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Circular Bearing Performance Parameters with Isothermal and Thermo-Hydrodynamic Approach Using Computational Fluid Dynamics

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Abstract- The current work aims on the prediction of the effect of journal speed on the pressure and temperature distribution in the circular journal bearing. The model has been simulated using the ANSYS Fluent Software which solves 3-Dimensional Navier Stokes and Energy equation for finding the thermal performance characteristics of the bearing. The lubricant flow has been considered as laminar. The distribution of pressure and temperature throughout the bearing has been obtained by iso-thermal approach and by thermo-hydrodynamic approach. The clearance space for the lubricant film between the journal and bearing surface has been considered as 200 microns. The analysis has been carried out at eccentricity = 0.8 and with speed of journal varying from 2500 rpm to 5500 rpm. It has been observed that the rise in temperature is less in thermo-hydrodynamic analysis as compared to iso-thermal analysis at all the journal speed because of the consideration of viscosity variation along the temperature. Also the pressure, temperature and oil force rises with the increase in the journal speed both in case of iso-thermal and thermo-hydrodynamic analysis.

Index Terms- Circular journal bearin;, ANSYS; iso-thermal; thermo-hydrodynamic.

1. INTRODUCTION

Hydrodynamic Journal Bearing is used to support the rotating shaft extensively in high speed machinery, example turbines, electric motors etc. Circular bearing is the most commonly used profile of these bearings. These bearing support the external load and the presence of thick film of lubricant between the clearance spaces avoid the metal contact of rotating part of machinery with the surface of bearing. During operation of the bearing, due to high journal speed, the variation in temperature along the lubricant film significantly affects the properties of the lubricating oil. Hence it affects the performance of the bearing as the lubricating oil inside the bearing depends upon the pressure and the temperature. The increase in the temperature of the oil film causes the breakage in the layers of the lubricating film which consequently leads to metal contact between the bearing and journal surface. Here, the lubricant film between the journal and the bearing is responsible for low friction and high load carrying capacity of such bearings. The variation in temperature and pressure of lubricant in journal bearings affects the performance of the bearing. Therefore the investigation of bearing performance based on a thermo-hydrodynamic (THD) analysis requires simultaneous solution of the complex equations of flow of lubricant, Reynolds Equation and the energy equation by consideration of conduction and convention effect of the fluid domain. Previously, the researchers investigate the performance of the lubricant by solving the complex equations through Finite Difference Method. With the progress of analysis softwares, many researchers use commercial computational fluid dynamics (CFD) software to solve this complex equation. CFD codes provides a solution to flow problems by solving the full Navier-Stokes equations instead of Reynold's Equation. Moreover, the CFD packages are applicable to the complex geometries to solve the complex simultaneous governing equations. The most of the researches does the THD analysis by solving the Reynolds and Energy equation only in the two dimensions by neglecting the variation of pressure and temperature across the film thickness. CFD software solves these equations in three dimensions by considering its variation along film thickness also. First a remarkable work on Thermo-hydrodynamic study of journal bearing was done by Hughes and Osterle [1]. The authors have found out a relation between viscosity as a function of temperature and pressure of the lubricant inside the journal bearing for adiabatic conditions. The authors have presented a numerical example to illustrate the method. Gertzos et al. [2] have investigated journal bearing performance with a Non-Newtonian fluid i.e. Bingham fluid considering the thermal effect. Chauhan et al. [3] have presented a comparative study on the thermal characteristics of elliptical and offset-halves journal bearings. It has been reported by the authors that the offset-halves bearing run cooler when compared with elliptical bearing with minimum power loss and good load capacity. Ouadoud et al. [4] have considered two numerical methods, the finite volume is used to determine the pressure, temperature and velocity distributions in the fluid film, the displacement field radial is obtained by the finite element method respectively by using numerical simulation, Computational Fluid Dynamic ”CFD” and Fluid Structure Interaction ”FSI”. The authors have analyzed the influence of the operating conditions on the pressure, temperature and displacement. Cupillard et al. [5] presented an analysis of a lubricated conformal contact to study the effect of surface texture on bearing friction and load carrying capacity using computational fluid dynamics. The work mainly concentrates on a journal bearing with several dimples. The authors have reported that the coefficient of friction can be reduced if a texture of suitable geometry is introduced and the same can be achieved either in the region of maximum hydrodynamic pressure for a bearing with high eccentricity ratio or just downstream of the maximum film for a bearing with low eccentricity ratio. Sahu et al. [6] have carried out THD analysis of a journal bearing as a tool. The authors have presented 2-dimensional distribution and 3-dimensional pressure of the lubricating film. Basri and Gethin [7] have investigated the thermal aspects of various non-circular journal bearings using adiabatic model. Hussain et al. [8] presented a work is on the prediction of temperature distribution in noncircular journal bearings: two-lobe, elliptical and orthogonally displaced bearings. The authors have presented results for these geometries including the conventional circular bearing. Liu et al. [9] have used computational fluid dynamics and fluid structure interaction method to study rotor-bearing system. The authors have investigated the dynamic response of the system with both the rigid and flexible bodies and made a assumption of isothermal behaviour for all the models. Further, the author have considered the cavitation within the fluid film and reported that the elastic deformation and dynamic unbalanced loading of the rotor have significant effects on the position of its locus. Li et al. [10] have presented a new method for studying the 3D transient flow of misaligned journal bearings in a flexible rotor-bearing system. The results presented by the author indicate that the bearing performances are greatly affected by misalignment and method presented by them can effectively predict the transient flow filed of the system under consideration. Panday et al. [11] have done the numerical unsteady analysis of thin film journal bearing using ANSYS fluent software and calculated the various bearing parameters like pressure distribution, wall shear stress at different eccentricities ratios.

Still the correlation of thermo-hydrodynamic analysis and FSI is incomplete. Also the effect of thermal analysis on the pressure distribution is reported by few of the researchers using CFD as a tool but the variation of temperature has not been analyzed. An effort has been made to analyze the thermohydrodynamic effect on variation of temperature in the lubricant of the journal bearing.

2. GOVERNING EQUATIONS

2.1. *Journal Bearing Geometry*

The journal which rotates inside the bearing generates the relative rotational velocity with each other. At stable position, the journal takes an eccentric position with respect to bearing. The amount of eccentricity is adjusted by the generated pressure inside the converging oil film balances the external load.

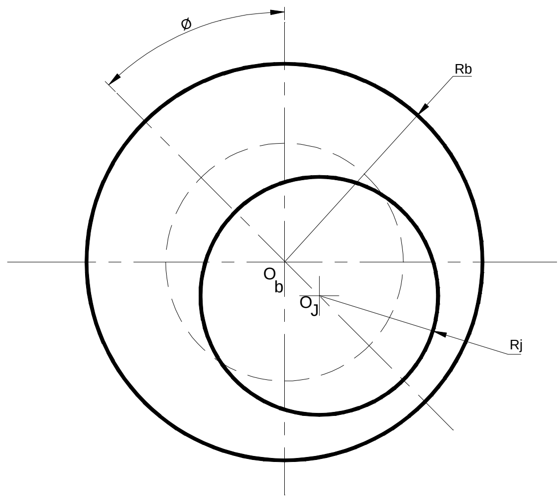

Fig. 1 Schematic diagram of circular journal bearing

Fig.1. represents the schematic diagram of circular journal bearing. It has been clear that the generation of pressure inside the journal bearing depends on the eccentricity, the viscosity of the lubricant, bearing internal surface geometry and the clearance space between the journal and the bearing. The pressure in the lubricant film depends upon the film thickness. The equation for calculation of film thickness for a circular journal bearing can be written as [4]:

$$h(\theta) = C + e\cos\theta = C(1+\varepsilon\cos\theta) \qquad (1)$$

where,C represents the radial clearance space between journal and bearing. ε represent the eccentricity ratio of the journal bearing. θ gives the value of film thickness along the circumferential direction, being measured from the maximum film thickness.

2.2. *Navier Stokes Equation*

The basic lubrication theory is based on the solution of a particular form of Navier-Stokes equations as described below. In this work the viscosity is kept constant and the pressure obtained is called isothermal pressure whereas when temperature dependent viscosity is taken into consideration, then the pressure obtained is called thermal pressure. Similarly same convention is followed in the case of temperature [4].

$$\rho\frac{\partial u}{\partial t} = -\frac{\partial p}{\partial x} + div(\eta \, grad \, u) + S_{Mx}$$
$$\rho\frac{\partial v}{\partial t} = -\frac{\partial p}{\partial y} + div(\eta \, grad \, v) + S_{My}$$
$$\rho\frac{\partial w}{\partial t} = -\frac{\partial p}{\partial z} + div(\eta \, grad \, w) + S_{Mz} \qquad (2)$$

2.3. *Energy Equation*

The energy equation in mechanics of viscous thin films is a particular form of shape differs from an author to another depending on changes made to it. The energy equation written for an incompressible Newtonian fluid and the simplified calculation domain is as follows [4]

$$\frac{\partial}{\partial t}(\rho C_P T) + \nabla(\rho \vec{v} C_P T) = \nabla(K.\nabla T) + \qquad (3)$$

where, Q_V represents volumetric heat source; C_P and K represents the Specific heat and thermal conductivity of the lubricant respectively.

2.4. *Thermo-hydrodynamic Consideration*

The effect of temperature is considered on the properties of the lubricating oil for calculation of bearing parameters. For this User Defined Function code is developed in the ANSYS software. By this method a governing function is coded which would control the variation of property of the fluid with respect to pressure or temperature or both. The following relation has been used for the variation of viscosity as a function of pressure and temperature. This equation has been adapted from the reference [1]

$$\mu = \mu_0 \, e^{\alpha(p-p_0)} \, e^{\beta(t-t_0)} \qquad (4)$$

where, α is the Barrus viscosity pressure index constant and β is the temperature viscosity index constant.

3. COMPUTATIONAL PROCEDURE

CFD package of ANSYS Fluent software has been used to solve the Navier Stokes equation. Three dimensional simulation model of circular journal bearing has been developed by modeling the fluid domain available between the journal and the bearing surface. The clearance for the lubricant to support the shaft between journal and bearing surface has been considered as 200 microns. The eccentricity ratio is taken as 0.8 with attitude angle of 56°. The dimensions of the plain journal bearing used in this simulation along with the lubricant properties are given in the Table 1. The analysis has been carried out at different journal speed varying from 2500 to 5500 rpm.

Table 1 : Operating Parameters

Shaft radius	50 mm
Internal bush radius	50.2 mm
Bearing length	100 mm
Lubricant density	850 Kg/m^3
Specific heat of the lubricant	2000 J/Kg ^oC
Thermal conductivity of lubricant	0.13 W/m ^oC
Convective heat transfer coefficient	100 $W/m^{2o}C$
Clearance	0.2 mm
Inlet lubricant temperature	30 ^oC
Rotational Speed	2500 to 5500 rpm
Initial viscosity	0.04986 Ns/m^2
Pressure Viscosity Index (α)	2.3 e–8
Temperature Viscosity Index (β)	0.034

The steady state condition has been assumed. The fluid flow is considered as laminar. The equations have been solved without assuming the body forces. A constant load has been assumed to be applied on the journal surface. For meshing, a hexahedral structure mesh is used. Name selection for various boundaries like fixed wall for bearing surface and moving wall for journal surface is done in ANSYS Meshing. SIMPLE algorithm is chosen for the solution of velocity pressure coupling equation. A first order upwind scheme is used for the solution of energy and momentum equation. Gravity forces have been considered on the journal bearing and axis of rotation has been taken at the eccentricity value of the journal. The convergence criteria of 10^{-6} have been used for all the residual terms.

4. RESULTS AND DISCUSSION

The bearing performance parameters have been carried out by two different approaches. In the first approach, the viscosity of the lubricant has been considered as constant. This approach is named as Iso-thermal approach that is there would not be any change in the viscosity of the lubricant due to pressure and temperature rise inside the oil film. In second approach, the viscosity of lubricant is varying. This approach is named as thermo-hydrodynamic approach as the viscosity of the lubricant varies as the function of pressure and temperature

The pressure distribution curve along the circumference of circular bearing at 3500 rpm has been shown in Fig. 2. It has been observed that the maximum pressure rise in Iso-thermal analysis is 9.962 MPa while 6.653 M Pa was observed in case of thermo-hydrodynamic Analysis. The variation of maximum pressure built in the oil film with different journal speed ranging from 2500 rpm to 5500 rpm has been plotted in Fig. 3. It has been observed from the graph that there was steep rise in pressure during iso-thermal analysis as compared to thermal analysis

(A) Iso-thermal Analysis (B) Thermo-hydrodynamic Analysis

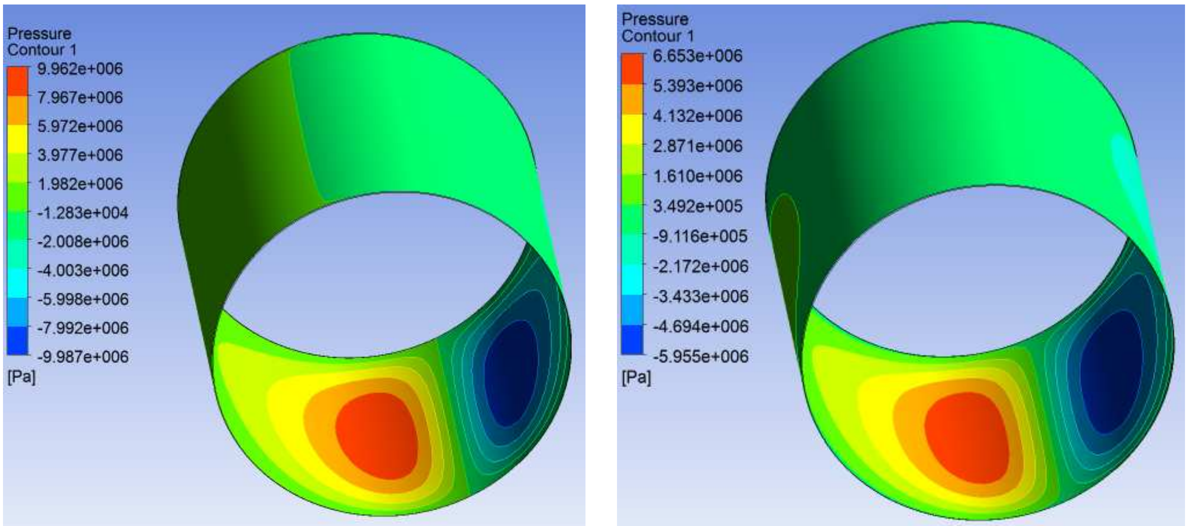

Fig. 2. Comparison of Pressure Distribution Profile for Isothermal and thermo-hydrodynamic analysis at journal speed = 3500 rpm

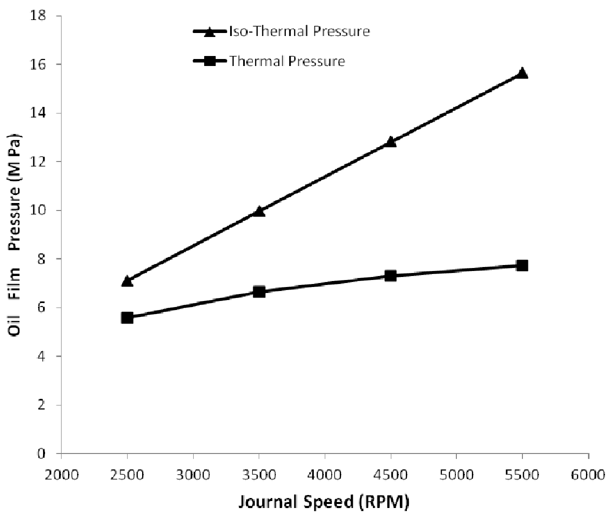

Fig. 3. Variation of oil film pressure with the increase in journal speed from 2500 rpm to 5500 rpm for iso-thermal and thermal analysis

Fig. 4 represents the temperature of the lubricant film along the periphery of circular journal bearing at eccentricity = 0.8 with journal speed = 5500 rpm. The maximum rise in temperature of the oil film has been observed as 95K in case of iso-thermal analysis but the maximum temperature rise has been observed as only 45 K in thermal analysis. The variation of maximum temperature rise in the oil film with different journal speed ranging from 2500 rpm to 5500 rpm has been plotted in Fig. 5. During Iso-thermal analysis, the rise in temperature has been occurred from 29 K to 95 K when journal speed increases from from 2500 rpm to 5500 rpm whereas during thermo-hydrodynamic analysis, this rise in temperature has been changed from 19 K to 45 K only. Therefore, there was steep rise in temperature during iso-thermal analysis as compared to thermal analysis.

The variation of wall shear stress at 5500 rpm of journal speed has been represented in Fig. 6. It has been found the maximum value of wall shear stress was 6.354e+4 and 2.94e+4 for iso-thermal and thermo-hydrodynamic analysis respectively.

The variation of oil film forces on the journal speed has been presented in Fig. 7. It has been observed that the 20 N oil film force at 2500 rpm and it rises to 45 N when the journal speed varies from 2500 to 5500 rpm in case of iso-thermal analysis. The

A. Iso-Thermal Analysis

B. Thermo-hydrodynamic Analysis

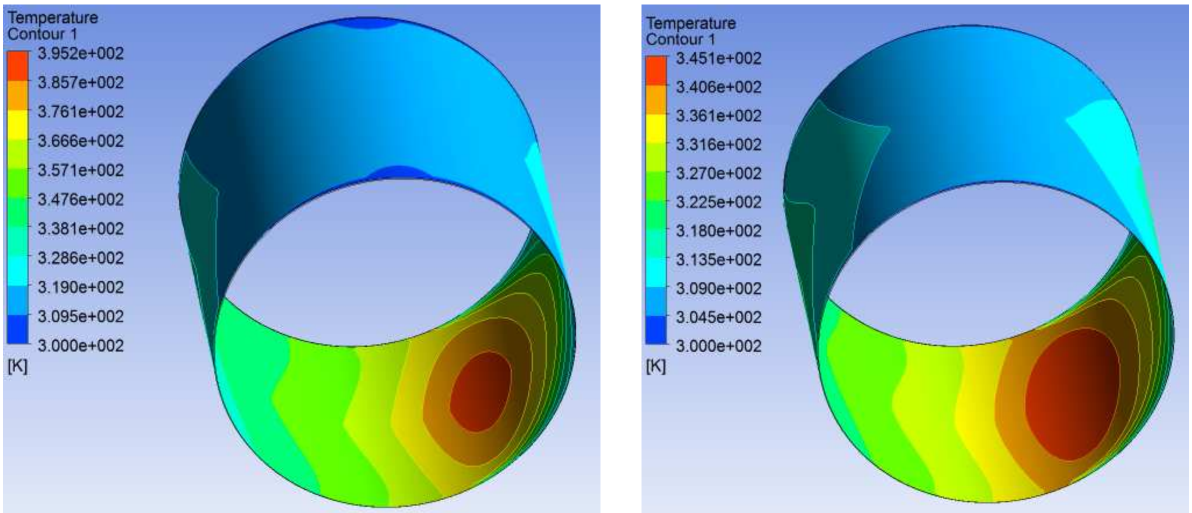

Fig. 4. Comparison of Temperature Distribution Profile for Isothermal and thermo-hydrodynamic analysis at journal speed = 5500 rpm

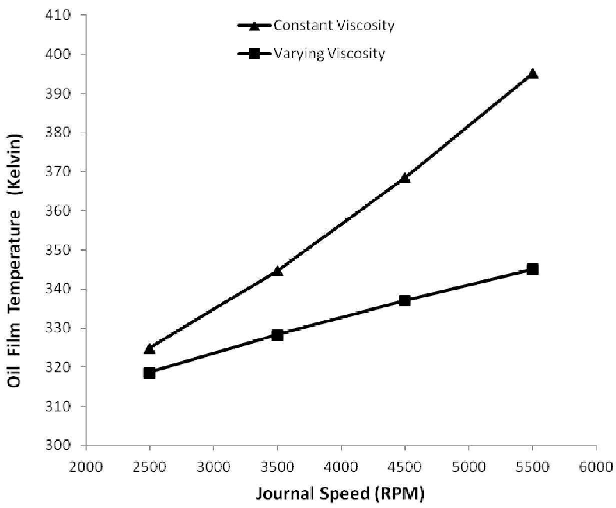

Fig. 5. Variation of oil film temperature with the increase in journal speed from 2500 rpm to 5500 rpm for iso-thermal and thermo-hydrodynamic analysis

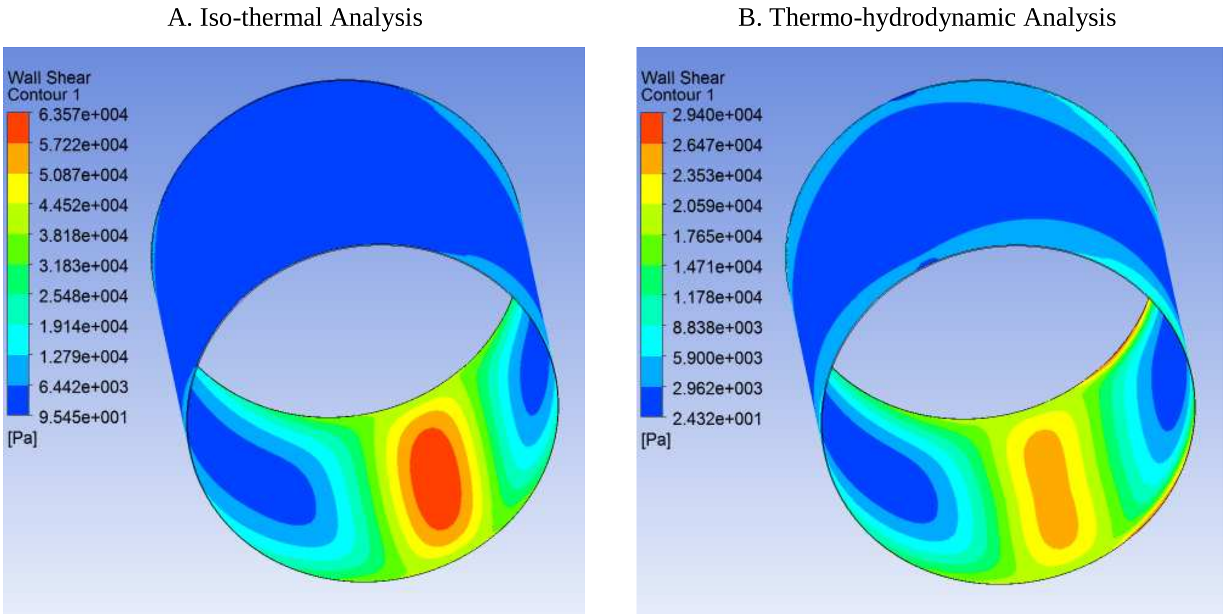

Fig. 6. Comparison of Wall Shear Stress for Isothermal and thermo-hydrodynamic analysis at journal speed = 5500 rpm

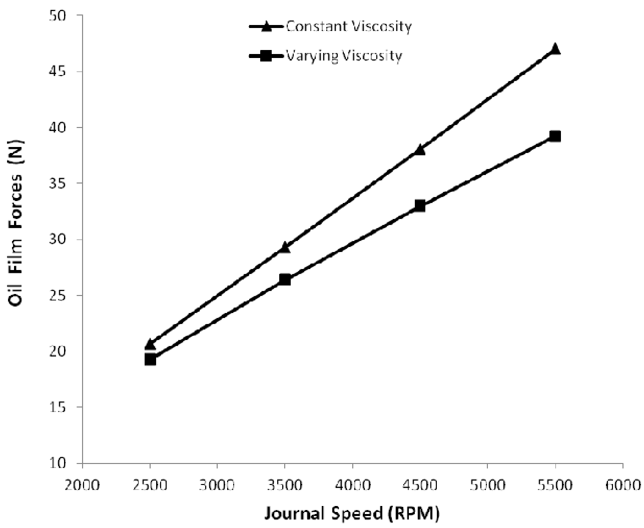

Fig. 7. Variation of oil film forces with the increase in journal speed from 2500 rpm to 5500 rpm for iso-thermal and thermo-hydrodynamic analysis

rise of oil film force for thermo-hydrodynamic analysis has been found from 19 N to 36 N when the journal speed varies from 2500 rpm to 5500 rpm.

5. CONCLUSIONS

Thermo-hydrodynamic analysis for circular journal bearing has been carried out using the application of Computational Fluid Dynamics. The bearing performance parameters such as pressure, temperature, oil film forces and wall shear stress has been evaluated at a journal speed varying from 2500 rpm to 5500 rpm taking eccentricity ratio = 0.8, radial clearance = 200 micron and L/D ratio of 1. It has been found that when viscosity is kept constant the temperature rise is more in the lubricant and the maximum pressure obtained is also high. But it does not represent real life time scenario as when temperature increases, viscosity of lubricant decreases which affects the load carrying capacity of bearing. Therefore obtaining the bearing performance characteristics by keeping constant viscosity may gives wrong prediction about the bearing. So the present analysis may be helpful in prediction of bearing performance parameters in actual working conditions and may help in increased life of the bearing.

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