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NUMERICAL INVESTIGATION ON THE INFLUENCE OF SURFACE TEXTURE HYDRODYNAMIC JOURNAL BEARING USING CFD

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ABSTRACT

Thermohydrodynamic (THD) analysis should therefore be carried out to obtain the realistic performance characteristics of the bearing. So many researchers have been done in this field. Most of these analyses used two dimensional energy equation to find the temperature distribution in the fluid film by neglecting the temperature variation in the axial direction and two dimensional Reynolds equation was used to obtain pressure distribution in the lubricant flow by neglecting the pressure variation across the film thickness. In this paper a numerical approach which can accurately predict the performance characteristics of a surface textured (grooving) journal bearing have been presented. In this present investigation, some parameters are changed like; groove diameter, number of groove or pattern. After analysis the results are validated in some good literature which mentioned in reference. The three dimensional momentum equations and the continuity equations governing the flow field in the positive pressure fluid film region in the clearance space of the finite bearing are used to obtain velocity and pressure field in the fluid flow. Three dimensional energy equations is used to obtain the temperature distribution in the fluid film. The solutions of the lubricant flow and thermal equations are obtained using a CFD software Fluent.

KEYWORDS: Textured Journal bearing, Lubrication, ANSYS CFD Fluent, etc.

INTRODUCTION

The behavior of many grease lubricants, as well as electrorheological (ER) and magneto-rheological (MR) fluids [1] proposed as ‘‘smart’’ lubricants, is well described by the Bingham model of non-Newtonian fluid flow. A major difference from the Newtonian fluid flow is that the Bingham model is characterized by two parameters: (a) yield stress and (b) viscosity. When the stress on the lubricant is less than the yield stress, the material is rigid and a region called the ‘‘core’’ is formed; exceeding the yield stress leads to a quasi-Newtonian flow. Studies of Bingham fluid lubrication date back to the experimental work of Cohen et al. [2], on greases which act like Bingham materials. Milne [3] examined a journal-bearing model both experimentally and theoretically and concluded that cores are formed near the bearing at the region of maximum film thickness and near the moving shaft at the region of the minimum film thickness. The extent of this core formation depends only on the geometrical condition at the bearing and the dimensionless yield stress T_0. Batra [4] studied only the case of attached cores in a journal bearing, but showed that both floating and attached cores may occur. He found that the load capacity and the moment of friction of the bearing with a Bingham material are larger than that with a Newtonian material.

1.1 TRIBOLOGY AND JOURNAL BEARING

Tribology is the word derived from Greek word ‘Tribos’, means rubbing process. Tribology mainly deals with technique of lubrication and mechanism of friction and wear. The loss of input energy in any mechanism is mainly due to friction and lubrication is the most efficient way to reduce friction. From this point of view the subject tribology has immense importance. Through research and development in the field of tribology if mechanical systems can be run more with higher efficiency then in turn it will save huge money and can contribute significantly in the progress of the human society. For this reason the subject tribology has been the subject of extensive research for many a decades.

But tribology itself a very vast subject, which includes chemistry of lubricants, physics of fluid flow, surface topology, contact mechanics, material science and mechanical engineering. Therefore for a meaningful and effective development of this subject efforts are needed from chemists, physicists, mathematicians and engineers. For this reasons many governments are putting emphasis separately on development of tribological societies.

In a mechanical system bearing has an important role to play. A bearing is a system of machine elements whose function is to support an applied load by reducing friction between the relatively moving surfaces. The load may be radial or axial or combination of these. Bearings are classified according to the direction of applied load. If the bearing supports radial loads, it is called radial or journal bearing. On other hand, a thrust bearing supports a thrust or axial load. Some bearing can support both radial as well as axial load and they are known as conical bearings.

In the clearance space of a bearing a substance called lubricant is introduced. Any substance which has some amount of viscosity is known as lubricant. The most common lubricants are oils and greases. With the use of lubricant the energy lost by friction is reduced. The lubricant also serves as a cooling agent by carrying away the heat generated by fluid friction.

There are mainly two common types of bearing are used in practice. They are rolling element and fluid film bearings. Rolling-element bearings have much wider use in industries since rolling friction is lower than the sliding friction. Lubricant used in rolling bearing is usually grease.

If two mating surfaces during operated conditions are completely separated by fluid film, such a type of lubrication is called fluid film lubrication. Bearings operated under this type of lubrication, is sometimes known as perfect lubrication. Fluid film bearings are lubricated by hydrodynamic flow which is generated by relative surface motion and/or external pressurization. A fluid film bearing operating on the principle of hydrodynamic lubrication is called ‘self-acting’ bearing, in which the load is supported due to wedging effect of the fluid caused by the relative tangential motion between two surfaces. In an ‘externally pressurized’ (sometimes called hydrostatic) bearing, the load is supported due to pressure in the fluid which is supplied from an external source. In addition to these two types, there is another class of fluid film bearing known as ‘squeeze-film’ bearings, which support a load because of oscillating relative normal motion. In this last two categories, viz., externally pressurized and squeeze-film, the pressure within the clearance space can be generated without the wedging effect (or converging film).

Hydrodynamic journal bearings are considered to be a vital component of all rotating machinery. It is used to support radial loads under high speed operating conditions. Journal bearing may be divided into full journal bearing, where the contact angle of the bushing with the journal is 360º, and partial journal bearing, in which the contact angle is either 180º or less. Full (360º) journal bearings are widely used bearing in industrial machinery. These bearings are widely used bearings in industrial machinery. These bearings can take up rotating load. The partial journal bearings have limited applications and are used when the direction of radial loads does not change. Figure below shows the full and partial journal bearings.

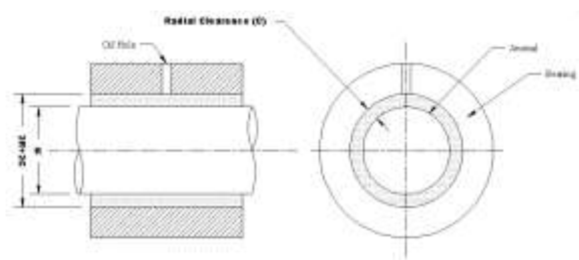

Fig 1.1(a): Full Journal Bearing.

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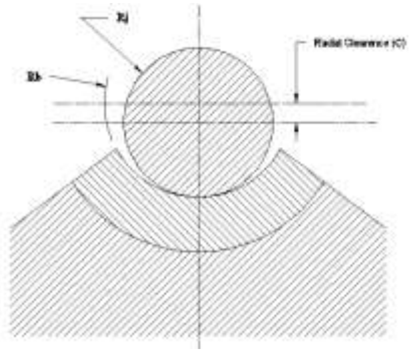

Fig 1.1(b): Partial Journal Bearing.

1.2 JOURNAL BEARINGS SURFACE TEXTURING

Journal bearings with modified surface creating by surface texturing have attracted considerable interest as it is a promising way to enhance their hydrodynamic performance by reducing friction [1]. A different bearing shape by creating texture leads to create a different flow pattern in the lubricant film. Most texturing researchers were motivated by the idea that the surface texturing provides micro reservoirs to enhance lubricant retention or micro traps to capture wear debris. Tala-Ighil et al. [2] analyzed a journal bearing textured with spherical dimples at an eccentricity ratio of about 0.6 and concluded that the presence of texture on the whole bearing surface has a negative influence on the performance whereas a partially textured bearing can have positive effects on the bearing characteristics. Sahu, et al. [3] has also done the thermal analysis of journal bearing. But in their work they simulated a journal bearing without considering the texture effect. So, their work does not say about the effect of texture on bearing and also they don't show the temperature contour. The Thermal analysis of laminar flow in journal bearings was done by Cupillard et. al. [4] using CFD technique. That work was based on the numerical solution of the full three dimensional Navier-Stokes equation, coupled with the energy equation in the lubricant flow and the heat conduction equations in the bearing and the shaft. Discredited forms of the transformed equations were obtained by the control volume method. Similarly Sinanoglu [5] studied the effect of texture surface on shaft of bearing and found that load carriage capacity is significantly increased with threaded textured shaft surfaces to the shafts with non-textured surfaces.

MATHEMATICAL MODELLING

2.1 Basic laws of Tribology

Journal bearing is a hydrodynamic load bearing. In the journal bearing, the lubricant in between the journal and bearing rotates with the journal and supports the load. Due to the physical configuration of journal bearing a wedge shaped fluid film is created as shown in the figure below-

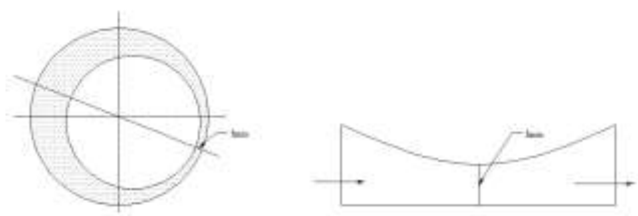

Fig 2.1: Fluid film development in a journal bearing.

Now this wedge shaped fluid film obeys the laws of-

- Conservation of mass flow
- Conservation of momentum in X, Y and Z directions
- Conservation of energy

Besides these the lubricants physical property obeys the equation of states.

Now if the fluid obeys the Newtonian laws of viscosity then we have to modify the above conservation laws as shown below.

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2.1.1 Mass flow conservation

This law is stated as follows-

Rate of increase of mass in the fluid element = Net rate of flow of mass into fluid element

The equation representing above law can be written in vector notation as-

$$\frac{\partial \rho}{\partial t} + div(\rho \vec{\mathbf{V}}) = 0 \text{ -------------------------------- } (1)$$

The above equation is the unsteady three dimensional conservation of mass or continuity equation at a point in a compressible fluid.

If we consider an incompressible fluid then the density (ρ) will be constant and the equation reduces to-

$$\mathrm{div}(\rho\mathbf{V}) = 0$$

$$=> \mathrm{div}(\mathbf{V}) = 0$$

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

2.1.2 Conservation of momentum

The statement for this law can be stated as-

Rate of increase of momentum in the fluid particle in any particular direction = Sum of forces on the fluid particle in that direction.

Now if we impose this law on the flow in X- direction, the mathematical expression becomes-

$$\rho \frac{\mathrm{D}u}{\mathrm{D}t} = \frac{\partial(-\mathrm{p} + \tau_{xx})}{\partial x} + \frac{\partial \tau_{yx}}{\partial y} + \frac{\partial \tau_{zx}}{\partial z} \text{ -- } (3\text{ a})$$

Where,

- $\frac{\mathbf{Du}}{\mathbf{Dt}}$ is the total derivative of 'U' with respect to time't'
- $\boldsymbol{\tau_{xx}}$ is the shear stress in X- plane of differential cubic fluid element having dimensions $\boldsymbol{\delta_x} \times \boldsymbol{\delta_y} \times \boldsymbol{\delta_z}$
- $\boldsymbol{\tau_{yx}}$ is that of Y- plane in X- direction
- $\boldsymbol{\tau_{zx}}$ is that of Z- plane in X- direction

Similarly for Y and Z directions we can write the momentum conservation equation as following-

$$\rho \frac{\mathrm{D}v}{\mathrm{D}t} = \frac{\partial \tau_{xy}}{\partial x} + \frac{\partial(-\mathrm{p} + \tau_{yy})}{\partial y} + \frac{\partial \tau_{zy}}{\partial z} \text{ -- } (3\text{ b})$$

And

$$\rho \frac{\mathrm{D}w}{\mathrm{D}t} = \frac{\partial \tau_{xz}}{\partial x} + \frac{\partial \tau_{yz}}{\partial y} + \frac{\partial(-\mathrm{p} + \tau_{zz})}{\partial z} \text{ -------------------------- } (3\text{ c})$$

2.1.3 Conservation of energy

We can state the conservation of energy as follows-

Rate of increase of energy of a fluid particle = Net rate of heat added to the fluid particle + Net rate of work done on the fluid particle

The mathematical expression for the above law is-

$$\rho \frac{\mathrm{DE}}{\mathrm{D}t} = -div\left(\mathrm{p}\vec{\boldsymbol{V}}\right) \left[\frac{\partial(u\tau_{xx})}{\partial x} + \frac{\partial(u\tau_{yx})}{\partial y} + \frac{\partial(u\tau_{zx})}{\partial z} + \frac{\partial(v\tau_{xy})}{\partial x} + \frac{\partial(v\tau_{yy})}{\partial y} + \frac{\partial(v\tau_{zy})}{\partial z} + \frac{\partial(w\tau_{xz})}{\partial x} + \frac{\partial(w\tau_{yz})}{\partial y} + \frac{\partial(w\tau_{zz})}{\partial z}\right] + div(k\ grad\ \mathrm{T}) + \mathrm{S_E} \text{ --------------}(4\mathrm{a})$$

Now we know that $\mathrm{E} = \mathrm{i} + \frac{1}{2}(\mathrm{u}^2 + \mathrm{v}^2 + \mathrm{w}^2)$, we get–

$$\rho\frac{\mathrm{D}i}{\mathrm{D}t} = -div\left(\mathrm{p}\vec{V}\right) + div(k\ grad\ \mathrm{T}) + \tau_{xx}\frac{\partial u}{\partial x} + \tau_{yx}\frac{\partial u}{\partial y} + \tau_{\mathfrak{z}x}\frac{\partial u}{\partial \mathfrak{z}} + \tau_{xy}\frac{\partial v}{\partial x} + \tau_{yy}\frac{\partial v}{\partial y} + \tau_{\mathfrak{z}y}\frac{\partial v}{\partial \mathfrak{z}} + \tau_{x\mathfrak{z}}\frac{\partial w}{\partial x} + \tau_{y\mathfrak{z}}\frac{\partial w}{\partial y} + \tau_{\mathfrak{z}\mathfrak{z}}\frac{\partial w}{\partial \mathfrak{z}} + \mathrm{S}_i \quad \text{------------------------(4b)}$$

Where, $= \mathrm{S_E} - \vec{\mathrm{V}}.\overrightarrow{\mathrm{S_m}}$

In special case of an incompressible fluid we have, i = cT and $\mathrm{div}(\vec{\mathrm{V}}) = 0$, then we get-

$$\rho c\frac{\mathrm{DT}}{\mathrm{D}t} = div(k\ grad\ \mathrm{T}) + \tau_{xx}\frac{\partial u}{\partial x} + \tau_{yx}\frac{\partial u}{\partial y} + \tau_{\mathfrak{z}x}\frac{\partial u}{\partial \mathfrak{z}} + \tau_{xy}\frac{\partial v}{\partial x} + \tau_{yy}\frac{\partial v}{\partial y} + \tau_{\mathfrak{z}y}\frac{\partial v}{\partial \mathfrak{z}} + \tau_{x\mathfrak{z}}\frac{\partial w}{\partial x} + \tau_{y\mathfrak{z}}\frac{\partial w}{\partial y} + \tau_{\mathfrak{z}\mathfrak{z}}\frac{\partial w}{\partial \mathfrak{z}} + \mathrm{S}_i \quad \text{-----------------------(5)}$$

Now we know-

$$\mathrm{h} = \mathrm{i} + (\mathrm{p/e}) \text{ and } \mathrm{h_0} = \mathrm{h} + \tfrac{1}{2}(\mathrm{u^2 + v^2 + w^2})$$

Considering the above relations we get-

$$\mathrm{h_0} = \mathrm{i} + \mathrm{p/e} + \frac{1}{2}(\mathrm{u^2 + v^2 + w^2}) = \mathrm{E} + \mathrm{p/e}$$

So the above energy equation becomes-

$$\rho\frac{\mathrm{DE}}{\mathrm{D}t} + div\left(\mathrm{p}\vec{V}\right) \quad div(k\ grad\ \mathrm{T}) \quad \frac{\partial \mathrm{p}}{\partial t} \quad \left[\frac{\partial(u\tau_{xx})}{\partial x} + \frac{\partial(u\tau_{yx})}{\partial y} + \frac{\partial(u\tau_{\mathfrak{z}x})}{\partial \mathfrak{z}} + \frac{\partial(v\tau_{xy})}{\partial x} + \frac{\partial(v\tau_{yy})}{\partial y} + \frac{\partial(v\tau_{\mathfrak{z}y})}{\partial \mathfrak{z}} + \frac{\partial(w\tau_{x\mathfrak{z}})}{\partial x} + \frac{\partial(w\tau_{y\mathfrak{z}})}{\partial y} + \frac{\partial(w\tau_{\mathfrak{z}\mathfrak{z}})}{\partial \mathfrak{z}}\right] + \mathrm{S_h} \quad \text{-----------------(6)}$$

Besides above mentioned conservation equations, the two equations of states have to be considered. These are-

$$\mathrm{p} = f(\rho, \mathrm{T}), \text{and ------------(7)}$$
$$i = f(\rho, \mathrm{T}) \text{ ------------(8)}$$

For perfect gas these are—

$$\mathrm{p} = \rho\mathrm{RT}, \text{and ------------(9)}$$
$$i = \mathrm{C_v T} \text{ ------------(10)}$$

Now we are going to combine the above five relations i.e. mass conservation equation, three momentum conservation equations (which are familiar as Navier-Stoke's equation) and one energy conservation equation along with the Newton's viscosity equations.

If we consider the fluid to be a Newtonian fluid then we have to consider the following equations of viscosity-

$$\tau_{xx} = 2\eta\frac{\partial u}{\partial x} + \lambda div\vec{\mathrm{V}},\ \tau_{yy} = 2\eta\frac{\partial v}{\partial y} + \lambda div\vec{\mathrm{V}}$$
$$\tau_{\mathfrak{z}\mathfrak{z}} = 2\eta\frac{\partial w}{\partial \mathfrak{z}} + \lambda div\vec{\mathrm{V}},\ \tau_{xy} = \tau_{yx} = \eta\left(\frac{\partial u}{\partial y} + \frac{\partial v}{\partial x}\right)$$
$$\tau_{x\mathfrak{z}} = \tau_{\mathfrak{z}x} = \eta\left(\frac{\partial w}{\partial x} + \frac{\partial u}{\partial \mathfrak{z}}\right),\ \tau_{y\mathfrak{z}} = \tau_{\mathfrak{z}y} = \eta\left(\frac{\partial v}{\partial \mathfrak{z}} + \frac{\partial w}{\partial y}\right) \quad \text{----------------------(11)}$$

Now if we combine the above viscosity relations with momentum conservation equations or Navier-Stoke's relations we get—

$$\rho\frac{\mathrm{D}u}{\mathrm{D}t} = -\frac{\partial \mathrm{p}}{\partial x} + div(\eta\ grad\ u) + S_{\mathrm{Mx}} \quad \text{----------------------(12a)}$$
$$\rho\frac{\mathrm{D}v}{\mathrm{D}t} = -\frac{\partial \mathrm{p}}{\partial y} + div(\eta\ grad\ v) + S_{\mathrm{My}} \quad \text{----------------------(12b)}$$
$$\rho\frac{\mathrm{D}w}{\mathrm{D}t} = -\frac{\partial \mathrm{p}}{\partial \mathfrak{z}} + div(\eta\ grad\ w) + S_{\mathrm{M}\mathfrak{z}} \quad \text{----------------------(12c)}$$

And if we combine viscous equations with energy conservation equation, we get—

$$\rho \frac{Di}{Dt} = -p\, div\vec{V} + div(k\ grad\ T) + \Phi + S_i \text{ -------------------(13)}$$

Where ϕ is called dissipation function and can be expanded as—

$$\Phi = \eta\left\{2\left[\left(\frac{\partial u}{\partial x}\right)^2 + \left(\frac{\partial u}{\partial x}\right)^2 + \left(\frac{\partial u}{\partial x}\right)^2\right] + \left(\frac{\partial u}{\partial y} + \frac{\partial v}{\partial x}\right)^2 + \left(\frac{\partial u}{\partial y} + \frac{\partial v}{\partial x}\right)^2 + \left(\frac{\partial u}{\partial y} + \frac{\partial v}{\partial x}\right)^2\right\} + \lambda\left(div\vec{V}\right)^2$$

Here we can see clearly form the above equation of dissipation function φ is a non-negative function.

2.2 Governing equation

For a steady-state and laminar flow, incompressible oil and isothermal condition; Reynolds equation in cylindrical coordinates for texture bearing surface is:

$$\frac{\partial}{\partial\theta}\left(H^3 \frac{\partial P}{\partial y}\right) + r^2 \frac{\partial}{\partial y}\left(H^3 \frac{\partial P}{\partial y}\right) = 6\eta r\left(U \frac{\partial H}{\partial \theta}\right) \text{ ---------------------------(14)}$$

Where θ and y are the coordinates in radial and longitudinal directions, P is hydrodynamic pressure for texture bearing and H is film thickness for texture bearing whereas, η, r and U are notations for lubricant's viscosity, shaft radius and shaft speed respectively.

2.1 Bearing performance parameters

The following performance parameters are calculated for texture journal bearing: the load carrying capacity (W_T), the friction force (F_T) on the journal, and the friction coefficient (f_T) on the journal. The listed performance parameters are obtained as:

$$W_T = \int_0^I \int_0^{2\pi} Prd\theta dy \text{ ----------------------(15)}$$

Percentage variation in load = [(WT −W)/W] ∗ 100 ------------------------(16)

$$F_T = \int_0^I \int_0^{2\pi} \tau rd\theta dy \text{ -------------------(17)}$$

Where shear stress, $\tau = \left(\frac{H}{2r}\right)\left(\frac{\partial P}{\partial\theta}\right)\left(\frac{\eta U}{H}\right)$

Percentage variation in friction force

$$= [(FT - F)/F] * 100 \text{ -------------------------(18)}$$

Load carrying capacity and friction force are obtained numerically through double integration by Simpson's 1/3rd rule.

The coefficient of friction for textured bearing is computed as:

$$f_T = F_T/W_T \text{ -----------------------(19)}$$

Percentage variation in friction coefficient

$$= \left[\frac{F_T - f}{f}\right] * 100 \text{ -------------------------(20)}$$

In above relations W, F, and f are load carrying capacity, friction force, and friction coefficient for smooth bearing respectively.

2.2 Boundary conditions

The boundary condition for the Reynolds equation for the smooth and texture bearings is [23]:

P = 0 at θ = 0◦, 360◦

P = 0 and $\frac{\partial P}{\partial\theta} = 0$ at the rupture limits of the film lubricant. --------------------- (21)

2.3 Texture model

The texture in the present case has been taken to be of negative nature at different locations i.e. by extruding the bearing surface at Configuration I, II and III. In order to obtain the negative texture the conventional sinusoidal wave is converted to negative wave by using Fourier series adopted from Gupta and Kumar [22].

The negative full and half wave rectifier equation is given below as:

$$\delta_f = \left(\frac{4\times d}{\pi}\right)\left(\sum_{n=2,4,6......\infty} \frac{\cos\left(\frac{\eta\times\pi\times r\times\theta}{\omega}\right)}{n^2-1}\right) - \left(\frac{2\times d}{\pi}\right) \text{ -----------------(22)}$$

$$\delta_h = \left(\frac{2\times d}{\pi}\right)\left(\sum_{n=2,4,6......\infty} \frac{\cos\left(\frac{\eta\times\pi\times r\times\theta}{\omega}\right)}{n^2-1}\right) - \left(\frac{d}{\pi}\right) - \frac{\sin\left(\frac{\eta\times r\times\theta}{\omega}\right)}{2} \text{ ----------(23)}$$

Where, δ_f & δ_h = Surface texture variation for full and half wave respectively, m; w = cavity width = (Length of cavity section)/Number of cavities;
d = cavity depth, m; n = Even Integers i.e. 2, 4, 6, 8, 10, 12, In work, maximum number of terms (n) = 100000; r = Radius of shaft, m.
Resultant equation for film thickness H of textured surface is obtained by using (22) and (23) as:

$$H = h + \text{abs}(\delta f),$$
$$H = h + \text{abs}(\delta h)$$

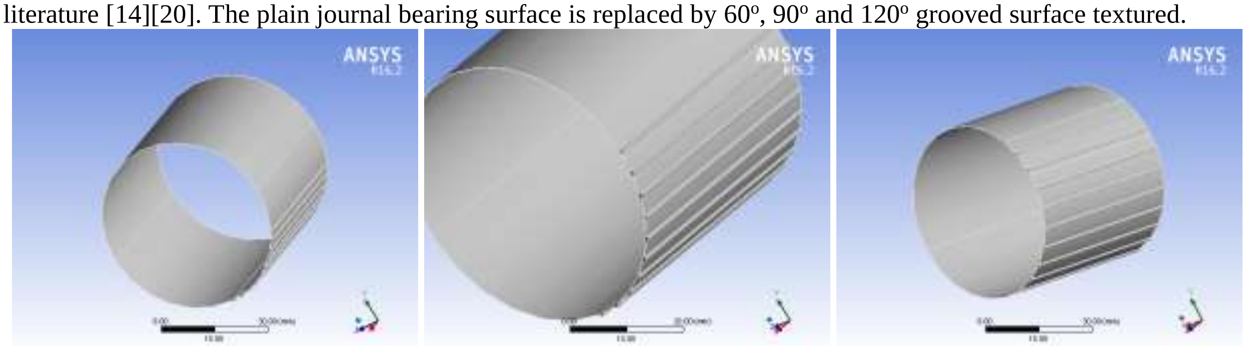
Where

$$h = c\,(1 + \varepsilon \cos\theta) \text{ ---------------------- } (24)$$

Where 'c' is the radial clearance, 'ε' = e/c is the eccentricity ratio and 'e' is the relative eccentricity of the journal.

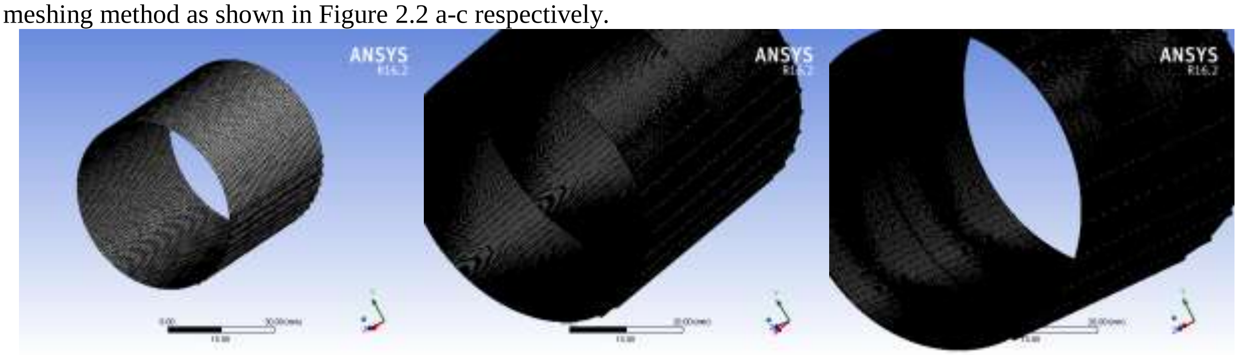
SIMULATION IN COMPUTATIONAL FLUID DYNAMICS (CFD)
In a simulation technique in computational fluid dynamic the following procedure are follows:

3.1 Geometry Modelling
The figure 2.1 a-c are shows that different case has been taken for analysis purpose. This different cases taken from literature [14][20]. The plain journal bearing surface is replaced by 60°, 90° and 120° grooved surface textured.



(a) (b) (c)
Fig 2.1: (a) (b) (c): 3D geometry of 60°, 90° and 120° Textured surface

3.2 Mesh Generation or Discretization
After modelling the product in ANSYS fluent and generate mesh in model. In this process can uses unstructured meshing method as shown in Figure 2.2 a-c respectively.



(a) (b) (c)
Fig 2.2 (a) (b) (c): Meshing view of 60°, 90° and 120° surface textured journal bearing

3.3 Boundary Conditions
It has three surface boundaries like bearing, end plane, journal and middle plane.

Table 3.1 Boundary Condition for all Surface textured journal bearing [14] [20]

Properties	**Value**
Shaft radius	50 mm

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Internal bush radius	50. 1 mm
Bearing length	100 mm
Lubricant density	850 Kg/m^3
Specific heat of the lubricant	2 000 J/Kg °C
Thermal conductivity of lubricant	0. 13 W/m °C
Convective heat transfer coefficient	100 W/m^2°C
Clearance	0. 1 mm
Inlet lubricant temperature	30 °C
Rotational Speed	2 000 rpm (210 rad/sec)
Initial viscosity	0. 04986 Ns/m^2
Eccentricity ε	0.8
Diameter of the groove [20]	1.27mm
Diameter of groove [present]	1.1mm

The above table 3.1 indicated that the operating condition parameters used for the present simulation.

Result and Discussion

In this chapter discussed about the results and validated the all results in reputed literatures. The solution is obtained from ANSYS Fluent 16.2 which are shown in table 4.1 and 4.2.

Table 4.1: Validation of pressure obtained from present and literature [14] and [20]

Work	**Pressure (MPa)**			
	0° Plain Surface	**60° Surface Texture**	**90° Surface Texture**	**120° Surface Texture**
Singla et al. [14]	22.7	-	-	-
Pranab et al. [20]	19.8	27.1	26.7	25.22
Present (ANSYS Fluent 16.2)	-	18.57	29.7	22.27

Table 4.2: Validation of Temperature obtained from present and literature [14] and [20]

Work	**Temperature (k)**			
	Plain Surface	**60° Surface Texture**	**90° Surface Texture**	**120° Surface Texture**
Singla et al. [14]	318.9	-	-	-
Pranab et al. [20]	307.7	302.3	301.06	300.1
Present (ANSYS Fluent 16.2)	-	318.8	300.6	300.2

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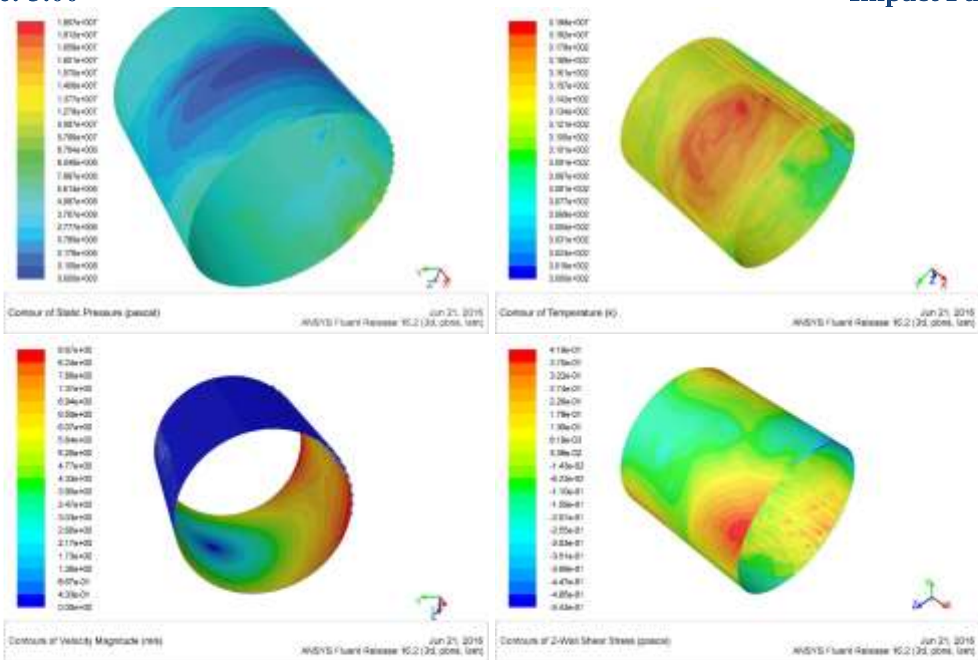


Fig 4.1: Pressure, Temperature, Velocity and wall shear stress contour in 60° surface textured journal bearing resp.

The figure 4.1 shows that the results obtained from ANSYS 16.2 after simulation. The above figures are shows that the pressure, temperature, velocity and wall shear stress generated in the 60° surface textured journal bearing.

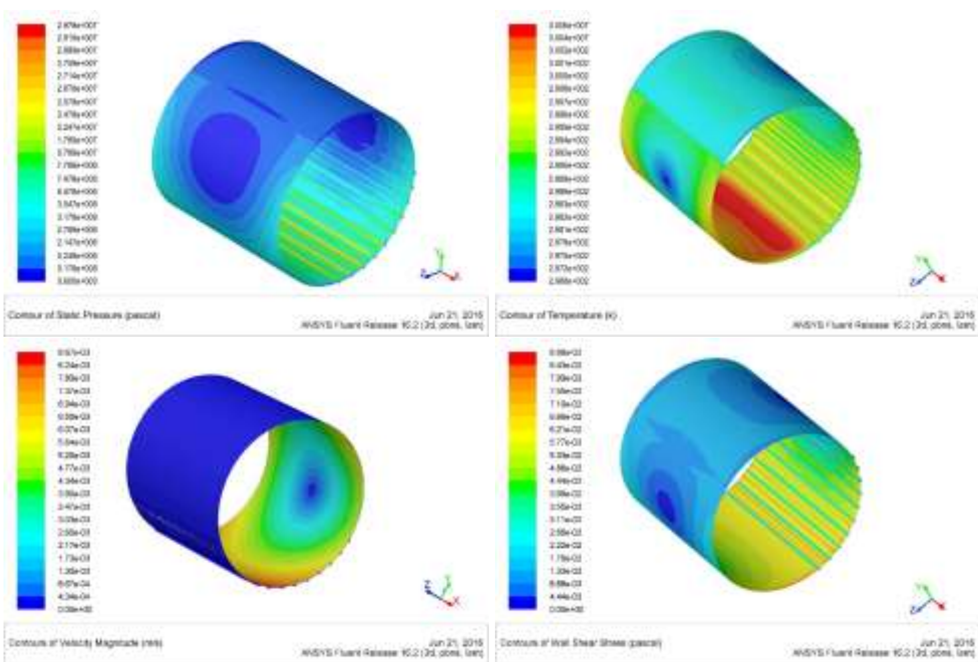


Fig 4.2: Pressure, Temperature, Velocity and wall shear stress contour in 90° surface textured journal bearing resp.

The figure 4.2 shows that the results obtained from ANSYS 16.2 after simulation. The above figures are shows that the pressure, temperature, velocity and wall shear stress generated in the 90° surface textured journal bearing.

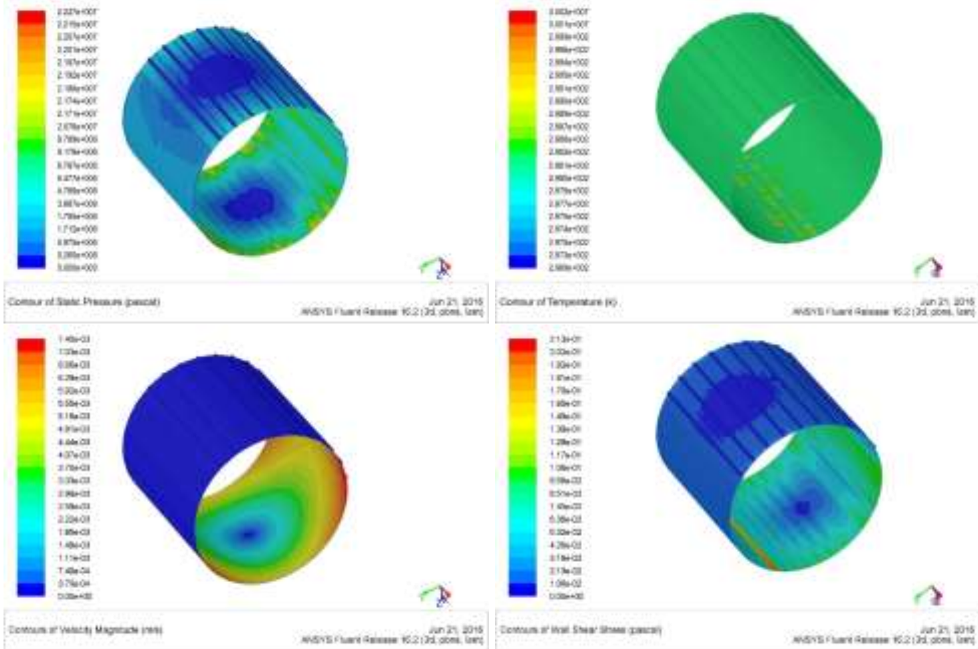
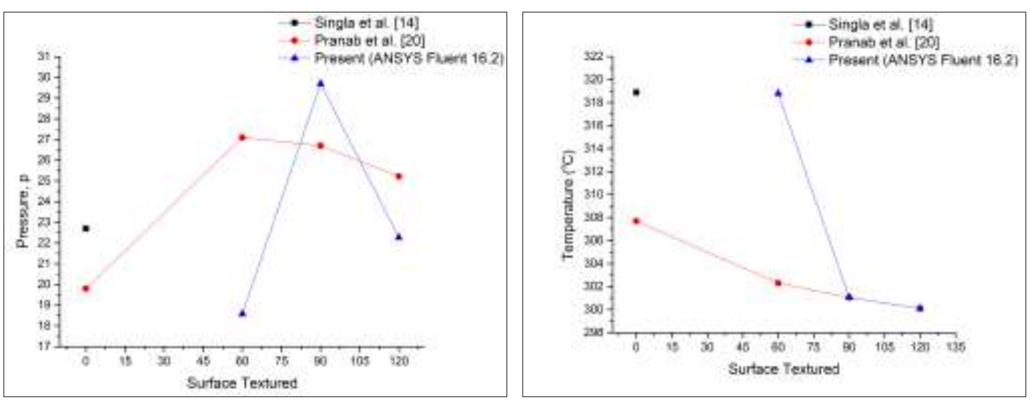


Fig 4.3: Pressure, Temperature, Velocity and wall shear stress contour in 120° surface textured journal bearing resp.

The figure 4.3 shows that the results obtained from ANSYS 16.2 after simulation. The above figures are shows that the pressure, temperature, velocity and wall shear stress generated in the 120° surface textured journal bearing.



(a) (b)

Fig 4.4 (a) (b): Validation of pressure and temperature results between Singla et al. [14], Pranab et al. [20] and Present investigation

The above figure 4.4 (a) is indicated that the validation results of the pressure arises in different journal bearing texture. The figure is shows that, pressure in plain (0°) is less as compare to surface textured bearing as literature [14] and [20]. Also the pressure increases the present investigation by changing some journal surface texture parameter like; diameter of groove and number of pattern.

In present investigation pressure is 23.56%, 33.33%, and 10% more as compare plain (0°), 60°, 90° and 120° respectively mentioned in Singla et al. [14] and Pranab et al. [20]. It is concluded that the increasing pressure in journal bearing is more effective for journal bearing.

The above figure 4.4 (b) is indicated that the validation results in temperature in different journal bearing texture surface. The figure is shows that, temperature distribution in plain (0^o) is more as compare to surface textured bearing as literature [14] and [20]. Also the temperature decreases the present investigation by changing some journal surface texture parameter like; diameter of groove and number of pattern.

In present investigation temperature falls down is 5.73%, 2.3%, and 0.152% more as compare plain (0^o), 60^o, 90^o and 120^o respectively mentioned in Singla et al. [14] and Pranab et al. [20].

It is concluded that the decreasing of temperature cools the interior part of the journal bearing if it was textured in appropriate position and it is also increases the load carrying capacity of the journal bearing.

Overall in present investigation is concluded that the 90^o surface textured journal bearing is gives maximum pressure and mean minimum temperature, hence it is the best for as compare to other (plain or surface texture like 60^o or 120^o).

CONCLUSION

- Pressure in plain (0^o) is less as compare to surface textured bearing as literature [14] and [20]. Also the pressure increases the present investigation by changing some journal surface texture parameter like; diameter of groove and number of pattern.
- In present investigation pressure is 23.56%, 33.33%, and 10% more as compare plain (0^o), 60^o, 90^o and 120^o respectively mentioned in Singla et al. [14] and Pranab et al. [20].
- It is concluded that the increasing pressure in journal bearing is more effective for journal bearing.
- From the present work, it can therefore be concluded that the presence of texture/groove may be fruitful in decreasing the friction coefficient and average temperature with suitable location of dimples/ grooves.
- This is concluded that load carrying capacity is more in case of texture surface than plane surface.
- Temperature of journal bearing can be reduced by creating texture surface in suitable position.
- The micro grooving was able to increase the cooling effect of the air flow.

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] T. P. Indulekha, M. L. Joy, K. Prabhakaran Nair, *"Fluid flow and thermal analysis of a circular journal bearing"*, Wairme- und Stoffubertragung 29(1994) 367-371.

[3] 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.

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

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

[6] 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.

[7] 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(J_2), 2008, page 97-107.

[8] E. Feyzullahoglu, *"Isothermal Elastohydrodynamic Lubrication of Elliptic Contacts"*, Journal of the Balkan Tribological Association, Vol. 15, No 3, 438—446 (2009).

[9] Samuel Cupillard, Sergei Glavatskih, Michel J.Cervantes, *"3D thermohydrodynamic analysis of a textured slider"*, Tribology International 42 (2009) 1487–1495.

[10] Ravindra R. Navthar et al., "*Stability Analysis of Hydrodynamic Journal Bearing using Stiffness Coefficients"*, International Journal of Engineering Science and Technology Vol.2 (2), 2010, page 87-93.

[11] Majumder B. C. 'Introduction to Tribology of Bearings', A. H. Wheeler & Co publication.

[12] Verseteeng H. K. & Malalasekera W. 'An Introduction to Computational Fluid Dynamics', Longman Scientific & Technical publication.

[13] Niyogi P., Chakrabarty S. K., Laha M. K. 'Introduction to Computational Fluid Dynamics', Pearson Education publication.
[14] Amit Singla, Paramjit Singh, Abhay Kumar and Arnit Chauhan "Thermo-Hydrodynamic Analysis on Temperature Profile of Circular Journal Bearing using", Proceedings of 2014 RAECS UIET Panjab University Chandigarh, 06 - 08, March, 2014, 978-1-4799-2291-8/14.
[15] Computational Fluid Dynamics Lord Corporation. Designing with MR fluids. Engineering note /www.lord.-comS, Rev 12/99.
[16] Cohen G, Oren JW. Film pressure distribution in grease lubricated journal bearing. Trans ASME 1949;71:555.
[17] Milne AA. A theory of rheodynamic lubrication. Kolloid Z 1954;139:96–100.
[18] Wada S, Hayashi H, Haga K. Behavior of Bingham solid in hydrodynamic lubrication. Part 3, application to journal bearing. Bull JSME 1974;111(17):1182–91.
[19] Raimondi AA, Boyd J. A solution for the finite journal bearing and its application to analysis and design: III. Trans ASLE 1958;1(1):194–209.
[20] Pranab Samanta*, S. Pradhan and N. C. Murmu, "Thermal Analysis of Textured Journal Bearing Using Computational Fluid Dynamic Technique", International Journal of Engineering Research & Technology (IJERT) Vol. 2 Issue 8, August – 2013, pp 2378-2383
[21] Tala-ighil N, Maspeyrot P, Fillon M, Bounif A (2007) Effects of surface texture on journal bearing characteristics under steady state operating conditions. Proc Inst Mech Eng, Part J J Eng Tribol 221(6):623–633.
[22] Gupta SL, Kumar V (2003) Handbook of electronics, 31st edn. Pragati Prakashan Publication.