolds, energy, geometry and viscosity equations were solved numerically under assumption of incompressible lubricant, the laminar and adiabatic flow of oil in the bearing gap of finite length bearing. Aligned and misaligned orientation of bush and journal axis without deflections of bearing and journal was assumed.

NOMENCLATURE

- \( d \) journal diameter (m)
- \( D \) bush diameter (m)
- \( e \) journal eccentricity (m)
- \( h \) oil film thickness (\( \mu m \))
- \( \bar{H} \) dimensionless oil film thickness
- \( K_r \) heat number, \( K_r = \omega \eta \rho \cdot T_c \cdot \psi^2 \)
- \( L \) bearing length
- \( \bar{p} \) dimensionless oil film pressure, \( \bar{p} = p \cdot \psi^2 / \eta \cdot \omega \)
- \( r, R_b \) journal and bush radius (m)
- \( q \) inclination ratio, \( q = (L / s) \cdot \tan \kappa \)
- \( t \) time
- \( \tau \) dimensionless axial co-ordinate
- \( \eta \) dynamic viscosity of oil, (Ns/m²)
- \( \eta_o \) dynamic viscosity at ambient temperature (Ns/m²)
- \( \psi \) relative clearance of bearing, \( \% \)
- \( \psi_c \) lobe relative clearance
- \( \bar{\eta} \) dimensionless viscosity of oil, \( \bar{\eta} = \eta / \eta_o \)
- \( \varphi \) circumferential co-ordinate (°)
- \( \varepsilon \) relative eccentricity, \( \varepsilon = e / (R_b - r) \)
- \( \omega \) angular velocity (1/s)
- \( \phi \) dimensionless time, \( \phi = \omega t \)
- \( \kappa \) inclination angle in the middle plane of bearing
- \( O_b, O_j \) centre of: bearing and journal
OIL FILM PRESSURE AND TEMPERATURE DISTRIBUTIONS

Solution of basic equations of thermo-hydrodynamic theory of lubrication allows obtaining the data on the oil film pressure, temperature distributions, the maximum value of pressure and temperature of oil film, the minimum oil film thickness, oil flow and friction forces, that are the static characteristics determining the input variables for the design of bearing [3].

The oil film thickness is given by the following equation [2,3]:

\[ H = 1 + \varepsilon \cdot \cos(\varphi - \alpha) + q \cdot \zeta \cos(\varphi - \alpha) \] (1)

Pressure distribution was computed from Reynolds equation (2) on the assumption of variable viscosity [1-4].

\[ \frac{\partial}{\partial \varphi} \left( \frac{H^3 \partial p}{\eta \partial \varphi} \right) + \frac{D}{L} \left( \frac{H^3 \partial p}{\eta \partial z} \right) = \frac{6}{H} \frac{\partial H}{\partial \varphi} + 12 \frac{\partial H}{\partial \varphi} \] (2)

Oil film temperature and viscosity distribution has been found by the iterative solution of Eqn. (1), (2) and energy one [1-3].

RESULTS OF CALCULATIONS

Numerical calculations were performed for two different values of length to diameter ratio i.e. L/D = 0.5 and 0.8, clearance ratio \( \psi = 0.0 \), relative inclination ratio \( q = 0.0 \) and rotational speed of journal \( n = 6000 \) rpm.

These sets of parameters allow to determine the effect of bush geometry i.e. bearing length to diameter ratio L/D, relative inclination ratio \( q \) on the bearing operating characteristics which, e.g. include the oil film: pressure \( \overline{p} \), temperature \( \overline{T} \) distributions, Sommerfeld number \( So \), static equilibrium position angle \( \alpha_{eq} \). Some results of calculations for assumed journal relative eccentricity are presented in Fig. 2 and Fig. 3.

FINAL REMARKS

In the range of assumed value of inclination coefficient, the misalignment of journal and bearing axis affects the oil film pressure and temperature distributions; larger effects are in the pressure distribution. The profile of bearing bore changes the oil film pressure and temperature distributions.

REFERENCES