Thermodynamic Analysis of an Journal Bearing with Dimple Textures on Bearing Surface using CFD

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***Abstract*— In current era machineries in modern industries are operated a high speed and subjected to various static and dynamic loads. Apart from this journal bearing journal bearing are also used rural area in form of machines used for irrigation purpose, in Agricultural Machines like Tractors, irrigation Pumps, Harvester, etc. In such application hydrodynamic journal bearings are employed. When bearing subjected to critical loads heat is generated due to presence of friction which alters the lubricating film which ultimately affects the performance characteristic of bearing and the rise in temperature lower the viscosity of the bearing lubricant. In order to analyze performance characteristic of thin film lubrication thermo hydrodynamic analysis should be carried out by the means of three dimensional computational fluid dynamics analysis. The Navier Stokes compressible equations for 3D model were adopted to simulate the flow. The simulation is been carried out by using GAMBIT software in which the bearing is simulated a different boundary condition and parametric analysis has been carried to in order to validate the obtained results shows good agreement with the compared results and are within acceptable limits.**

***Keywords-* Component; Viscosity, Temperature, CFD, Bearing**

I. Introduction

A bearing is a system of machine elements whose function is to support an applied load by reducing friction between the relatively moving surfaces. As modern science and technology is developing, one discovered that the lubricant is affected by the gap of bearing. The interaction degree between the lubricant and solid surface influences the lubrication property of the bearing clearly. A thin film lubricated journal bearing is to develop positive pressure by virtue of relative motion of two surfaces separated by a fluid film. Thin film bearings are those in which although lubricant is present, the working surfaces partially contact each other at least part of the time. These bearings are also called Boundary Lubricated Bearings. Recently, following the progress in computer technology, many researchers began to use commercial computational fluid dynamics (CFD) programs in their investigations. The main advantage of CFD code is that it uses the full Navier–Stokes equations and provides a solution to the flow problem, whereas finite difference codes are based on the Reynolds equation. The results obtained by the two approaches are therefore likely to differ. Moreover, the CFD packages are applicable in very complex geometries. The performance accuracy of journal bearing is predicted and judged on the basis of oil flow pattern within the bearing surface at various temperatures. Therefore thermo hydrodynamic analysis is generally requires simultaneous solution of equations governing the flow of lubricant.

II. Literature Review

In 1958 Hughes et al. perform remarkable work on Thermo hydrodynamic analysis of journal bearing and develop a relation between viscosity, temperature and pressure. Moreover also reveal that the viscosity is the function of temperature and pressure of the lubricant. In 2007 Nassab and Moayeri perform thermal analysis for axial groved and the impact of temperature distribution has been seen similarly Mishra, Wang et al.2007 and Jiao did a monotonous remarkable work in this field. Gertzos et al. 2008 employed electro-rheological fluids (ERFs) grease and the electro-rheological (ER) and magneto-rheological (MR) to analyze fluids behavior of the non-Newtonian fluid flow and present the bearing characteristics result in Raimondi and Boyd charts form, and which can be utilized for analysis and design of journal bearings lubricated with Bingham fluids. Manshoor et al 2013 conducts 3D CFD analysis in order to evaluate performance characteristics of a thin film lubricated plain journal bearing along with several numerical analyses and compare three turbulent models with different aspect ratio of L/D. moreover design parameters such as static pressure, wall shear stress & dimensionless load carrying capacity are simulated in transient mode.Deligant et al. 2011 compute friction losses occurred in turbocharged journal bearing at various oil entrance temperatures and rotational speeds. The obtained result shows good agreement with the CNAM experimental result in which special turbocharger test rig

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integrated with a torque meter. Dimitrios et al. 2011 applied CFD and FEM in MRF magneto rheological (MRF) fluid journal bearing and bearing characteristics such as, attitude angle, eccentricity, oil flow & friction coefficients are determined. And found that on operating bearing at high eccentricity ratio specific procedure and efficient meshing is required. Omidbeygi and Hashemabadi 2013 investigated hydrodynamic characteristics of Magneto rheological (MR) fluid flow within an eccentric annulus under tangential flow and the obtained results shows good agreement on comparing with 2D cfd result and at last they concludes that on increasing magnetic field and eccentricity ratio, viscosity significantly increases which results in enhancement of yield stresses and total torque with in inner cylinder for rotation. Gengyuan et al. 2014 perform Numerical analysis of journal bearing lubricated by water. A comparative analysis is also been perform in order to validate the result on considering water and oil properties separately at different rotational speed and given load.Zhang et al. 2014 provide an efficient method for influential the stiffness coefficients of hydrodynamic plain journal bearings lubricated by water and also consider the cavitation effect. The result shows that on implementing stiffness coefficients and load for bearings with different relative clearances, different length-to-diameter ratios and different rotational speeds the stability can be improved

III. MATHEMATICAL MODELING

Governing Equation

The basic lubrication theory is based on the solution of a particular form of Navier-Stokes equations shown below.

$$\rho\frac{Du}{Dt}=\frac{\partial p}{\partial x}+div(\eta\cdot grad\cdot u)+S_{Mx} \quad (1)$$

The generalized Reynolds Equation, a differential equation in pressure, which is used frequently in the hydrodynamic theory of lubrication, can be deduced from the Navier-Stokes equations along with continuity equation i.e.

$$\frac{\partial u}{\partial x}+\frac{\partial v}{\partial x}+\frac{\partial w}{\partial x}=0 \quad (2)$$

under certain assumptions. The parameters involved in the Reynolds equation are viscosity, density and film thickness of the lubricant. However, an accurate analysis of the hydrodynamics of fluid film can be obtained from the simultaneous equations of Reynolds equation, the energy equation i.e.

$$\rho\frac{Di}{Dt}=-p\,div\vec{V}+div(k\cdot grad\cdot T)+\Phi+S_i \quad (3)$$

and the equations of state i.e.

$$p=\rho RT \text{ and } i=C_V T \quad (4)$$

Reynolds in his classical paper derived the equation which is true for incompressible fluid. Here the generalized Reynolds equation will be derived from the Navier-stokes equations and the continuity equation after making a few assumptions which are known as the basic assumptions in the theory of lubrication. The equation which will be derived will be applicable to both compressible and incompressible lubricants.

The assumptions to be made are as follows-

Inertia and body force terms are negligible as compared to viscous and pressure forces.

1) There is no variation of pressure across the fluid film i.e. $\frac{\partial p}{\partial y}=0$
2) There is no slip in the fluid-solid boundaries (as shown in the figure below).
3) No external forces act on the film.
4) The flow is viscous and laminar (as shown in the figure below).
5) Due to the geometry of fluid film the derivatives of u and w with respect to y are much larger than other derivatives of velocity components.

The height of the film thickness '**h**' is very small compared to the bearing length '**l**'. A typical value of h/l is about $\mathbf{10^{-3}}$.

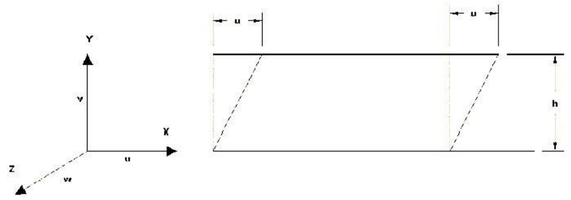


Figure 1 Fluid film depicting the Shear

With the above assumptions, the Navier-Stokes equations are reduced to-

$$\frac{\partial p}{\partial x}=\frac{\partial}{\partial y}\left(\eta\frac{\partial u}{\partial y}\right), \qquad \frac{\partial p}{\partial z}=\frac{\partial}{\partial y}\left(\eta\frac{\partial w}{\partial y}\right) \quad (5)$$

As 'p' is function of x and z, above equations can be integrated to obtain generalized expressions for the velocity gradients. The viscosity η is treated as constant.

$$\frac{\partial u}{\partial y}=\frac{1}{\eta}\frac{\partial p}{\partial x}y+C_1 \quad (6),$$

$$\frac{\partial w}{\partial y}=\frac{1}{\eta}\frac{\partial p}{\partial z}y+C_2 \quad (7)$$

Where C1 and C2 are constants Integrating above equations once more we get

$$u=\frac{1}{2\eta}\frac{\partial p}{\partial x}y^2+C_1y+C_3 \quad (8),$$

$$w=\frac{1}{2\eta}\frac{\partial p}{\partial z}y^2+C_2y+C_4 \quad (9)$$

Where C3 and C4 are constants

The boundary conditions of 'u' and 'w' are

i) At y=0, $u=u_b$, $w=w_b$

ii) At y=h, $u=u_a$, $w=w_a$

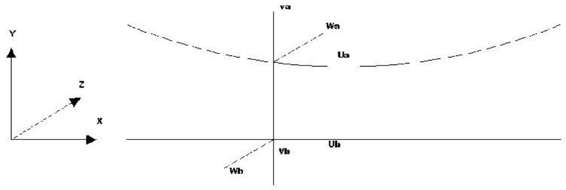


Figure 2 Fluid film depicting the velocity components

Imposing above boundary conditions we get-

$$u=\frac{1}{2\eta}\frac{\partial p}{\partial x}y\left(y-h\right)+\left(\frac{h-y}{h}\right)u_b+\frac{y}{h}u_a \qquad (10)$$

$$w=\frac{1}{2\eta}\frac{\partial p}{\partial z}y\left(y-h\right)+\left(\frac{h-y}{h}\right)w_b+\frac{y}{h}w_a \qquad (11)$$

Now using above expressions of velocity components in continuity equation i.e. Eqn (4) we get

$$\begin{aligned}\frac{\partial(\rho v)}{\partial y}&=-\frac{1}{2}\left\{\int_0^h\frac{\partial}{\partial x}\left[\frac{\rho}{\eta}\frac{\partial p}{\partial x}y\left(y-h\right)\right]dy+\int_0^h\frac{\partial}{\partial z}\left[\frac{\rho}{\eta}\frac{\partial p}{\partial z}y\left(y-h\right)\right]dy\right\}\\&-\int_0^h\frac{\partial}{\partial x}\left[\rho\left\{\left(\frac{h-y}{h}\right)u_b+\frac{y}{h}u_a\right\}\right]dy-\int_0^h\frac{\partial}{\partial x}\left[\rho\left\{\left(\frac{h-y}{h}\right)u_b+\frac{y}{h}u_a\right\}\right]dy\\&-h\frac{\partial p}{\partial t}\frac{\partial}{\partial x}\left(\frac{\rho h^3}{12\eta}\frac{\partial p}{\partial x}\right)+\frac{\partial}{\partial z}\left(\frac{\rho h^3}{12\eta}\frac{\partial p}{\partial z}\right)=\frac{\partial}{\partial x}\left[\frac{\rho\left(u_a-u_b\right)h}{2}\right]\\&+\frac{\partial}{\partial z}\left[\frac{\rho\left(w_a-w_b\right)h}{2}\right]+\rho\left(v_a-v_b\right)-\rho u_a\frac{\partial h}{\partial x}-\rho w_a\frac{\partial h}{\partial z}+h\frac{\partial \rho}{\partial t}\end{aligned} \qquad (12)$$

Now imposing the boundary conditions

i) At y=0, $v=v_b$,

ii) At y=h, $v=v_a$,, we get

$$\begin{aligned}\rho\left(v_a-v_b\right)&=-\frac{1}{2}\left\{\int_0^h\frac{\partial}{\partial x}\left[\frac{\rho}{\eta}\frac{\partial p}{\partial x}y\left(y-h\right)\right]dy+\int_0^h\frac{\partial}{\partial z}\left[\frac{\rho}{\eta}\frac{\partial p}{\partial z}y\left(y-h\right)\right]dy\right\}\\&-\int_0^h\frac{\partial}{\partial x}\left[\rho\left\{\left(\frac{h-y}{h}\right)u_b+\frac{y}{h}u_a\right\}\right]dy-\int_0^h\frac{\partial}{\partial x}\left[\rho\left\{\left(\frac{h-y}{h}\right)u_b+\frac{y}{h}u_a\right\}\right]dy\\&-h\frac{\partial p}{\partial t}\frac{\partial}{\partial x}\left(\frac{\rho h^3}{12\eta}\frac{\partial p}{\partial x}\right)+\frac{\partial}{\partial z}\left(\frac{\rho h^3}{12\eta}\frac{\partial p}{\partial z}\right)=\frac{\partial}{\partial x}\left[\frac{\rho\left(u_a-u_b\right)h}{2}\right]\\&+\frac{\partial}{\partial z}\left[\frac{\rho\left(w_a-w_b\right)h}{2}\right]+\rho\left(v_a-v_b\right)-\rho u_a\frac{\partial h}{\partial x}-\rho w_a\frac{\partial h}{\partial z}+h\frac{\partial \rho}{\partial t}\end{aligned} \qquad (13)$$

Integrating the Eqn (13) we get-

$$\begin{aligned}&\frac{\partial}{\partial x}\left(\frac{\rho h^3}{12\eta}\frac{\partial p}{\partial x}\right)+\frac{\partial}{\partial z}\left(\frac{\rho h^3}{12\eta}\frac{\partial p}{\partial z}\right)=\\&\frac{\partial}{\partial x}\left[\frac{\rho(u_a-u_b)h}{2}\right]+\frac{\partial}{\partial z}\left[\frac{\rho(w_a-w_b)h}{2}\right]+\rho(v_a-v_b)-\rho u_a\frac{\partial h}{\partial x}-\rho w_a\frac{\partial h}{\partial z}+h\frac{\partial \rho}{\partial t}\end{aligned} \qquad (14)$$

The two terms of left hand side of the Eqn (14) is due to pressure gradient and first two terms of the right hand side of the Eqn (14) is due to surface velocities. These are called Poiseuille and Couette terms respectively.

Now if we impose the following boundary conditions-

$$w_a=w_b=0,\ v_a=u_a\frac{\partial y}{\partial x},\ \mathrm{U}=\frac{u_a+u_b}{2} \qquad (15)$$

From Eq. (15)

$$\frac{\partial}{\partial x}\left(\frac{\rho h^3}{12\eta}\frac{\partial p}{\partial x}\right)+\frac{\partial}{\partial z}\left(\frac{\rho h^3}{12\eta}\frac{\partial p}{\partial z}\right)=U\frac{\partial}{\partial x}\left(\rho h\right) \qquad (16)$$

If the fluid property ρ does not vary, as in the case of incompressible lubricant we can write Eqn (16) as follows

$$\frac{\partial}{\partial x}\left(\frac{h^3}{12\eta}\frac{\partial p}{\partial x}\right)+\frac{\partial}{\partial z}\left(\frac{h^3}{12\eta}\frac{\partial p}{\partial z}\right)=U\frac{\partial h}{\partial x} \qquad (17)$$

If we assume the bearing is of infinite length, then and the Eqn (20) becomes

$$\frac{\partial}{\partial x}\left(h^3\frac{\partial p}{\partial x}\right)=12\eta U\frac{\partial h}{\partial x} \qquad (18)$$

A Journal bearing designed to support a radial load is the most familiar of all bearings. The sleeve of the bearing system is wrapped partially or completely around a rotating shaft of journal.

Now if we consider velocity of the journal as 'U', then as per Eqn (17) the governing equation of the journal bearing becomes

$$\frac{\partial}{\partial x}\left(h^3\frac{\partial p}{\partial x}\right)=6\eta U\frac{\partial h}{\partial x} \qquad (18)$$

Using polar coordinates-

$$x=R\theta\ and dx=Rd\theta$$

The equation (18) becomes—To find the solution of above equation _h' has to be expressed in terms of 'θ' and the final expressions come as- h= C+ e cosθ

Or, h= C(1+ε cosθ)

Where, ε = e/C and known as eccentricity ratio.

Integrating above equations we get expression for pressure distribution as

$$\mathrm{p}=\frac{6\eta \mathrm{UR}}{\mathrm{C}^2}\left[\int\frac{\mathrm{d}\theta}{\left(1+\varepsilon\cos\cos\theta\right)^2}-\frac{\mathrm{h_m}}{\mathrm{C}}\int\frac{\mathrm{d}\theta}{\left(1+\varepsilon\cos\cos\theta\right)^3}\right]\mathrm{C}_1 \qquad (19)$$

At p=0, θ=0 and At p=0 θ=2π we get equation

$$p=\frac{6\eta \mathrm{UR}\varepsilon}{\mathrm{C}^2}\frac{\left(2+\varepsilon\cos\cos\theta\right)\sin\sin\theta}{\left(2+\varepsilon^2\right)\left(1+\varepsilon\cos\cos\theta\right)^2} \qquad (20)$$

The following parameters are calculated for a fixed eccentricity: the load carrying capacity (W), the friction force (Fr), and the friction coefficient (f). The load carrying capacity is calculated from the integration of the pressure acting on the shaft

$$\mathrm{W}=\mathrm{L}\sqrt{\left(\int_0^{2\pi}\mathrm{p}\cos\cos\theta\mathrm{R}d\theta\right)^2+\left(\int_0^{2\pi}\mathrm{p}\sin\sin\theta\mathrm{R}d\theta\right)^2} \qquad (21)$$

The friction force is calculated from the following equation-

$$\mathbf{F}_r=L\int_0^{2\pi}\tau\mathbf{R}^2d\theta \qquad \mathbf{(22)}$$

The friction coefficient is the ratio of friction force and load carrying capacity. It is expressed as-

$$f = \frac{F_r}{W} \quad (23)$$

Performance of any bearing is judged by calculating the friction coefficient and comparing the value among the considered bearings.

TABLE I BEARING PARAMETERS

Length of the bearing (L)	133mm
Radius of Shaft (Rs)	50mm
Radial Clearance (C)	0.145mm
Eccentricity ratio(ε)	0.61
Angular Velocity (ω)	48.1 Rad/sec
Lubricant Density (ρ)	840 Kg/m^3
Viscosity of the lubricant (η)	0.127 Pas Kg/m.Sec

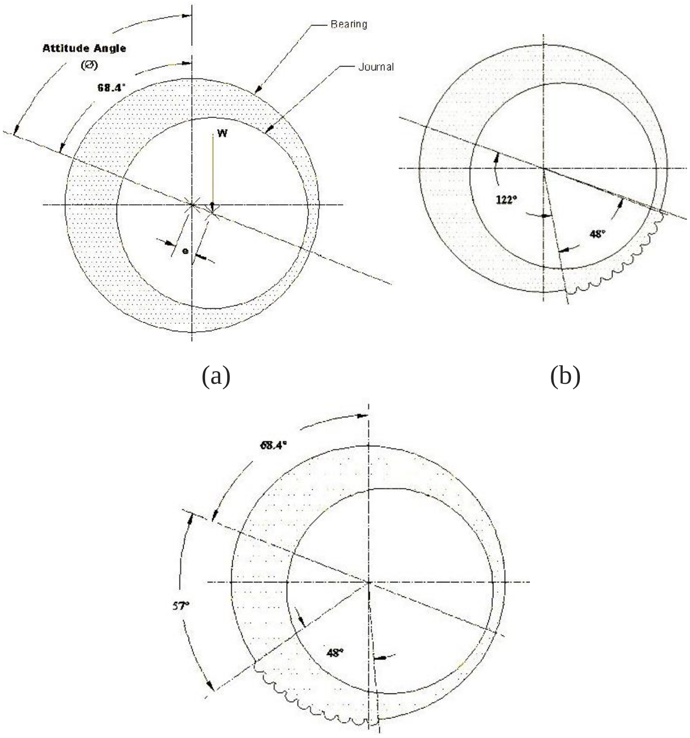
According to the above topological data other derived data would be like—

Radius of Bearing (Rb): (Rs + C) = 50.145mm

Attitude angle (φ):68.4^0 . (as per reference [6])

Eccentricity (e):(ε × C)=(0.61 × 0.145) =0.08845mm.

Now details for cavitation model are as follows as per reference [6].



(a) (b)

Fig 3 Schematic diagram of smooth journal bearing, dimple starting angle 57° and dimple starting angle 122°.

IV. METHODOLOGY

A 3-D simulation model is developed by using the CFD package FLUENT 6.3. The pre-processor Gambit 2.3 is used for the grid generation. The individualism of the problem examined here (the clearance size is very small compared to journal diameter and length) enforces to adopt only hexahedral cells because the use of tetrahedral cells leads to an enormous number of cells. Eight divisions were used across the journal-bearing film, 360 divisions were used in the circumferential direction, and 25 divisions were used in the axial direction. The number of cells is 72000. One hundred and twenty divisions were used in the circumferential direction initially, giving unacceptable results for the angle of the maximum pressure.

The dimensions of the journal bearing used in this simulation are: diameter 100mm and radial clearance of 0.145 mm. The C/D value is 0.00145, denoting a clearance in the upper limit of acceptable. The C/D value is 1×10^3, denoting a clearance in the lower limit of acceptable values. The results of the two above cases are almost identical, denoting that the results are independent of the value of clearance. The viscosity is 0.0127 Pa s while the density is 840 kgm^3. The rotational speed 48.1 rad/sec, the viscous model is set to Standard k-epsilon with standard wall function.

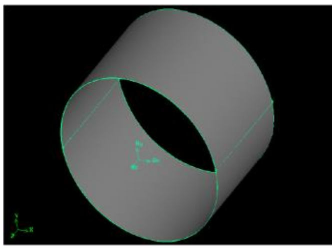


Figure 4 Model geometry

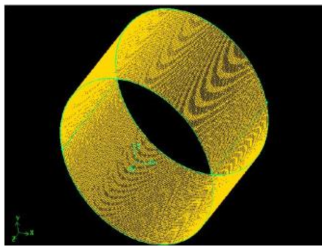


Figure 5 Mesh Model

V. RESULTS AND DISCUSSIONS

Transient response of a dynamically loaded thin film lubricated journal bearing at different configuration is simulated at unsteady condition. The results obtained for a bearing with the following parameters are presented here: journal diameter D = 100 mm; Radial clearance= 0.147mm; journal speed 48.1rad/sec; viscosity of lubricant μ = 0.0127 kg/m-s. The transient variation of oil film thickness and oil pressure are studied and on bases of simulation load bearing capacity , frictional force and coefficient of friction are determined and compared & tabulate in Table 2.

TABLE II LOAD CAPACITY AND FRICTION COEFFICIENT

S no.	Bearing Configuration	Load carrying capacity (W) in newton	Friction Force (Fr) in Newton	Coefficient of friction	%Deviation
1	**Smooth bearing**	4664.100	13.7205	0.002941725	
2	**Start angle 57°**	3581.287	12.47	0.003481988	18.3655287
3	**Start angle 122°**	4527.330	12.275	0.002711311	-7.8326153

Static pressure and Temperature distribution on the walls of the bearing is presented in the Fig. 6-10 and the path lines of pressure variations with respective to time is presented.

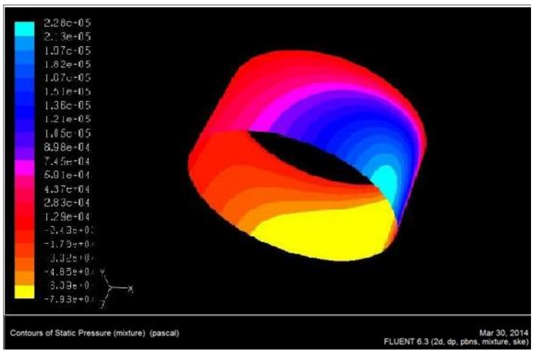


Figure 6 Contour of Static Pressure for Journal surface starting from a plane 10% of length

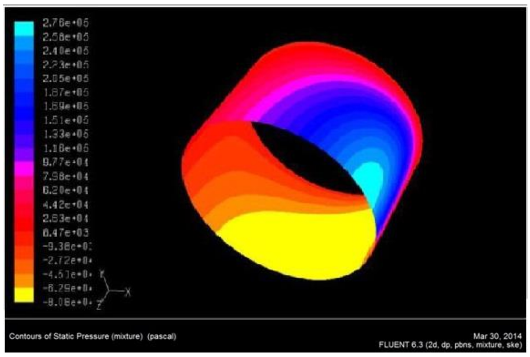


Figure 7 Pressure contour on Journal surface starting from plane of symmetry

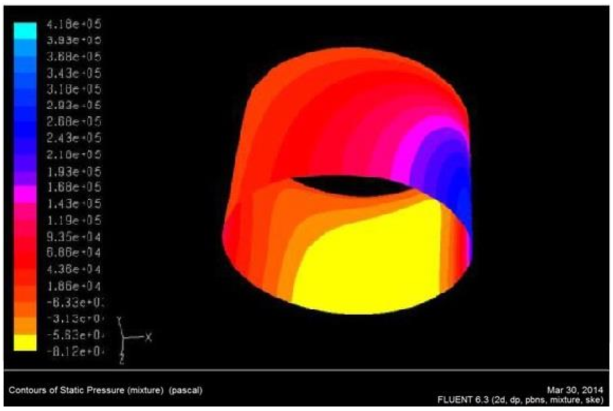


Figure 8 Contour of Static Pressure for Journal surface considering temperature effect.

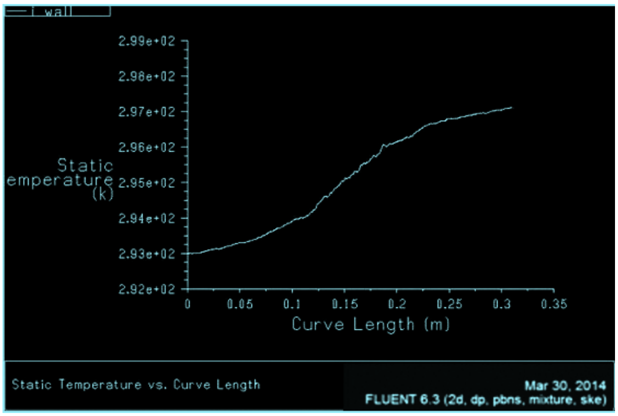


Figure 9 Temperature Distribution on Journal Surface.

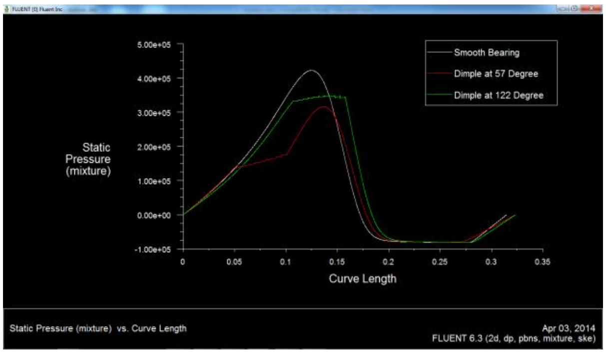


Figure 10 Pressure Distribution on Journal Surface.

From the fig 9 it has been seen that the temperature of journal bearing surface significantly increases throughout the wall surface. And in figure 10 comparative analysis has been done between smooth, dimple at 57^0 and dimple at 122^0.it has been seen that the static pressure significantly increases till 0.15 of bearing surface and gradually starts decline till it reaches 0.2 and remain constant upto0.3 and afterwards again starts increasing. The same trend has been seen for other Dimple engraved bearing but in lesser limit on comparison with sooth bearing.

VI. Conclusion

Dynamic behavior of thin film lubricated journal bearing system have been studied and on bases of result following conclusions as are drawn. Due to friction the bearing temperature significantly increases. On increasing bearing temperature the viscosity of the lubricant decreases within bearing. The loads bearing capacity of smooth bearing is more as compared with dimple bearing. The Dimple at 570 as less static pressure distribution in comparison with 1220 dimple bearing.

References

[1] W. F. Hughes, F. Osterle, "Temperature Effects in Journal Bearing Lubrication", Tribology Transactions, 1: 1, 210 — 212, First published on: 01 January 1958 (iFirst).

[2] S. A. Gandjalikhan Nassab, M S Moayeri, "Three-dimensional thermohydrodynamic analysis of axially grooved journal bearings", Proc Instn Mech Engrs Vol 216 Part J: J Engineering Tribology, December 2001, Page: 35-47.

[3] Prakash Chandra Mishra, "Thermal Analysis of Elliptic Bore Journal Bearing", Tribology Transactions, 50: 137-143, 2007.

[4] Wei Wang, Kun Liu, Minghua Jiao,"Thermal and non Newtonian analysis on mixed liquid- solid lubrication‖", Tribology International 40 (2007) 1067-1074.

[5] K.P. Gertzos, P.G. Nikolakopoulos, C.A. Papadopoulos,"CFD analysis of journal bearing hydrodynamic lubrication by Bingham lubricant", Tribology International 41 (2008) 1190– 1204.

[6] S Cupillard, S Glavatskih, and M J Cervantes, "Computational fluid dynamics analysis of a journal bearing with surface texturing", Proc. IMechE, Part J: J. Engineering Tribology, 222(J2), 2008, page 97-107.

[7] K.P. Gertzos, P.G. Nikolakopoulos, C.A. Papadopoulos ,"CFD analysis of journal bearing hydrodynamic lubrication by Bingham lubricant", Tribology International, Volume 41, Issue 12, December 2008, Pages 1190-1204

[8] M. Deligant, P. Podevin, G. Descombes , "CFD model for turbocharger journal bearing performances", Applied Thermal Engineering, Volume 31, Issue 5, April 2011, Pages 811-819

[9] Dimitrios A. Bompos, Pantelis G. Nikolakopoulos, "CFD simulation of magnetorheological fluid journal bearings", Simulation Modelling Practice and Theory, Volume 19, Issue 4, April 2011, Pages 1035-1060

ISSN: 2278-0181
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[10] B. Manshoor, M. Jaat, Zaman Izzuddin, Khalid Amir, "CFD Analysis of Thin Film Lubricated Journal Bearing", Procedia Engineering, Volume 68, 2013, Pages 56-62

[11] F. Omidbeygi, S.H. Hashemabadi, "Exact solution and CFD simulation of magnetorheological fluid purely tangential flow within an eccentric annulus", International Journal of Mechanical Sciences, Volume 75, October 2013, Pages 26-33

[12] Gengyuan Gao, Zhongwei Yin, Dan Jiang, Xiuli Zhang, "Numerical analysis of plain journal bearing under hydrodynamic lubrication by water", Tribology International, Volume 75, July 2014, Pages 31-38

[13] Xiuli Zhang, Zhongwei Yin, Gengyuan Gao, Zheng Li, "Determination of stiffness coefficients of hydrodynamic water-lubricated plain journal bearings", Tribology International, In Press, Accepted Manuscript, Available online 31 December 2014