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Performance Analysis of Non-Newtonian Thermoelastohydrodynamic Journal Bearings

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***Abstract*— Today's compact design of rotating systems requires their supporting journal bearings to operate under more severe operating conditions in which fluid-film thickness becomes thin enough. The strong dependency of lubricant viscosity on fluid-film temperature necessitates a better knowledge of fluid-film temperature rise and its variation across and along the fluid-film for the determination of realistic lubricant viscosity. In such case, thermal consideration becomes most important and realistic condition in the analysis and design of journal bearings operating under full lubrication condition. In addition to this, the structural distortion due to thermo-elastic deformation as a result of hydrodynamic fluid-film pressure.**

Further, high molecular weight polymers such as polymethacrylate or polyisobutylene are added to mineral lubricants in order to conserve optimum pro perties under different operating conditions. These polymer added oil no longer behaves as Newtonian fluid and there exists non-linear relationships between shear stress and shear strain rate. Therefore, a study that predicts the influence of non-Newtonian behavior of the lubricant on journal bearing performance is also more essential. Due to the present day trend towards higher speed as well as the use of unconventional low viscosity lubricants such as water, liquid metal and synthetic oils etc., it is possible to arrive at such a situation where flow is laminar but fluid inertia forces cannot be neglected. Thus, a non-Newtonian thermo-elastohydrodynamic (TEHD) analysis of journal bearing which couples the combined influence of surface roughness, thermal, elastic distortion of bearing, non-Newtonian behavior of lubricant and fluid inertia effects is more realistic and appropriate. The present work predicts the influence of surface roughness and fluid-inertia on non-Newtonian thermoelastohydrodynamic (TEHD) performance of hydrodynamic journal bearing systems under more realistic operating condition of bearing by considering bearing flexibility, thermal and non-Newtonian behavior of lubricant.

I. INTRODUCTION

Today's compact design of rotating systems requires their supporting journal bearings to operate under more severe operating conditions in which fluid-film thickness becomes thin enough. As a result, some asperities on the mating surfaces begin to interfere inevitably, hence temperature increases due to asperity interaction and fluid shear action. This temperature rise in the lubricant results into the variation of viscosity along and across the fluid-film. The strong dependency of lubricant viscosity on fluid-film temperature necessitates a better knowledge of fluid-film temperature rise and its variation across and along the fluid-film for the determination of realistic lubricant viscosity. In such case, thermal consideration becomes most important and realistic condition in the analysis and design of journal bearings. In addition to this, the structural distortion due to thermo-elastic deformation as a result of hydrodynamic fluid-film pressure may significantly cause the change of the fluid-film thickness and hence the bearing performance.

From last few decades, the inclusion of surface roughness effects into the analysis of bearing lubrication problems has been conveniently carried out using statistical method. The following subsection details the inclusion of surface roughness into the analysis of lubrication problems through statistical analysis.

Due to unprecedented technological advances during last few decades, the operating conditions of rotating machines are becoming more stringent and exact. This has necessitated many new developments in the area of hydrodynamic journal bearings, since large number of machine parts use hydrodynamic journal bearing and operates continuously at high speed. As a consequence, a considerable amount of research activities have been devoted towards the study of hydrodynamic journal bearing systems and analysis has been carried out by incorporating many physical effects to get much more realistic performance data.

Furthermore, most of the available studies which deals with surface roughness effects in the analysis of hydrodynamic journal bearings have been carried out with the general assumption of negligible fluid inertia forces. However, due to the present day trend towards higher speed as well as the use of unconventional low viscosity lubricants such as water, liquid metal and synthetic oils etc., the fluid inertia becomes important for some range of moderately large Reynolds number. When the reduced

STATISTICAL ANALYSIS OF SURFACE ROUGHNESS

Surface roughness in the form of fine irregularities exists on all finished surfaces of tribological elements and it is inherent to the characteristics of surface finishing process such as turning, grinding, polishing, sand blasting, etc. Fortunately, for many rough surfaces of engineering importance, the surface height, curvature and peak height distributions are random in nature and are represented by a Gaussian probability distribution

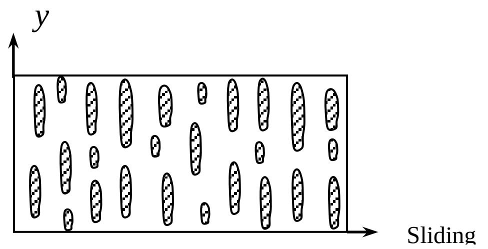
SURFACES WITH DIRECTIONAL PATTERNS

Many engineering surfaces exhibits roughness with directional patterns those resulting from different manufacturing process. The directional patterns (roughness orientations) of the rough surfaces may be isotropic or anisotropic depending upon the nature of the surface finishing process. The anisotropic roughness orientations are generally classified as transverse and longitudinal roughness patterns depending upon the principle directions of the surface textures with the sliding direction of opposing surfaces.

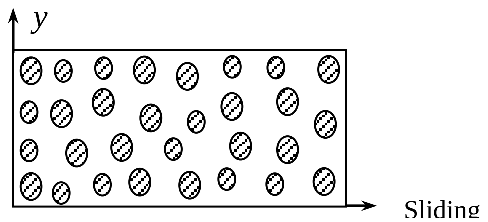
To study the effects of roughness orientations on lubrication problems, Peklenik used a parameter known as surface pattern parameter, γ, which is also popularly known as Peklenik number. This surface pattern parameter or Peklenik number is defined as

$$\gamma = \frac{\lambda_{0.5x}}{\lambda_{0.5y}} \qquad (1)$$

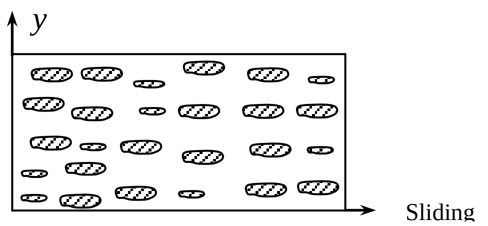
where $\lambda_{0.5x}$ and $\lambda_{0.5y}$ are the autocorrelation lengths of x- profile (i.e. along the sliding direction of opposing surface) and y-profile (i.e. in the direction perpendicular to the sliding direction of opposing surface) at which their value reduces to 50 percent of its original value.



(a) Transversely oriented



(b) Isotropically oriented



(c) Longitudinally oriented

Fig. 1.3 - Contact areas of different roughness orientations on unwrapped bearing surface

LITERATURE SURVEY

The literature review presented in this thesis mainly concerns with theoretical investigations of mixed lubrication performance of hydrodynamic journal bearing including the influence of surface roughness coupled with the effects of bearing shell flexibility, non-Newtonian behavior of lubricant and variation in viscosity of the lubricant due to temperature rise in fluid film. Over the last few decades numerous studies on hydrodynamic journal bearings have been carried out and reported in literature. Since the available literature in the area of these journal bearings is quite vast and abundant it is rather difficulty to present all these information. Therefore, few important studies which are most relevant to the present work are reviewed in the following section.

Among the many studies related to the non-Newtonian behavior of lubricant on performance of journal bearing, Dien and Elrod [5] derived a generalized Reynolds equation for the non-Newtonian fluids using perturbation technique. They selected the power law model as the constitutive relation for the non-Newtonian fluids and carried out a study on slider and journal bearings. Using this generalized Reynolds equation, Wada and Hayashi [7,8] derived a modified Reynolds equation for non-Newtonian lubricants obeying cubic shear stress law model [7]. It has been clarified experimentally that the flow characteristics of pseudo-plastic fluids are approximately expressed by a cubic shear stress model containing first and third powers of shear stress [8]. The experimentally measured pressure distribution and load carrying capacity of a journal bearing were found to match with theoretical results obtained from cubic shear stress model. Tayal solved the Navier-Stokes equations by finite element method. They used power law model to study the static performance characteristics of journal bearing. Their results demonstrated that the static performance characteristics of bearing are significantly affected by non-Newtonian behavior of lubricant.

The literature review presented in this subsection clearly indicates that the values of critical mass and threshold speed obtained from linear analysis do not set correct stability margins. On the other hand the nonlinear analysis of transient response generally predicts a higher value of stability parameters than that obtained from linear analysis. This in fact depends on the consideration of various factors such as bearing/rotor flexibility effect, non-Newtonian effect, loading conditions etc. in the analysis.

METHODOLOGY

AVERAGE FLUID-FILM THICKNESS

The coordinate system and geometry of the journal bearing system along with rough surface profile is shown in Fig. 2.1. The local fluid-film thickness h_L at any point inside the clearance space between rough journal and bearing surfaces is sum of the nominal fluid-film thickness (h) and the roughness amplitudes of journal and bearing surfaces (δ_J, δ_b) measured from their mean levels as shown in Fig. 2 may be expressed as

$$h_L = h + \delta_J + \delta_b = h + \delta \quad (2)$$

where $\delta = \delta_J + \delta_b$ is the combined roughness amplitude of two surfaces. The nominal fluid-film thickness (h) is defined as the distance between the mean levels of two surfaces as shown in Fig.2.

The average fluid-film thickness h_T, which is equal to the expected or mean value of local fluid-film thickness (h_L),

$$h_T = E[h+\delta] = \int_{-\infty}^{\infty} (h+\delta)\,\psi(\delta)\,d\delta \quad (3)$$

where $\psi(\delta)$ is the probability density function of combined roughness δ.

As most of the engineering surfaces follow Gaussian height distribution, the present work assumes Gaussian height distribution of the surfaces. The probability density function of this Gaussian distribution is expressed as

$$\psi(\delta) = \frac{1}{\sqrt{2\pi}\sigma} e^{-\delta^2/2\sigma^2} \quad (4)$$

where σ is the combined rms or standard deviation of roughness and is expressed as $\sigma = (\sigma_J^2 + \sigma_b^2)^{1/2}$.

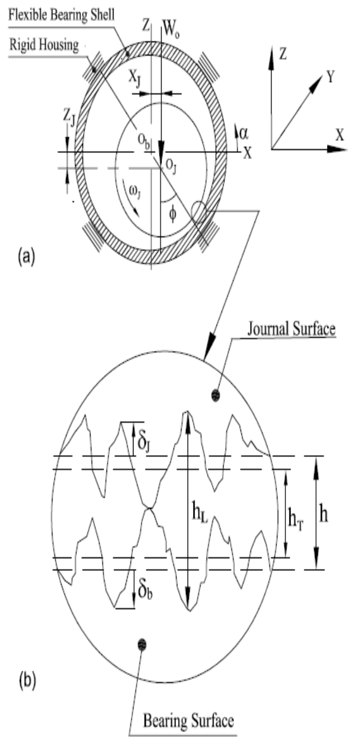


Fig.– Bearing geometry and surface profile

AVERAGE-REYNOLDS EQUATION

For the laminar flow of an incompressible, non-Newtonian lubricant, the modified average Reynolds equation in terms of flow factors, average fluid-film thickness and inertia term can be expressed as (see Appendix-1)

$$\frac{\partial}{\partial\alpha}\left(\frac{\bar{h}^3}{12\bar{\mu}}\phi_x\frac{\partial\bar{p}_f}{\partial\alpha}\right)+\frac{\partial}{\partial\beta}\left(\frac{\bar{h}^3}{12\bar{\mu}}\frac{\partial\bar{p}_f}{\partial\beta}\right)=\frac{\Omega}{2}\frac{\partial\bar{h}_T}{\partial\alpha}+\frac{\Omega}{2\Lambda}\frac{\partial\phi_s}{\partial\alpha}+\Omega\frac{\partial\bar{h}_T}{\partial\bar{t}}-\frac{\bar{R}_e^*}{12\bar{\mu}\Omega}\left[\frac{\partial}{\partial\alpha}\left(\bar{h}_T^2\bar{G}_x\right)+\frac{\partial}{\partial\beta}\left(\bar{h}_T^2\bar{G}_y\right)\right]$$

Where $\bar{\mu}$ is the local average (expected) viscosity corresponds to the local average temperature It may be noted that for the flow of lubricant through the clearance space between two smooth surfaces, the pressure flow factors $\phi_x = \phi_y = 1$ and shear flow factor $\phi_s = 0$. Then the modified Reynolds equation (Eq. (5)) reduces to the generalized Reynolds equation governing the flow of lubricant in the clearance space of a smooth journal bearing.

The parameters $\bar{G}_x$ and $\bar{G}_y$ appearing in last term of Eq. (5) are known as inertia functions in x and y directions.

FINITE ELEMENT FORMULATION

Lubricant flow field is discretized by using four-noded quadrilateral isoparametric elements. The fluid-film pressure variation is assumed to vary linearly over an element. Then, the unknown pressure is expressed in non-dimensional form as

$$\bar{p} = \sum_{j=1}^{n_l^e} \bar{p}_j N_j$$

Where N_j is elemental shape function and n_l^e is number of nodes per element of two-dimensional flow-field solution domain.

Using Galerkin's orthogonality conditions for the modified average Reynolds equation (Eq. (5)) and following the usual assembly procedure for all the elements, the global system equation for the entire lubricant flow field is expressed as

$$\left[\bar{F}\right]\{\bar{p}\} = \{\bar{Q}\} + \{\bar{R}_H\} + \dot{\bar{X}}_J\{\bar{R}_{X_J}\} + \dot{\bar{Z}}_J\{\bar{R}_{Z_J}\} - \{\bar{G}\}$$

where,

$[\bar{F}]$ = Assembled fluidity matrix,

$\{\bar{p}\}$ = Nodal pressure vector,

$\{\bar{Q}\}$ = Nodal flow vector,

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$\{\bar{R}_H\}$ = Column vectors due to hydrodynamic terms,

$\{\bar{R}_{X_J}\},\{\bar{R}_{Z_J}\}$ = Global right hand side vectors due to journal center velocities

$\{\bar{G}\}$ = Global right hand side vector due to inertia functions

For an e^{th} element, the elements of the above matrices are defined as follows:

$$F_{ij}^e = \iint \left[\phi_x \frac{\bar{h}^3}{12\bar{\mu}} \frac{\partial N_i}{\partial \alpha} \frac{\partial N_j}{\partial \alpha} + \phi_y \frac{\bar{h}^3}{12\bar{\mu}} \frac{\partial N_i}{\partial \beta} \frac{\partial N_j}{\partial \beta} \right] |J| \, d\xi d\eta$$

(2.6b)

$$\bar{R}_{Hi}^e = \frac{\Omega}{2} \iint \left(\bar{h}_T + \frac{\phi_s}{\Lambda} \right) \frac{\partial N_i}{\partial \alpha} |J| \, d\xi d\eta$$

(2.6c)

$$\bar{R}_{X_{Ji}}{}^e = \Omega \iint_{A^e} \frac{1}{2} \left[1 + erf\left(\frac{\Lambda \bar{h}}{\sqrt{2}} \right) \right] N_i \cos\alpha \; d\alpha \; d\beta$$

(2.6d)

$$\bar{R}_{Z_{Ji}}{}^e = \Omega \iint_{A^e} \frac{1}{2} \left[1 + erf\left(\frac{\Lambda \bar{h}}{\sqrt{2}} \right) \right] N_i \sin\alpha \; d\alpha \; d\beta$$

(2.6e)

$$\bar{G}_i^e = \frac{\bar{R}_e^*}{12\bar{\mu}\Omega} \iint \bar{h}_T^2 \left(\bar{G}_x \frac{\partial N_i}{\partial \alpha} + \bar{G}_y \frac{\partial N_i}{\partial \beta} \right) |J| \, d\xi d\eta$$

(2.6f)

$$\bar{Q}_i^e = \int_{\Gamma^e} \left\{ \left(\bar{h}^3 \frac{\bar{h}^3}{12\mu_a} \left(\phi_x \frac{\partial \bar{p}}{\partial \alpha} + \phi_y \frac{\partial \bar{p}}{\partial \beta} \right) - \frac{\Omega}{2} \left(\bar{h}_T + \frac{\phi_s}{\Lambda} \right) + \frac{\bar{R}_e^*}{12\bar{\mu}\Omega} \bar{h}^2 \left(\bar{G}_x + \bar{G}_y \right) \right) \right\} N_i d\bar{\Gamma}^e$$

where i, j =1,2 … n_l^e (number of nodes per element).

BOUNDARY CONDITIONS

The boundary conditions relevant to the lubricant flow field (Eq. 2.6a) are as described below:

1. At external boundary of the bearing, the pressure is set to the atmospheric pressure,
   $\bar{p}\,|_{\beta=\mp 1.0} = 0.0$
2. At the supply hole, $\bar{p} = \bar{p}_s = 1$
3. Periodicity is enforced where the bearing wraps around
4. At the trailing edge of the positive region traditional Reynolds boundary conditions are employed, $\bar{p} = \frac{\partial \bar{p}}{\partial \alpha}$ =0.0

5 MEAN VELOCITIES DUE TO PRESSURE INDUCED FLOW

The mean velocity components due to pressure induced flow of lubricant in x and y directions are expressed (Appendix-1) as

$$\bar{U}_m = \frac{\bar{h}^2}{12\bar{\mu}} \phi_x \frac{\partial \bar{p}_f}{\partial \alpha} + \frac{\bar{R}_e^* \bar{h}_T}{12\bar{\mu}\Omega} \bar{G}_x$$

$$\bar{V}_m = \frac{\bar{h}^2}{12\bar{\mu}} \phi_y \frac{\partial \bar{p}_f}{\partial \beta} + \frac{\bar{R}_e^* \bar{h}_T}{12\bar{\mu}\Omega} \bar{G}_y$$

The parameters $\bar{G}_x$ and $\bar{G}_y$ appearing in Eqs. (2.5) and (2.11) are called inertia functions in x and y directions. These inertia functions are expressed in non-dimensional form (Appendix-1) as

$$\bar{G}_x = \bar{h}_T \left[\frac{12}{5}\bar{U}_m - \Omega\left(1 + \frac{\phi_s}{\Lambda}\right) \right] \frac{\partial \bar{U}_m}{\partial \alpha} + \left\{ \bar{U}_m \left[\frac{6}{5}\bar{U}_m - \Omega\left(1 + \frac{\phi_s}{\Lambda}\right) \right] + \frac{\Omega^2}{3} \left[1 + \frac{\phi_s}{\Lambda}\left(2 + \frac{\phi_s}{\Lambda}\right) \right] \right\} \frac{\partial \bar{h}_T}{\partial \alpha}$$
$$+ \frac{\Omega \bar{h}_T}{\Lambda} \left[\frac{2}{3}\Omega\left(1 + \frac{\phi_s}{\Lambda}\right) - \bar{U}_m \right] \frac{\partial \phi_s}{\partial \alpha} + \frac{6}{5}\bar{V}_m \bar{h}_T \frac{\partial \bar{U}_m}{\partial \beta} + \frac{\bar{h}_T}{2} \left[\frac{12}{5}\bar{U}_m - \Omega(1 + \frac{\phi_s}{\Lambda}) \right] \frac{\partial \bar{V}_m}{\partial \beta}$$

(2.12a)

$$\bar{G}_y = \frac{12}{5} \frac{\bar{h}_T}{2} \bar{V}_m \frac{\partial \bar{U}_m}{\partial \alpha} + \frac{\bar{h}_T}{2} \left[\frac{12}{5}\bar{U}_m - \Omega\left(1 + \frac{\phi_s}{\Lambda}\right) \right] \frac{\partial \bar{V}_m}{\partial \alpha} + \frac{\bar{V}_m}{2} \left[\frac{12}{5}\bar{U}_m - \Omega\left(1 + \frac{\phi_s}{\Lambda}\right) \right] \frac{\partial \bar{h}_T}{\partial \alpha}$$
$$- \left(\frac{\bar{h}_T}{2} \frac{\Omega}{\Lambda} \bar{V}_m \right) \frac{\partial \bar{\phi}_s}{\partial \alpha} + \frac{12}{5} \bar{V}_m \bar{h}_T \frac{\partial \bar{V}_m}{\partial \beta}$$

1.1.1 FLUID-FILM VELOCITY COMPONENTS

The flow of the lubricant between two rough surfaces of bearing and journal can be modeled by an equivalent flow model as shown in Fig. 2.4. The equivalent model is defined as two smooth surfaces separated by a clearance equal to the average fluid-film thickness ($\bar{h}_T$). Then, the mean or expected velocity components can be obtained by modifying the Poiseuille and Couette terms in the expression of local velocity components using Patir and Cheng's flow factors. They can be expressed in nondimensional form as

$$\bar{u} = \phi_x \frac{\bar{h}^2}{12\bar{\mu}} \frac{\partial \bar{p}}{\partial \alpha} \left(\bar{z}^2 - \bar{z} \right) + \Omega \bar{z} + \frac{\Omega}{\Lambda \bar{h}_T} \phi_s \bar{z}$$

$$\bar{v} = \phi_y \frac{\bar{h}^2}{12\bar{\mu}} \frac{\partial \bar{p}}{\partial \beta}\left(\bar{z}^2 - \bar{z}\right)$$

(2.13b)

where $\bar{z} = z/h_T$ in the equivalent flow model.

The fluid-film velocity component across the fluid-film is obtained from the continuity equation and it may be expressed in non-dimensional form as

$$\tilde{w} = \bar{w} - \bar{z}\bar{u}\frac{\partial \bar{h}_T}{\partial \alpha} - \bar{z}\bar{v}\frac{\partial \bar{h}_T}{\partial \beta} = -\left[\frac{\partial}{\partial \alpha}\int_0^{\bar{z}} \bar{h}_T \bar{u}\, d\bar{z} + \frac{\partial}{\partial \beta}\int_0^{\bar{z}} \bar{h}_T \bar{v}\, d\bar{z}\right]$$

(2.13c)

1.1.2 NON-NEWTONIAN LUBRICANT MODEL

The available studies indicate that the power law model accurately approximates the physical behavior of many polymer-thickened lubricating oils [5]. Therefore, the power law model is used in the present study to express the nonlinear relationship between shear stress and shear strain rate of non-Newtonian lubricants. The constitutive relation for a simple non-Newtonian lubricant is expressed in non-dimensional form as

$$\bar{\tau} = \bar{m}(\bar{\dot{\gamma}})^n$$

(2.14a)

where n and $\bar{m}$ are known as the power law index and consistency index respectively. The shear strain rate ($\bar{\dot{\gamma}}$) is made independent of direction by considering it as a function of the second strain invariant of shear strain rate, I_2 and is expressed in nondimensional form as

$$\bar{\dot{\gamma}} = \left[\left(\frac{1}{\bar{h}_T}\frac{\partial \bar{u}}{\partial \bar{z}}\right)^2 + \left(\frac{1}{\bar{h}_T}\frac{\partial \bar{v}}{\partial \bar{z}}\right)^2\right]^{1/2}$$

Differentiation of Eqs. (2.13a) and (2.13b) and substitution into the above equation yields

$$\bar{\dot{\gamma}} = \left[\left\{\phi_x \frac{\partial \bar{p}}{\partial \alpha}\frac{\bar{h}^2}{\bar{h}_T \bar{\mu}}\left(\bar{z} - \frac{1}{2}\right) + \frac{\Omega}{\bar{h}_T}\left(1 + \frac{\phi_s}{\Lambda \bar{h}_T}\right)\right\}^2 + \left\{\phi_y \frac{\partial \bar{p}}{\partial \beta}\frac{\bar{h}^2}{\bar{h}_T \bar{\mu}}\left(\bar{z} - \frac{1}{2}\right)\right\}^2\right]^{1/2}$$

The viscosity of non-Newtonian lubricant is described by an apparent viscosity ($\bar{\mu}_a$) and is defined as a function of shear strain rate ($\bar{\dot{\gamma}}$).

$$\bar{\mu}_a = \bar{\tau}/\bar{\dot{\gamma}}$$

1.1.3 TEMPERATURE-VISCOSITY RELATION

The viscosity $\bar{\mu}$ is assumed to be dependent on temperature and is defined by the exponential law [26],

$$\mu = \mu_r \exp\left(-\beta_v (T_f - T_r)\right)$$

where μ_r is the viscosity of the lubricant at reference temperature T_r. The above relation may be expressed in non-dimensional form as

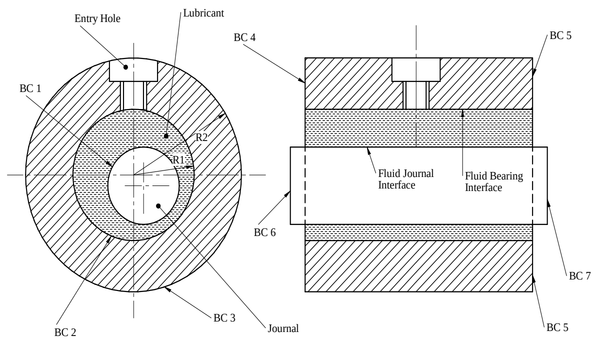

Fig. 3 – Thermal boundaries of big-end bearing system

$$\bar{\mu} = \frac{\mu}{\mu_r} = \exp\left[\bar{\beta}_v\left(\frac{\bar{T}_f + 273.12/T_r}{1 + 273.12/T_r} - 1\right)\right]$$

(2.20a)

where $\bar{\beta}_v$ is the non-dimensional temperature-viscosity coefficient determined from the known values of viscosity at specified temperature for a particular lubricant and is expressed as

$$\bar{\beta}_v = \beta_v\left(T_r(1 + 273.14/T_r)\right)$$

For the non-Newtonian lubricant, the temperature-viscosity relation may be expressed as

$$\bar{\mu} = \bar{\mu}_a \exp\left[\bar{\beta}_v\left(\frac{\bar{T}_f + 273.12/T_r}{1 + 273.12/T_r} - 1\right)\right]$$

(2.20b)

where $\bar{\mu}_a$ is the apparent viscosity of non-Newtonian lubricant. It may be noted that, for Newtonian lubricant $\bar{\mu}_a = 1$ and hence for Newtonian-thermal analysis Eq. (2.20b) reduces to Eq. (2.20a). On the other hand for non-

Newtonian isothermal analysis $\overline{T}_f = 1$ and Eq. (2.20b) reduces to $\overline{\mu} = \overline{\mu}_a$, the apparent viscosity of lubricant.

PERFORMANCE CHARACTERISTICS

After establishing the steady-state matched solutions for the pressure, thermal and elastic deformation fields, the static and dynamic performance characteristics of journal bearing system are computed. In the present work the performance characteristics of hydrodynamic journal bearing is studied under both fully lubricated lubrication conditions of the bearing. For the fully lubricated condition of the bearing, static performance characteristics are computed and presented.

Static Performance Characteristics

The static performance characteristic includes the load carrying capacity of fluid-film pressure, bearing flow, frictional torque at journal surface due to shear viscous of lubricant, mid-film temperature, etc. and dynamic performance characteristics includes fluid-film stiffness and damping coefficients and critical journal mass.

SOLUTION PROCEDURE

A computer program based on the analysis is developed to obtain the performance characteristics of hydrodynamic journal bearings. In case of a non-Newtonian TEHD analysis of a journal bearing, the fluid-film thickness, pressure, temperature, lubricant viscosity and bearing deformation are interdependent variables. Therefore, it requires the simultaneous solution of all the governing equations to establish the matched solution of modified average Reynolds, elasticity, energy and heat conduction equations.

RESULTS-AND DISCUSSIONS

The mathematical models and solution algorithms presented in the previous chapters have been used to compute the static and dynamic performance characteristics of a journal bearing. The influence of surface roughness under the realistic operating conditions of bearing is presented and discussed by considering the bearing deformation, cross-film viscosity variation due to non-Newtonian behavior of the lubricant and rise in fluid-film temperature and fluid inertia. The variations in the bearing performance characteristics are presented for various values of surface roughness parameter, Λ (i.e. for the inverse rms values of combined roughness heights) and roughness orientations such as

- **Transversely oriented roughness pattern (γ = 1/6)**
- **Isotropically oriented roughness patterns (γ=1)**
- **Longitudinally oriented roughness patterns (γ = 6)**

The static performance characteristics of journal bearing are computed for the geometric and operating parameters of the bearing and the results are computed for these nondimensional values unless otherwise mentioned in the text or figures.

Further, since the influence of surface roughness on lubrication process of bearing under the fully lubricated condition is more clearly understandable, the results showing the influence of surface roughness, bearing flexibility, thermal, non-Newtonian behavior of lubricant and fluid-inertia on fully lubricated bearing performances are presented and discussed in the first part. The computed results of performance characteristics are presented and discussed in the following sections.

PERFORMANCE CHARACTERISTICS UNDER FULL LUBRICATION

The results of static performance characteristics of hydrodynamic journal bearing as computed from IHD,THD and TEHD analysis of bearing operating under fully lubricated condition of bearing are presented in Fig 4 These results are discussed in the following sections. As seen from Fig. 4.12, the transverse, isotropic and longitudinal roughness patterns enhance the load carrying capacity of two-sided type rough bearing as compared to the corresponding smooth bearing for both Newtonian and non-Newtonian lubricant cases and this trend is also observed for both fluid inertia-less and inertia solutions. Irrespective of the analysis cases considered, the longitudinal roughness, which restrict the dominant pressure induced flow of lubricant in axial direction, is observed to provide maximum enhancement in the load carrying capacity. Though the transverse roughness pattern with longer asperities in axial direction enhance the pressure induced axial flow, it is observed to provide a marginal enhancement in the load carrying capacity of bearing by restricting the pressure induced circumferential flow of lubricant at converging section of the bearing.

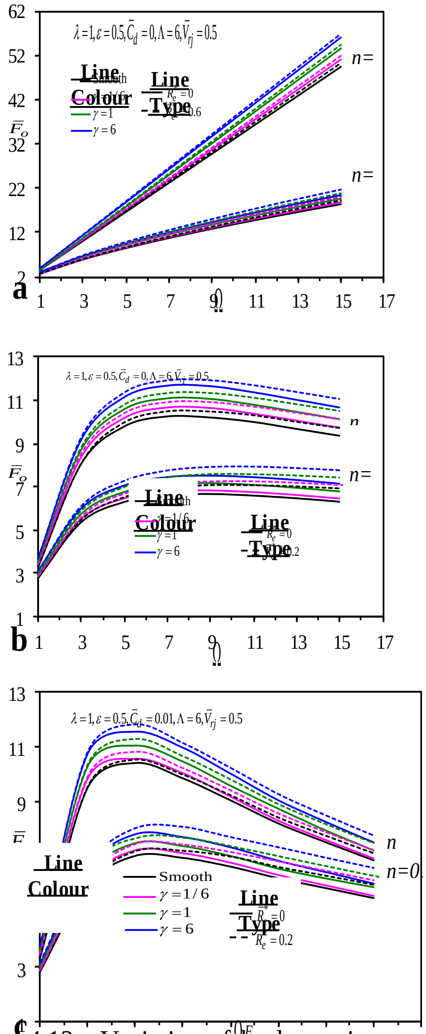

Fig. 4.12 - Variation of load carrying capacity ( ) of rough bearing with non-Newtonian

CONCLUSIONS

The present study investigate the influence of surface roughness on non-Newtonian thermoelastohydrodynamic (TEHD) performance of hydrodynamic journal bearing including fluid-inertia effects. The modified form average Reynolds equation is derived in terms of Patir and Cheng's flow factors and inertia functions to include the surface roughness and fluid-inertia. The mean pressure induced velocity components are also modified to include surface roughness in fluid-inertia analysis. The expressions for pressure derivatives which are required to compute fluid-film dynamic coefficients are also developed.

1. In general influence of fluid-inertia on performance characteristics of bearing is insignificant when viscosity variation due to rise in fluid-film temperature and non-Newtonian behavior of lubricant is not considered. It became considerable when this viscosity variation is considered.
2. The influence of fluid-inertia is to increase the bearing load carrying capacity and lubricant side leakage while the influence of non-Newtonian behavior of lubricant is to reduce these parameters.
3. The transverse, isotropic and longitudinal roughness patterns enhance the load carrying capacity of two-sided type rough bearing as compared to the corresponding smooth bearing for both Newtonian and non-Newtonian lubricant cases and this trend is also observed for both fluid inertia-less and inertia solutions.
4. The longitudinal roughness, which restrict the dominant pressure induced flow of lubricant in axial direction, is observed to provide maximum enhancement in the load carrying capacity.
5. the transverse roughness pattern with longer asperities in axial direction enhance the pressure induced axial flow, it is observed to provide a marginal enhancement in the load carrying capacity of bearing by restricting the pressure induced circumferential flow of lubricant at converging section of the bearing.

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