THERMOHYDRODYNAMIC PERFORMANCE OF A JOURNAL BEARING WITH 3D-SURFACE ROUGHNESS AND FLUID INERTIA EFFECTS

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Abstract – This theoretical work describe the combined influence of surface roughness, thermal and fluid-inertia effects on performance characteristics of hydrodynamic journal bearing. The average Reynolds equation that modified to include the surface roughness, viscosity variation due to temperature rise in lubricant fluid-film and fluid-inertia is used to obtain pressure field in the fluid-film. The matched solutions of modified average Reynolds, energy and conduction equations are obtained using finite element method and appropriate iterative schemes. The effects of surface roughness parameter, roughness orientation, and roughness characteristics of opposing surfaces on circumferential fluid-film pressure distribution, load carrying capacity and stability threshold speed of the bearing are studied by considering thermal and fluid-inertia effects.

***Keywords** - Hydrodynamic journal bearing, surface roughness, fluid inertia.*

I. INTRODUCTION

In the present day technological scenario, the hydrodynamic journal bearings are quite often required to operate under stringent conditions of heavy load and high speed. When the fluid-film thickness in a journal bearing system is of the order of few micrometers, the surface roughness topography has a profound effect on bearing performance. Due to the dependence of lubricant viscosity on temperature, the rise in fluid-film temperature due to viscous shear alters the lubricant viscosity. Thus the viscosity of the lubricant gets reduced which intern results in lower value of the minimum fluid-film thickness. Thus analyses based on assumptions of smooth surfaces and isothermal conditions may not be appropriate. Furthermore, in the analysis of hydrodynamic journal bearings the effect of fluid inertia is generally neglected in view of its negligible contribution compared to viscous forces. However, in the bearings operating under high speed and reduced lubricant viscosity, the flow may considered to be laminar but the fluid inertia forces cannot be neglected. Thus, the combined influence of surface roughness, thermal and fluid inertia effects on the bearing performance characteristics is needed to be considered.

Remarkable progress has been made in the investigation of thermal effects dealing with lubrication mechanism of hydrodynamic bearings based on the smooth surface assumption. Dowson (1) derived a generalized Reynolds equation that includes the variation of viscosity along and across the fluid film. Ferron et al.(2), Khonsari and wang (3) and Sinhasan and Chandrawat (4) simultaneously solved the generalized Reynolds equation along with the energy and conduction equations and studied the effect of viscosity variation due to rise in temperature of the fluid film. The study by Branagan and Barrett (5) illustrates that the effect of consideration of cross film viscosity variation, as opposed to a mean viscosity value based on the average cross film temperature, changes the bearing performance characteristics when the journal temperature deviates from the average fluid film temperature. Sharma, et al. (6) studied the thermal effect on the performance of a slot entry type of non-recessed hybrid journal bearing. This study indicates a significant variation in the static and dynamic performance of a slot entry hybrid journal bearing due to the temperature rise in the lubricant fluid film.

Many mathematical models which represent the lubricant flow field in a rough bearing have also been developed using statistical approaches. Several concepts have been proposed to derive an average Reynolds type equation governing the expected or mean pressure in rough bearings such that the governing equation rely on statistical quantities of surface roughness rather than its specific topography. Among the several concepts, Christensen [7] used the concept of stochastic process theory to derive stochastic Reynolds equation for one-dimensional transverse and longitudinal roughness patterns. However, this stochastic Reynolds equation is limited to one-dimensional ridges oriented either transversely or longitudinally and is not applicable to the 3D area distributed surface roughness. Patir and Cheng [8, 9] introduced a new concept of flow factors for deriving an average Reynolds equation applicable for any general roughness structure. Many investigators used this concept of average Reynolds equation to study the 3D surface roughness effects on lubrication problems. Ramesh et al. [10] investigated the combined influence of 3D surface roughness and thermal effects on performance of a submerged oil journal bearing and they assumed average cross-film viscosity variation in their study. Recently, Nagaraju et al. [11, 12] demonstrated the individual as well as the combined influence of surface roughness and bearing flexibility on the performance of hybrid journal bearing using Patir and Cheng's average flow model. Later, Nagaraju et al. [13] and Sharma et al. [14] modified the average Reynolds equation to include the viscosity variation across and along the fluid-film and investigated non-Newtonian behavior of lubricant and thermal effects on journal bearing performance. They separately studied the combined influence of surface roughness and non-Newtonian behavior of lubricant [13] and the influence of roughness and thermal effects [14] on the performance of hybrid journal bearing systems.

Consideration of fluid inertia effect in the study of journal bearing systems is one of the important areas of lubrication theory and many investigations were made to study its effect on journal bearing performances. Among the few studies related to the fluid inertia effect, Constantinescu and Galetuse [15] evaluated the momentum equations for laminar and turbulent flows by assuming that the shape of the velocity profiles is not strongly affected by the presence of inertia forces. Tichy and Bou-Said [16], Bou-Said and Ehret [17] and Kakoty and Majumdar [18 - 20] used the method of averaged inertia in which the inertia terms are integrated over the film thickness to account the inertia effect in their studies. The above studies [15-20] were mainly based on ideally smooth bearing and journal surfaces assumption. A thorough scan of literature reveals that none of the studies included 3D surface roughness effect to study the combined influence of fluid inertia and thermal effects on journal bearing performance. Hence the present work is undertaken to bridge this gap in the open literature.

In the present work, the modified average Reynolds equation that includes the Patir and Cheng's flow factors, cross-film viscosity integrals, average fluid-film thickness and inertia term is used to study the combined influence of surface roughness, thermal and fluid-inertia on bearing performance. The effects of surface roughness parameter, Λ, roughness orientation, γ, and roughness characteristics of opposing surfaces, $\overline{V}_{rj}$, on circumferential fluid-film pressure distribution, load carrying capacity and stability threshold speed of the bearing are studied by considering thermal and fluid-inertia effects.

II. MATHEMATICAL FORMULATION

A. Average Reynolds Equation

For the inclusion of fluid inertia forces in THD analysis, the average Reynolds equation can be derived from the Navier-Stokes equations and flow continuity equation using Patir and Cheng's [13,14] flow factors. The modified average Reynolds equation in terms of flow factors, cross-film viscosity integrals, average fluid-film thickness and inertia term. It is expressed as

$$\frac{\partial}{\partial\alpha}\left(\phi_x\bar{h}^3\bar{F}_2\frac{\partial\bar{p}}{\partial\alpha}\right)+\frac{\partial}{\partial\beta}\left(\phi_y\bar{h}^3\bar{F}_2\frac{\partial\bar{p}}{\partial\beta}\right)=\Omega\frac{\partial}{\partial\alpha}\left(1-\frac{\bar{F}_1}{\bar{F}_0}\right)\bar{h}_T$$
$$+\frac{\Omega}{\Lambda}\frac{\bar{F}_1}{\bar{F}_0}\frac{\partial\phi_s}{\partial\alpha}+\Omega\frac{\partial\bar{h}_T}{\partial\bar{t}}-R_e^*\bar{F}_2\left[\frac{\partial}{\partial\alpha}\left(\bar{h}_T^3\bar{G}x\right)+\frac{\partial}{\partial\beta}\left(\bar{h}_T^3\bar{G}y\right)\right] \quad (1)$$

Where $\bar{F}_0=\int_0^1\frac{d\bar{z}}{\bar{\mu}}$, $\bar{F}_1=\int_0^1\frac{\bar{z}}{\bar{\mu}}d\bar{z}$ and

$$\bar{F}_2=\int_0^1\frac{\bar{z}}{\bar{\mu}}\left(\bar{z}-\frac{\bar{F}_1}{\bar{F}_0}\right)d\bar{z}$$

The ϕ_x,ϕ_y are known as pressure flow factors and ϕ_s is known as shear flow factor. These pressure flow and shear flow factors can be obtained from Patir and Cheng [13,14]

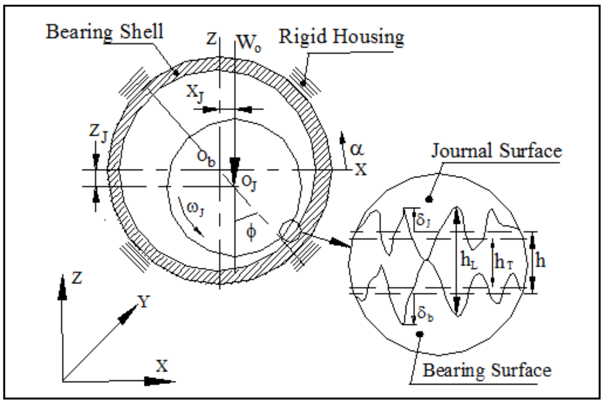


Fig. 1 : Bearing Geometry and Surface Profile.

B. Average Fluid-Film Thickness:

The non-dimensional form of average fluid-film thickness, $\bar{h}_T$, in fully lubricated ($\Lambda\bar{h}\geq 3.0$) and mixed lubricated ($\Lambda\bar{h}<3.0$) regions can be expressed as

$$\bar{h}_T=\begin{cases}\bar{h} & \text{for } \Lambda\bar{h}\geq 3\\ \frac{\bar{h}}{2}\left(1+erf\left(\frac{\Lambda\bar{h}}{\sqrt{2}}\right)\right)+\frac{1}{\Lambda\sqrt{2\pi}}e^{-(\Lambda\bar{h})^2/2} & \text{for } \Lambda\bar{h}<3\end{cases} \quad (2a)$$

Where $\Lambda(=1/\bar{\sigma})$ in the present work is defined as the surface roughness parameter and $\bar{h}$ is the nominal fluid film thickness. It is same as the fluid film thickness obtained for a smooth surface journal bearing and is expressed in non-dimensional form as

$$\bar{h}=1-\bar{X}_J\cos\alpha-\bar{Z}_J\sin\alpha$$
(2b)

where $\overline{X}_J$ and $\overline{Z}_J$ are the journal center co-ordinates and are expressed as

$\overline{X}_J = \varepsilon \sin\phi$ and $\overline{Z}_J = -\varepsilon \cos\phi$
(2c)

C. Temperature - Viscosity Relation:

The temperature viscosity relation is defined by an exponential law and is expressed in non-dimensional form [11] as

$$\overline{\mu} = \frac{\mu}{\mu_r} = \exp\left[\overline{\beta}_v \left(\frac{\overline{T}_f + 27312/T_r}{1 + 27312/Tr} - 1\right)\right] \quad (3)$$

D. Energy Equation:

When the bearing operated in a mixed lubricated condition, the heat is generated due to asperity contact and viscous shear of lubricant. Then, the non-dimensional form of 3-D energy equation used to determine fluid-film temperature is expressed [11] as

$$\overline{h}_T^2 \left[\overline{u}\frac{\partial \overline{T}_f}{\partial \alpha} + \overline{v}\frac{\partial \overline{T}_f}{\partial \beta} + \frac{\tilde{w}}{\overline{h}_T}.\frac{\partial \overline{T}_f}{\partial \overline{z}}\right] =$$
$$\overline{P}_e^* \left(\frac{\partial^2 \overline{T}_f}{\partial \overline{z}^2}\right) + \overline{D}_e \overline{\mu}\left[\left(\frac{\partial \overline{u}}{\partial \overline{z}}\right)^2 + \left(\frac{\partial \overline{v}}{\partial \overline{z}}\right)^2\right] + \Omega^* \overline{h}_T \frac{\overline{p}_c}{d\overline{z}}$$

(4)

where $\overline{p}_c$ is the asperity contact pressure.

E. Heat-Conduction Equations:

The non-dimensional form of a heat conduction equation used to establish the fluid-film bush interface temperature is expressed as

$$\overline{k}_b \left(\frac{\partial^2 \overline{T}_b}{\partial \overline{r}^2} + \frac{1}{\overline{r}}\frac{\partial \overline{T}_b}{\partial \overline{r}} + \frac{1}{\overline{r}^2}\frac{\partial^2 \overline{T}_b}{\partial \alpha^2} + \frac{\partial^2 \overline{T}_b}{\partial \beta^2}\right) = 0 \quad (5)$$

The journal temperature is obtained by considering the journal in thermal equilibrium. It is assumed that the journal is one dimensional heat flow element and its temperature is uniform in the circumferential direction due to its rotation. The heat flux continuity at the fluid-journal interface yields the amount of heat received by the journal as

$$\overline{q}_{j_i} = \int_{-\lambda}^{\lambda}\int_{0}^{2\pi} \frac{k_f R_j}{k_r c\overline{h}} \frac{\partial \overline{T}_f}{\partial \overline{z}}\Big|_{\overline{z}=1.0} d\alpha\, d\beta$$
(6a)

The free convective hypothesis at the two ends of the journal gives the amount of heat flowing out of the journal as

$$\overline{q}_{j_o} = \frac{2h_j A_j}{k_r R_j}\left(\overline{T}_j - \overline{T}_a\right) \quad (6b)$$

Where A_j is cross sectional area of the journal. The journal temperature $\left(\overline{T}_j\right)$ is obtained by equating Eqs. (6a) and (6b).

Equations (4), (5) and (6) are solved using finite element method using appropriate boundary conditions and iterative techniques to get temperature fields in fluid-film, bearing shell and journal respectively. Once the matched solutions for the fluid-film pressure and temperature fields are obtained, the load carrying capacity and stability threshold speed of the bearing are computed using relevant expressions given in Nagaraju et al [11].

III. RESULTS AND DISCUSSION

The results of thermohydrodynamic (THD) performance characteristics of bearing have been computed and presented for the following generally used non-dimensional values of bearing geometric and operating parameters.

Bearing aspect ratio, $\lambda = 1$

Speed parameter, $\Omega = 17.442$

Modified Reynolds number, $\overline{R}_e^* = 0$ and 1.2

Eccentricity ratio, $\varepsilon = 0.5$

Inverse Peclet number $\overline{P}_e^* = 1.6749$

Dissipation number $\overline{D}_f = 0.0072674$

Surface roughness parameter, $\Lambda = 6 - 12$

Surface pattern parameter, γ

For transverse roughness pattern, $\gamma = 1/6$

For isotropic roughness pattern, $\gamma = 1$

For longitudinal roughness pattern, $\gamma = 6$

Further, the effect of surface roughness of opposing surfaces (i.e. bearing and journal surfaces) has been studied by separately considering the roughness only on bearing surface, only on journal surface and on both bearing and journal surfaces. These cases are referred as stationary roughness, moving roughness and two-sided roughness respectively and are accounted by assigning the following values for the variance ratio, $\overline{V}_{rj}$.

1. Stationary roughness i.e. rough bearing and smooth journal,
2. Two-sided roughness, i.e. rough bearing and rough journal,
3. Moving roughness i.e. smooth bearing and rough journal,

The computed results of circumferential fluid-film pressure distribution, load carrying capacity and stability threshold speed of the smooth as well as rough bearings are presented in Figs. 2 to 5 and are discussed in the following

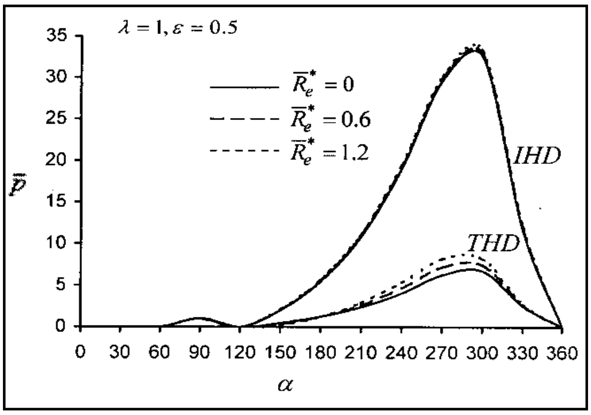


Fig. 2 : Circymferenceal fluid-film pressure distibustion of smooth bearing at its axial mid plane (β =0).

A. *Circumferential fluid-film pressure distribution:*

Figure 2 shows the influence of fluid inertia on circumferential fluid-film pressure distribution of smooth journal bearings at an axial mid-plane (β = 0.0) under isothermal hydrodynamic (IHD) and thermohydrodynamic (THD) analyses. As seen from figure, when the fluid inertia effect is considered (i.e. when $\bar{R}_e^* > 0$), the circumferential fluid-film pressure is observed to be increases in both IHD and THD analyses. Increase in the value of circumferential fluid-film pressure due to inertia effect is observed to be more when thermal effect is considered (THD analysis) as compared to IHD case. The increase in the value of maximum pressure is around 25% in THD analysis when the value of $\bar{R}_e^*$ changes from 0 to 1.2 while it is only around 2% in IHD analysis. These results clearly indicate the interactive influence of thermal and fluid inertia effects.

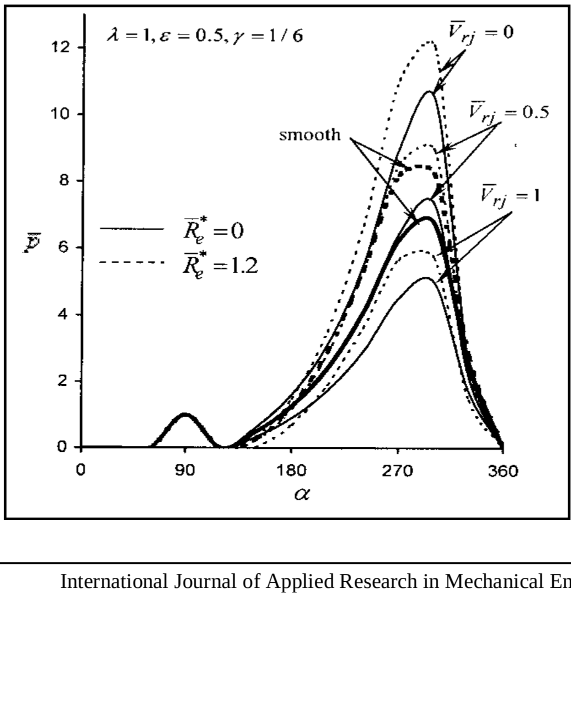
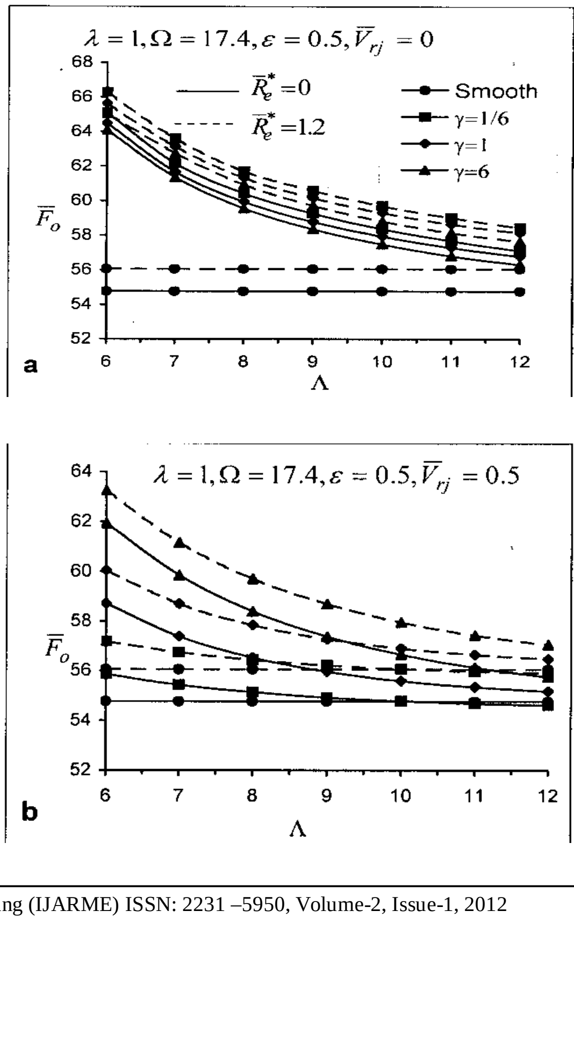


Fig. 3 : Circymferenceal fluid-film pressure distibustion of smooth and rough bearing at its axial mid plane (β =0) in THD analysis.

Figure 3 shows the influence of roughness characteristics of opposing surfaces (i.e. variance ratio, $\bar{V}_{rj}$) on circumferential fluid-film pressure distribution during THD analysis. As seen from Fig. 3, the stationary roughness (i.e. rough bearing and smooth journal, $\bar{V}_{rj} = 0$) provides the enhanced circumferential fluid-film pressure as compared to smooth bearing while the moving roughness (i.e. rough journal and smooth bearing, $\bar{V}_{rj} = 1$) provides reduced fluid-film pressure. The two-sided roughness (i.e. rough bearing and journal) is also provides enhanced circumferential fluid-film pressure but the stationary roughness provides maximum enhancement.

B. *Load Carrying Capacity ($\bar{F}_o$):*

Figure 4 shows the influence of surface roughness parameter (Λ), roughness orientation (γ) and fluid inertia on load carrying capacity of the bearings having stationary, two-sided and moving types of roughness during IHD analysis. In general, the load carrying capacity of all these bearings (i.e. the bearings with stationary, two-sided and moving types roughness) increases as the surface roughness parameter (Λ) decreases (i.e. as the combined roughness height increases).



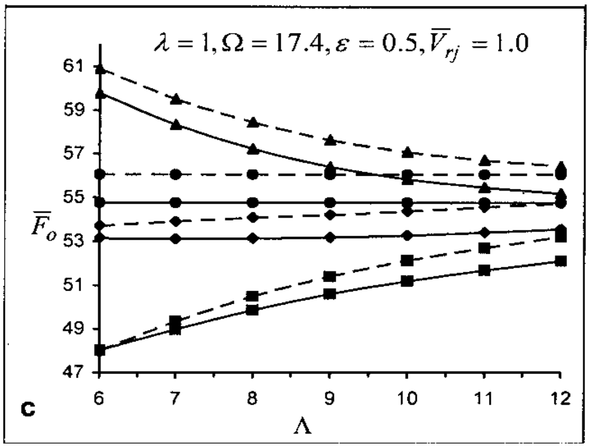


Fig. 4 : Influence of surface riugthness paremeters and fluid inertia effects on load carrying capacity of bearing (a) Statinary rought ness (b) two sided rought ness (c) Moving roughness.

The transverse, isotropic and longitudinal roughness patterns provides enhanced value of load carrying capacity than that of a similar smooth bearing in stationary roughness (Fig. 4(a)) and two-sided type roughness (Fig. 4(b)). Only the longitudinal roughness pattern shows this trend in moving roughness (Fig. 4(c)) whereas transverse and isotropic roughness patterns show the opposite trends. For a stationary type roughness case, the transverse roughness pattern (□=1/6) is seen to provide maximum enhancement in the value of load carrying capacity as compared to isotropic and longitudinal roughness patterns as shown in Fig. 4(a). For the two-sided and moving type roughness, the longitudinal roughness is observed to shows this trend as shown in Figs. 4(b) and 4(c). Further, the load carrying capacity of the bearings increases due to fluid inertia effects and the effects of surface roughness parameters on load carrying capacity of bearing is observed to be almost same in both inertia less and inertia solutions

TABLE I : PERCENTAGE CHANGE IN LOAD CARRYING CAPACITY

$\overline{R}_e^*$	$\overline{V}_{rj}$	γ	$\%\overline{F}_o$	
			IHD	THD
0	0	1/6	17.79	40.92
		1	16.30	32.07
		6	15.52	22.38
	0.5	1/6	2.21	1.99
		1	7.13	7.26
		6	12.62	13.14
	1.0	1/6	-12.37	-28.09
		1	-3.10	-14.66
		6	8.72	4.45
1.2	0	1/6	15.8	30.37
		1	17.01	27.11
		6	16.48	21.63
	0.5	1/6	2.28	2.40
		1	7.11	7.71
		6	12.49	13.49
	1.0	1/6	-14.95	-32.94
		1	-4.55	-16.43
		6	8.25	4.80

Table 1 shows the percentage change in load carrying capacity of rough bearing with respect to corresponding smooth bearing as computed with IHD and THD analyses. During THD analysis, the stationary roughness ($\overline{V}_{rj} = 0$) and transverse roughness pattern ($\gamma = 1/6$) combination provides around 40.92% enhanced load carrying capacity when the fluid inertia effect is not considered while it provides around 30.37% enhanced load carrying capacity when the fluid inertia ($\overline{R}_e^* = 1.2$) effect is considered. However, during IHD analysis these enhancements are limited to 17.79% and 15.8% only. From the results of table 1, the influence of surface roughness is observed to be more pronounced when thermal effects are considered.

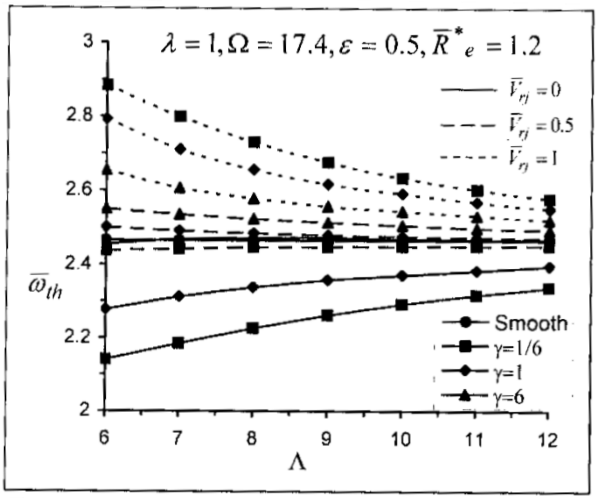


Fig. 5 : Influence of surface roughness papameter on stability threshold speed of bearing during IHD analysis.

C. Threshold Speed ($\overline{\omega}_{th}$):

Figure 5 shows the influence of surface roughness parameter (Λ), roughness orientation (γ), variance ratio ($\overline{V}_{rj}$) and fluid inertia effects on stability threshold speed ($\overline{\omega}_{th}$) of the bearing. The stationary roughness ($\overline{V}_{rj}$ =0) provides the reduced value of stability threshold speed margin $\overline{\omega}_{th}$ as compared to that of a corresponding smooth bearing while the moving roughness ($\overline{V}_{rj}$ =1) provides the enhanced value for all types of roughness orientations. The transverse roughness pattern provides largest enhancement in the value of stability threshold speed margin in moving roughness while it provides largest reduction in stationary roughness. The influence of two-sided roughness ($\overline{V}_{rj}$ =0.5) case on $\overline{\omega}_{th}$ is insignificant.

IV. CONCLUSIONS

The results presented in this work leads to the following conclusions:

1. The circumferential fluid-film pressure and load carrying capacity of the bearing increases due to fluid-film inertia. The increase in their values is more significant when thermal effect is considered.
2. In general, improved load carrying capacity of bearing than that of smooth bearing can be achieved from two-sided type roughness (i.e. rough bearing and rough journal) and stationary type roughness (i.e. rough bearing and smooth journal) for all roughness patterns considered in the present study.
3. Compared to two-sided type roughness, stationary type roughness, especially with transverse roughness pattern provides maximum enhancement in the value of load carrying capacity.
4. Moving type roughness was observed to provide enhanced stability threshold speed of the bearing than that of a smooth bearing but it provides reduced load carrying capacity.
5. Influence of surface roughness on bearing performance is observed to be more pronounced when thermal effects are considered

NOMENCLATURE

Dimensional Parameters

c = Radial clearance, mm

C_p = Specific heat, J. (Kg °K)$^{-1}$

D = Journal diameter, mm

e = Journal eccentricity, mm

h = Nominal fluid-film thickness, mm

h_b, h_J = Heat transfer coefficient of bearing and journal, W. mm^{-2}.K^{-1}

h_L = Local fluid-film Thickness, mm

h_T = Average fluid-film thickness, mm

k_b, k_J = Thermal conductivity of bearing and journal, W.mm^{-1}.K^{-1}

R_J = Radius of journal, mm

R_b = Radius of bearing, mm

p = Pressure, N.mm^{-2}

p_s = Supply pressure, N.mm^{-2}

u, v, w = Fluid velocity components, mm.sec^{-1}

z = Coordinate across fluid-film thickness, mm

δ = Combined roughness height, ($\delta = \delta_J + \delta_b$), μ m

$\lambda_{0.5x,y}$ = 0.5correlation lengths of the x and y profile, μ m

μ = Dynamic viscosity of lubricant, N.sec.m^{-2}

μ_r = Reference viscosity of lubricant, N.sec.m^{-2}

ρ = Density, Kg. m^{-3}

σ = RMS value of combined roughness, $\sqrt{\sigma_J^2 + \sigma_b^2}$, μ m

$erf(x)$ = Error function, $2/\pi \int_0^x \exp(-y^2)dy$

Non-dimensional Parameters

$\overline{D}_e$ = Dissipation number, $\left(\frac{\mu_r}{\rho_f c_{pf}}\right)\left(\frac{c^2 p_s}{\mu_r R_J}\right)\left(\frac{R_J}{c^2 T_r}\right)$

$\overline{F}_0$ = $F_0(\mu_r/h_L)$

$\overline{F}_1$ = $F_1(\mu_r/h_L^2)$

$\overline{F}_2$ = $F_2(\mu_r/h_L^3)$

$\overline{h}, \overline{h}_L, \overline{h}_T$ = $(h, h_L, h_T)/c$

$\overline{p}$ = $(p)/p_s$

$\overline{P}_e$ = Peclet number, $\left(\frac{k_r}{\rho_f c_{pf}}\right)\left(\frac{\mu_r R_J}{c^2 p_s}\right)\left(\frac{R_J}{c^2}\right)$

$\overline{R}_e^*$ = Modified Reynolds number, $\frac{\rho c^2 \omega_j}{\mu_r}$

$\overline{T}$ = Temperature, T/T_r

$\bar{t}$ = Time, $t\omega_j$

$\overline{u}, \overline{v}$ = $(u,v)(\mu_{rR_J} / c^2 p_s)$

$(\overline{V}_{rj}, \overline{V}_{rb})$ = Variance ratio of journal and bearing, $\left((\sigma_j, \sigma_b)/\sigma\right)^2$

$\overline{w}$ = $w(\mu_r R_J / c^2 p_s)(R_J / c)$

$\overline{z}$ = (z/h_L), in a rough bearing

$\overline{z}$ = (z/h_T), in an equivalent flow model

$(\overline{X}_J, \overline{Z}_J)$ = Journal center coordinates, $(X_J, Z_J)/c$

α = Circumferential coordinate, (X/R_J)

$\overline{\alpha}_b$ = Diffusivity Coefficient of bearing, $(k_b / \rho_b C_{pb} \omega_j R_j^2)$

$\overline{\alpha}_j$ = Diffusivity Coefficient of journal, $(k_r R_j^2 / M_j C_{pj} \omega_j c)$

β = Axial coordinate, (Y/R_J)

$\overline{\delta}$ = δ / c

ε = Eccentricity ratio, e/c

γ = Surface pattern parameter, $\dfrac{\lambda_{0.5x}}{\lambda_{0.5y}}$

Λ = Surface roughness parameter, (c/σ)

λ = Aspect ratio, L/D

$\overline{\mu}$ = μ/μ_r

Ω = Speed parameter, $\omega_J \left(\mu_r R_J^2 / c^2 p_s\right)$

Ω^* = Modified speed parameter, $\Omega(f p_s R_j / \rho_f C_{pf} T_r c)$

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