Thermal Analysis of Textured Journal Bearing Using Computational Fluid Dynamic Technique

Pranab Samanta*, S. Pradhan and N. C. Murmu
CSIR-Central Mechanical Engineering Research Institute
M. G. Avenue, Durgapur -713209

Abstract

The paper aims to investigate the performances of the textured journal bearing considering temperature profile and load carrying capacity as performance parameters. The performance parameters of the textured bearing have been compared with those for plain journal bearing. Computational Fluid Dynamics (CFD) is used to model the flow between journal and bearing. The Navier-Stokes equation is solved assuming under laminar, isothermal, and steady state conditions with commercial software. A three dimensional geometry is used to model the journal bearing. In this case, the journal rotates and the position of bearing is fixed and air is used as lubricant. From the investigation, it is observed that load carrying capacity for textured bearing is higher than the simple journal bearing. Considering thermal effect it is found that maximum temperature produced is lower in case of textured journal bearing than that of without textured journal bearing. This paper discusses the pressure and temperature profiles in depth and reported that textured bearing has significant advantages over the simple journal bearing.

1. Introduction

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. In this work, the simulation has been done in a 3-Dimensional flow region considering the texture effect by creating texture in bearing surface in order to provide clear picture about the temperature by showing the contours of temperature.

2. Governing Equations

Navier-Stokes equations are the classical governing equations that govern the dynamics of fluid motion. The Navier-Stokes equations, momentum equation (a) coupled with the continuity equation (b), are solved over the domain together with the energy equation (c), using the finite volume method. The flow is laminar. Thermal effects are considered in the 3D domain.

$$\frac{\partial \rho}{\partial t} + \frac{\partial(\rho u_i)}{\partial x_i} = 0 \qquad (a)$$

ρ is the density of the fluid, x_i represents the coordinate in the i-direction and $u_i$represents the velocity in the i-direction.

$$\frac{\partial}{\partial t}(\rho u_i) + \frac{\partial}{\partial x_j}(\rho u_i u_j) = -\frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_j}\left(\square\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right)\right) \qquad \text{(b)}$$

p is the fluid pressure and μ the dynamic viscosity of the fluid.

$$\frac{\partial(\rho h_{tot})}{\partial t} - \frac{\partial p}{\partial t} + \frac{\partial}{\partial x_i}(\rho u_i h_{tot}) = \frac{\partial}{\partial x_i}\left(\lambda\frac{\partial T}{\partial x_i}\right) + \frac{\partial u_i}{\partial x_j}\left[\mu\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_i}{\partial x_i}\right) - \delta ij\left(\frac{2}{3}\mu\frac{\partial u_k}{\partial x_k}\right)\right] \quad \text{(c)}$$

T is the fluid temperature, λ the thermal conductivity of the fluid and h_{tot} is thespecific total enthalpy.

where $\delta_{ij} = 1$ when $i = j$, and 0 when $i \neq j$. The total enthalpy is calculated by thefollowing expression:

$$h_{tot} = h_{stat} + \frac{1}{2}V^2 \qquad \text{(d)}$$

Here, $\frac{1}{2}V^2$ is representing the kinetic energy. The static energy The static enthalpy is calculated by integratingthe following expression:

$$dh = C_p + \frac{1}{\rho}\left[1 + \frac{T}{\rho}\left(\frac{\partial\rho}{\partial T}\right)_p\right]dP \quad \text{(e)}$$

3. Methodology

The model is created by using a commercial mechanical CAD/CAM package. It is one of the newer generations of systems that not only offer a full 3-D solid modeller, in contrast to purely 2-D and surface modellers, but also provide parametric functionality. The whole working process is described in theFigure 1.

3.1 Geometric Information

The geometry of the flow domain was created in Cartesian (X, Y, Z) coordinates, with the cylinder axial centreline along the Z axis. The geometry consisted of a cylinder having specification as describe in Table.1.

When the 2D sketch was completed, extrude the sketch up to 75 mm length. Since FLUENT® software have symmetry option for middle plane, so half length of Journal bearing i.e. 75 mm can be taken. The complete 3D model is shown in Figure 2.The surface texture adopted in this work as shown in Figure 3 is a series of grooves created on the bearing surface along the axial direction from the mid-plane of the bearing in different angular position count from vertical axis of bearing .

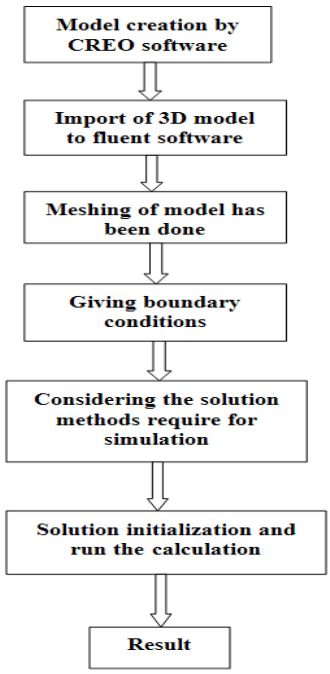

Figure.1: Flow chart of analysis methodology

Table 1: Input data for bearing

Length of the bearing (L)	150mm
Radius of Shaft (R)	100
Radial Clearance()	0.5mm
Eccentricity (ε)	0.4
Angular Velocity (ω	50 rad/s

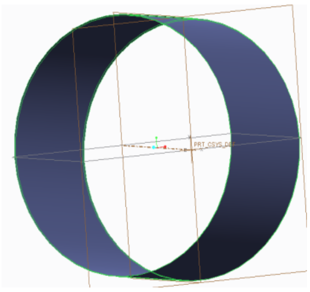

Figure.2: 3d-geometry of Plane surface

The different textures made up to 30°,60°,90° and 120° from vertical axis of bearing. The cross-section of the texture is chosen to be circular as this would be relatively easy to manufacture and quite convenient from a meshing point of view. A groove is characterized by its width (*w*), depth (*d*) and length (*l*). In this groove, *d* = 1.15 mm.

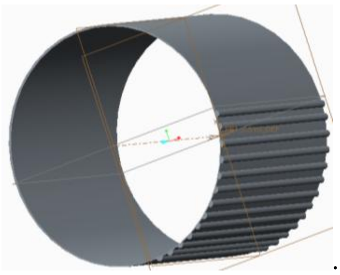

Figure.3: 90° textured surface

3.2 CFD Analysis

The number of simulation software on CFD is available in market. FLUENT® is one of the most popular and widely used amongst them. The software used in this project work to investigate the influence of texture surface on a journal bearing is FLUENT® (Release 14).

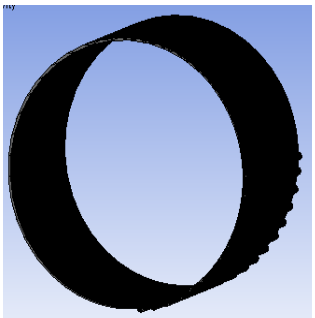

Figure 4: mesh generation of 90° texture model

After the 3D model is created, the continuous space of the flow domain is divided into sufficiently small discrete cells, the distribution of which determines the positions where the flow variables are to be calculated and stored meshing of model has been done by using meshing option of FLUENT®. 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.

3.3Boundary conditions

It has four surface boundaries like bearing, end plane, journal and middle plane. Bearing is defined as wall. Wall motion is stationary. No slip is taken in shear condition. The material of bearing is taken as Brass. End plane is taken as pressure outlet. Backflow total temperature is taken as 300k. Journal is defined as stationary while the wall is rotating. Speed is defined as 50 rad/s. The material of journal is taken as Stainless steel. Middle plane is taken as symmetry. The analysis is carried out under operating conditions of 101325 Pa. The thermal boundary conditions assumed are adiabatic at the fluid-solid interface. For this, zero heat flux is assumed between the fluid and the adjacent wall.

$$\lambda \frac{\partial T}{\partial n}\Big|_{wall} = 0$$

Analysis is carried out by importing the meshed file and then it is undergoes a checking of information of size. The solver is defined first. Solver is taken as pressure –velocity coupling i.e. SIMPLE algorithm and second order unwinding strategy are used.

Table 2 :Property of material

Property	Density (kg/m^3)	Specific heat capacity (J/kg-K)	Thermal conductivity (W/m-K)	Viscosity (kg/m-s)
Air	1.225	1006.43	0.0242	1.78 * 10^{-5}
Br ass	8600	380	109	-
Stainless steel	7700	490	19	-

In this simulation bearing is taken as brass material, journal is taken as stainless steel and air as lubricant. As we consider a laminar flow and air as lubricant. After the simulation, the temperature distribution or profile on journal surface has been found out as contour representation.

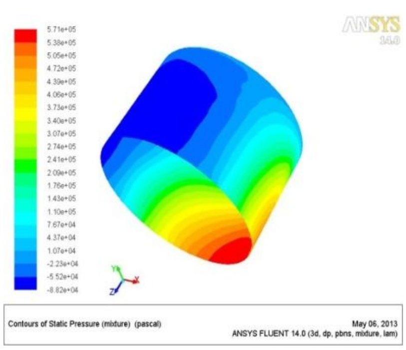

Fig. 5: Pressure Profile

In Figure 5, pressure distribution on journal surface of a journal bearing has been generated without considering the texture effect. The above result matches with the work of Sahu et al [3] as shown in Figure 6. As in their work they had not mentioned which type of material they use, so due to this reason slightly difference arises in our result.

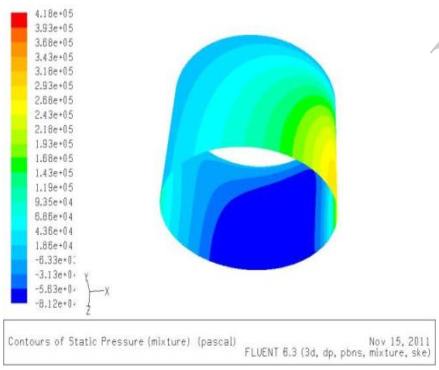

Fig. 6: Pressure profile reported in journal [3]

4. Results and discussion

In the present work, the thermal characteristic of journal bearings under laminar flow was examined by CFD technique. It was noted that the journal running under air lubricated conditions. To explain and compare the performance of the textured bearing with smooth bearing, the temperature contours of both bearing is obtained. Temperature profile of smooth bearing is depicted in Figure 7. Temperature profile of 30°,60°, 90° and 120° textured surface bearing have been shown in Figure 8 to 11 respectively. Maximum temperature value of corresponding bearing is tabulated in Table 3.

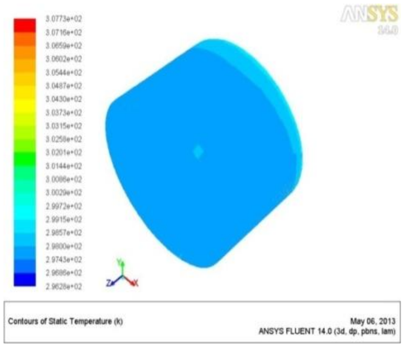

Figure.7: Temperature profile of plane bearing

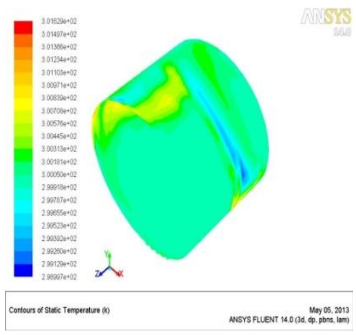

Figure.8: Temperature profile of 30° texture bearing

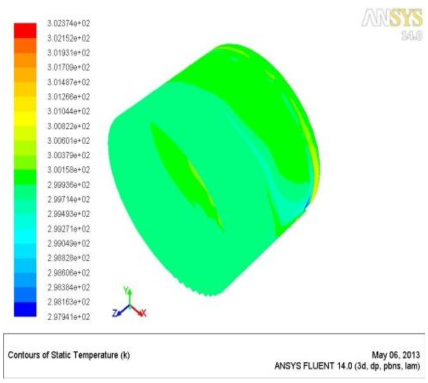

Figure.9: Temperature profile of 60° texture bearing

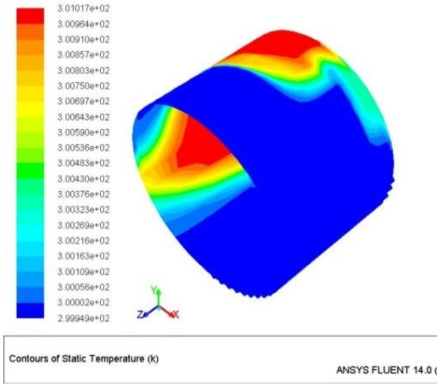

Figure.10: Temperature profile of 90° texture bearing

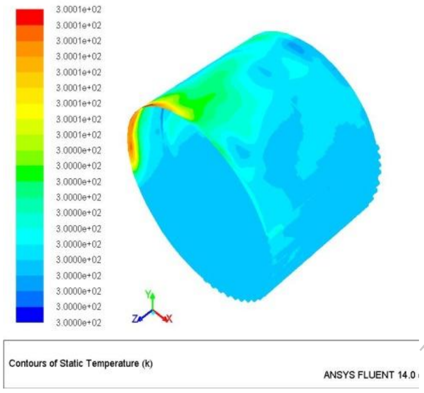

Figure.11: Temperature profile of 120° texture bearing

Pressure Profile:

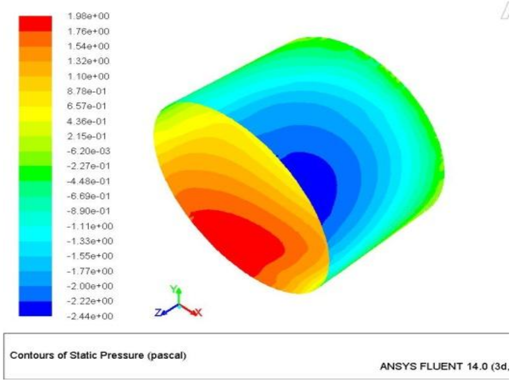

Figure.12: Pressure profile of plane texture bearing

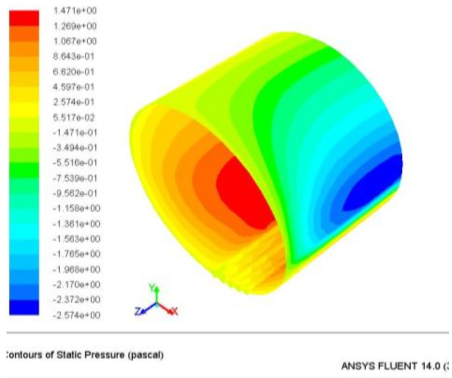

Figure.13: Pressure profile of 30° texture bearing

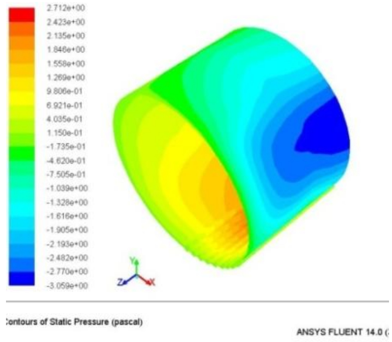

Figure.14: Pressure profile of 60° texture bearing

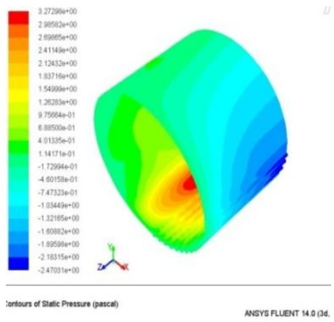

Figure.15: Pressure profile of 90° texture bearing

Table No.3: Value of Temperature from contour

Surface	Plane	30°	60°	90°	120°
Max(K)	307.7	301.16	302.3	301.01	300.01

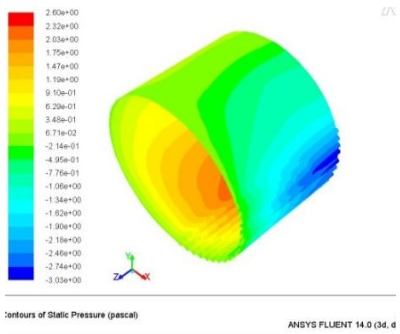

Figure.16: Pressure profile of 120° texture bearing

Table 4: Value of Pressure from contour

Surface	Plane	30°	60°	90°	120°
Max(Pa)	1.98	1.47	2.71	3.27	2.60

From Table 3, it is observed that textured bearing's peak temperature is minimum compared to smooth bearing. It is also examined that among different textured surface 120° textured bearing give minimum value of peak temperature. Pressure profile of smooth bearing is depicted in Figure 12. Temperature profile of 30°,60°,90° and 120° textured surface bearing have been shown in Figure 13 to Figure 16 respectively. Maximum pressure value of corresponding bearing is tabulated in Table 4. It is observed that among different textured surface 90° textured bearing gives maximum value of peak pressure. When we compare the result of plane surface with texture surface we can find that texture affect the thermal condition of the bearing. Making the texture in bearing surface, it reduces the temperature created from frictional force inside the journal bearing and also increases the value of pressure. From the above result it is clear that smooth surface have maximum temperature of 307 K but the texture surfaces in most cases it is around 301 K. That means texture cools the interior part of the journal bearing if it was textured in appropriate position. Similarly plane surface have maximum pressure of 1.98Pa but after texturing the surfaces, in most of the cases pressure increases. This signifies that load carrying capacity is more in case of texture surface than plane surface.

4. Conclusions

This work investigated the effects of textured surface on the performance parameters of a journal bearing i.e. load carrying capacity and temperature profile. The analysis of dimples created on the stationary surface is performed with different configurations. The overall conclusions from this study are as follows:

- 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.
- Texture surface have more load carrying capacity than plane bearing.
- Among different textured profiles 90° and 120° give better results compare to other profiles. Since 90° profile have edge over 120° in view point of maximum load carrying capacity 90° textured profile is an optimum choice for the proposed bearing geometry.

10. References

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