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Thermo-Hydrodynamic Analysis of Journal Bearing To Find Out Equivalent Temperature

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Abstract

The current work aims on determine the operating temperature of hydrodynamic journal bearing by using numerical method. To derive an alternative method to find out the operating temperature of journal bearing.The software using in numerical method is MATLAB Software.The paper introduces some problems of hydrodynamic journal bearing based on Thermo-hydrodynamic Analysis with special attention kept on the calculation of operating temperature of bearing that are based on developed programs of numerical simulation of bearing operation. Numericalmodelling is developed to analyse the bearing for the parameters like L/D ratio, viscosity and speed. When a bearing operates at high speed, the heat generated due to large shearing rates in the lubricant film raises its temperature which lowers the viscosity of the lubricant and in turn affects the performance characteristics. Thermo-hydrodynamic (THD) analysis is therefore carried out to obtain the realistic performance characteristics of the bearing. The average film temperature is also an important aspect is to be analysed during the operationNumerical Analysis is done by taking l/d ratio = 1 , applied load 9KN and speed 3600 r.p.m . Then results obtained by analytical method using Raimondi n Byod chart and numerical method i.e. by MATLAB are compare.
Keywords: Hydrodynamic Journal Bearing, Average Oil Film Temperature, Thermo-Hydrodynamics, Heat Generation

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I. 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. In order to investigate the influence of oil supply conditions on the performance of a journal bearing. The supply conditions considered were oil supply temperature, supply pressure, groove length and groove location. To carry out this study, the hydrodynamic pressure distribution inside the bearing has been determined using a mass-conserving cavitation model with realistic supply conditions. The energy equation and the heat conduction equation have been used for the determination of oil film and bush temperature distributions [1]. At low eccentricity tests the maximum temperature occurred at the unloaded lobe of the bearing, with the downstream groove contributing poorly to bearing cooling. As eccentricity increased, a temperature increase in the loaded lobe of the bearing was observed, along with a temperature decrease in the unloaded lobe. At high eccentricities the downstream groove was found to contribute significantly to bearing cooling. Shaft temperature and oil outlet temperature did not seem to be significantly affected by increasing load [2]. From parametric study it is found that the temperature of the fluid film raises due to frictional heat thereby viscosity, load capacity decreases. Increased shaft speed resulted in increased load carrying capacity, bush temperature, flow rate and friction variable [3]. The bearing has obvious effect on the temperature of lubricating oil film, maximal film pressure, leakage flow rate of ends and misalignment moment. The surface roughness and viscosity - pressure effect of lubricant have a great influence on the lubrication performance of misaligned journal bearing with larger eccentric ratio [4]. The temperature influence on the journal bearings performance is important in some operating cases, and that a progressive reduction in the pressure distribution, in the load capacity and attitude angle is a consequence of the increasing permeability. The Reynolds equation of thin viscous films was used taking into account the oil leakage into the porous matrix, by applying Darcy’s law to determine the fluid flow in the porous media [5]. To analyze the pressure distribution in hydrodynamic journal bearing for various loading conditions and various operating parameters. The space between the shaft and the bearing is called lubrication gap and is filled with lubricant. Journal bearing test rig is used to test the 140 mm diameter and 70 mm long bearing. Test bearing is located between two antifriction bearings. The bearing was loaded mechanically [6]. An experimental work was conducted to determine the temperature distribution around the circumference of a journal bearing. A journal diameter of 100mm with a ½ length-to-diameter ratio was used. Temperature results for different radial loads and speeds were obtained [7].The evolution of temperature with time in a deep-groove ball bearing in an oil-bath lubrication system is studied both experimentally and analytically. The test apparatus is a radially-loaded ball bearing instrumented to measure the frictional torque as well as the transient temperature of the outer race, oil and housing. Simulations results indicate that higher rotational speed, oil viscosity and housing cooling rate lead to the larger temperature gradient and thermally-induced preload in ball bearings [8]. Thermo-hydrodynamic analysis of circular journal bearing has been simulated by using Computational Fluid Dynamics approach. This approach solves the three dimensional Navier-stokes equation to predict the bearing performance parameters such as the pressure and temperature of the lubricant along the profile of the bearing. The CFD technique has been applied through ANSYS Fluent software [9].

II. ANALYTICAL METHOD

Raimondi and Byod chart is used in analytical method to find out the operating temperature or equivalent temperature of journal bearing.
The following parameters are taken for the analytical solution of a journal bearing:

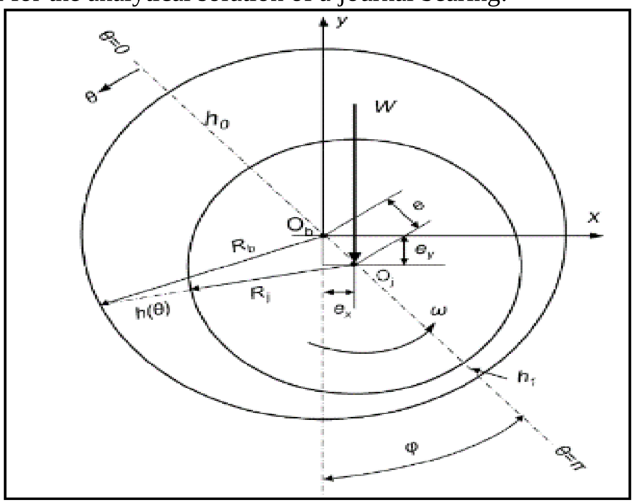

Fig. 1: Schematic Diagram of Journal Bearing

Table - 1
Parameter Involved In Modelling

Length of the bearing (L)	*80 mm*
Radius of Shaft (R_s)	*80 mm*
Radial Clearance (C)	*0.06mm*
L/D Ratio	*1*
Eccentricity ratio(ε)	*0.44*
RPM	*3600*
Lubricant density (ρ)	*840 Kg/m3*
Load	*9 KN*

Design procedure for analytical method using Raimondi and Byod chart: [10]
Data: d = 80 mm; l =80 mm; c = 0.06 mm; F = 9kN; n = 3600rpm = 60 rps; SAE 40 oil; To = 65OC; Ti=45oC.

1) Step 1: calculate value of Nominal Bearing Pressure (in projected area in journal) p= F /l d = 9 x1000 /80 x 40 = 1.4062 MPa
2) Step 2: Now for SAE grade 40 oil, the value of viscosit (μ) can be find out from the graph.. μ = 30 cP at 65^{o}C for SAE 40 oil from graph [10].
3) Step 3: calculate value of Sommerfeld number (S) using petroff's equation S= $(r/c)^2$ * (μn/p) = $(40/0.06)^2$ * (30*10^{-3} *60/ 1.4062*10^6)=0.5688
4) Step 4: For the calculated value of Sommerfeld number and l/d =1 we can find out the temperature rise variation from Raimondi and Boyd chart.(10). For S = 0.5688 and l /d = 1, Tvar = 50 from Raimondi - Boyd chart.
5) Step 5:Now calculate the change in temperature by using following equation, Tvar = $\lambda C_H(\Delta T/p)$ SO ΔT = 46.4 ^{0}C
6) Step 6: Average operating temperature or Equivalent temperature is calculated as Tav = Ti + 0.5 ΔT = 45 + 0.5 x 46 = 68.2^0 C Second iteration:
7) Step 7: Now at the calculated operating temperature or for SAE 40 oil the new value of viscosity is taken from the chart. At Tav = 68.2^{o}C, μ = 25 Pa.s from chart.

8) Step 8: Again using step 3,4 and 5 calculate the new value of change in temperature we get, S = 0.474, for this Tvar = 47.4 from chart , calculated value of $\Delta T = 44^{o}C$.
9) Step 9: New value of average operating temperature, $Tav = Ti + 0.5\ \Delta T = 45+0.5x44 = 67^{o}C$. Hence equilibrium temperature will be about 67 degree C.

III. NUMERICAL MODELLING

The friction torque may be calculated by the two methods. One is the analytical method in which the Raimondi and Boyd charts are used to find out the friction power. In another method it may be found with the help of mathematical modelling which is described below.

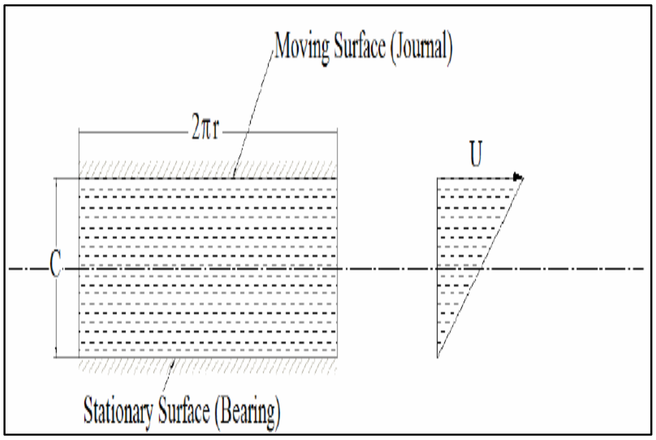

Fig. 2:

The length is $2\pi r$ and the width is L into the plane of the paper. Also, the film thickness is equal to the clearance i.e. $h = C$.
U= 2π N' and the force F= τ A
N' = journal speed
τ = shear stress acting on the fluid
$A = 2\pi rL$, area of the journal surface
Assuming constant coefficient of viscosity of the fluid and from Newton's law.

$$\tau = \mu \frac{U}{h} \qquad (1)$$

h is film thickness at a point

$$h = c(1+\varepsilon\cos\theta) \qquad (2)$$

$0^0 \leq \theta \leq 360^0$

$$\tau = \mu \frac{2\pi\, r\, N'}{h} \qquad (3)$$

$$F = \sum_{\theta=0}^{\theta=360} \frac{4\mu\Pi^2 r^2 N' L}{c(1+\varepsilon\cos\theta)} \qquad (4)$$

Frictional torque (T_f) = F. r

$$= Hg = \sum_{\theta=0}^{\theta=360} \frac{4\mu\Pi^2 r^3 N' L}{c(1+\varepsilon\cos\theta)} \qquad (5)$$

Governing Equation

When heat generates in the oil due to frictional torque, that heat comes to the surface of the bearing and dissipates through convection to the environment and it depends on the property of bearing material.

$$H_d = \frac{KxAx\Delta T}{(1+\alpha)} \qquad (6)$$

Whenever the heat dissipated and heat generated are not getting equal, till that the temperature of bearing surface rises. This rise in temperature attains a maximum value at which steady state is reached, that is heat generated and heat dissipated gets equal. To calculate the maximum temperature corresponding to steady state, an algorithm named "Average film temperature algorithm" is developed and is presented in figure

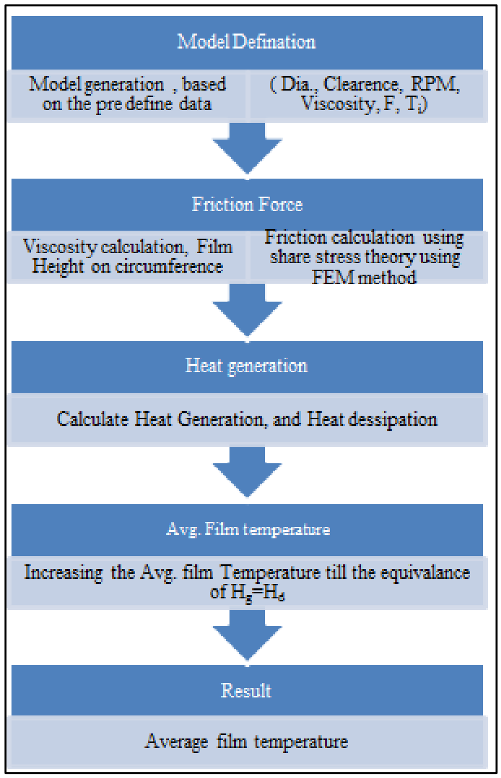

Fig. 3: Algorithms Adopted For Calculating Average Oil Film Temperature

IV. RESULT

Appling above methodology at the parameters used in analytical modelling we get the numerical modelling solution using MATLAB,

1) Heat generation= 19.6W
2) Heat dissipation= 20W
3) Average Film temperature= 62 Oc

The Equivalent temperature or Average operating temperature of a journal bearing is calculated by two different method one is Analytical method and other is numerical method. By using analytical method with the help of Raimondi and Byod charts the value of operating temperature is 67 0C . Using second approach i.e. numerical method the value of average oil film temperature found to be 62oC.

$$\text{\% Deviation of the result obtain by the analytical method} = \frac{(67-62)}{67} \times 100 == 7.4\%$$

% Deviation comes from the analytical method is within a acceptable limit so this numerical method can reduce the dependency of Raimondi and Byod chart while calculating the average oil film temperature .Variation of Film height , friction force and friction power over the circumference of journal bearing is given by the following graphs using the numerical modelling approach.

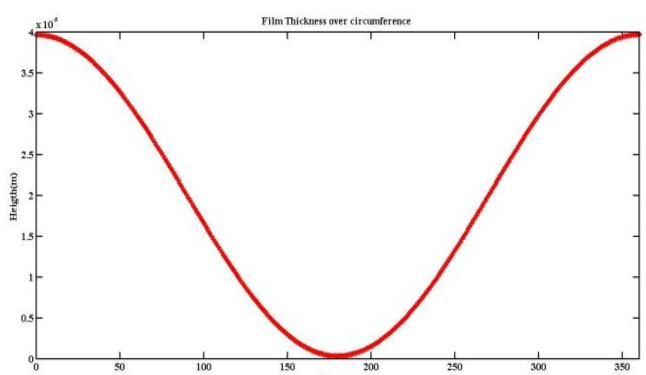

Fig. 4: Film Thickness Variation across Periphery

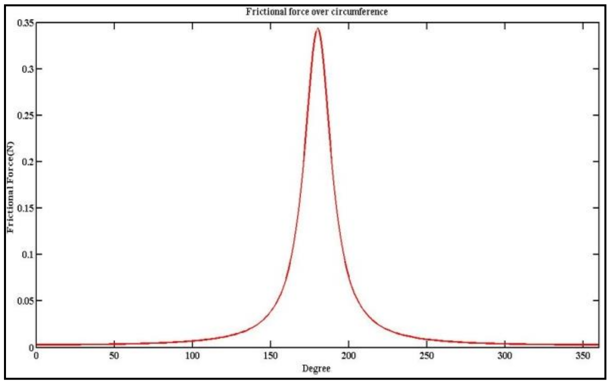

Fig. 5: Friction Force Variation across Periphery

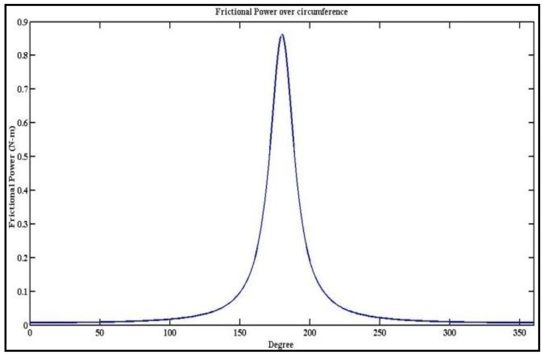

Fig. 6: Friction Power Calculation across Periphery

The following graphs show the oil film temperature at different r.p.m. for the different L/D Ratio for SAE 30 oil.

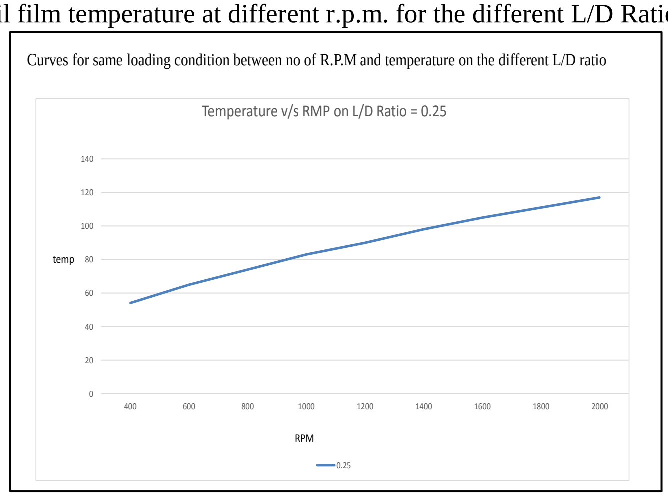

Fig. 7: Temperature Variation on Changing RPM at L/D ratio 0.25

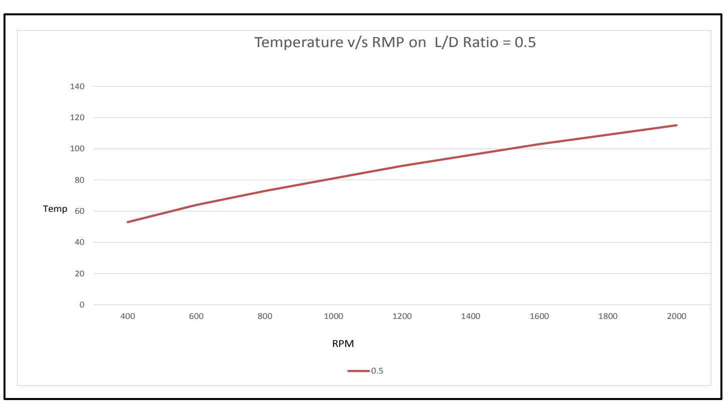

Fig. 8: Temperature Variation on Changing RPM at L/D Ratio 0.5

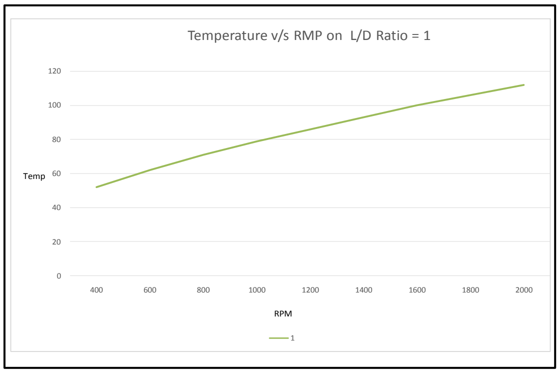

Fig. 9: Temperature Variation on Changing RPM at L/D ratio 1

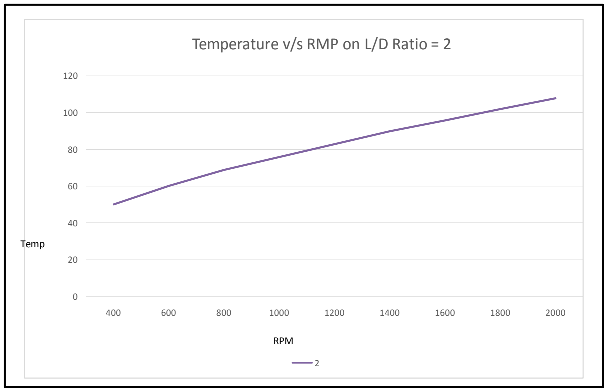

Fig. 10: Temperature Variation on Changing RPM at L/D Ratio 2

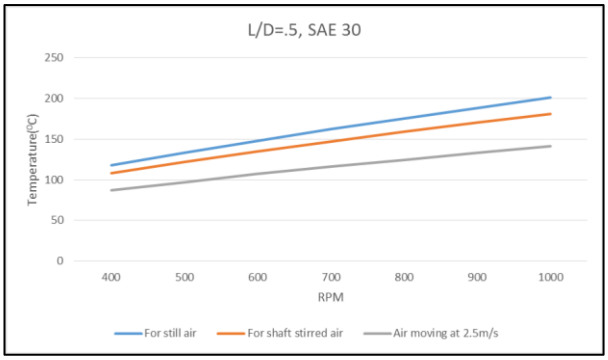

Fig. 11: Temperature Variation on Changing RPM at L/D Ratio 0.5 at Three Different Conditions

V. CONCLUSION

Thermo-hydrodynamic analysis for a journal bearing has been carried out using the application of MATLAB Software using Numerical method. The bearing performance parameter such as, pressure, friction forces, Heat generated, Heat dissipated and Average oil film Temperature has been evaluated at L/D ratio 1 , load 9KN and speed 3600 r.p.m. The Average oil film temperature comes out to be 62oCusing numerical method that is deviated by the 7.4% from the analytical result, which is under the acceptance limit. It has been concluded that this numerical method can reduce the dependency of Raimondi and Boyd chart while calculating the average oil film temperature.It has been found that when L/D ratio increases the rise in oil film temperature decreases for the same RPM at constant viscosity. The present analysis may be helpful in prediction in actual working condition temperature and may help in increased life of the bearing. It is analyses from the results that the temperature rise for still air condition is more

As compare to the two other working condition that is for moving air at 2.5 m/s and shaft stirred air for the same L/D ratio and Same RPM for the constant viscosity. So it is suggested that for less temperature rise we should not operate the bearing under the still air condition .Bearing should be operated at moving air condition.

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