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Thermohydrodynamic Analysis of a Journal Bearing Using CFD as a Tool

Mukesh Sahu, Ashish Kumar Giri, Ashish Das

Abstract- The current trend of modern industry is to use machineries rotating at high speed and carrying heavy rotor loads. In such applications hydrodynamic journal bearings are used. 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. Thermohydrodynamic (THD) analysis should therefore be carried out to obtain the realistic performance characteristics of the bearing. In the existing literature, several THD studies have been reported. 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 CFD technique has been used to accurately predict the performance characteristics of a plain journal bearing. Three dimensional study has been done to predict pressure distribution along journal surface circumferentially as well as axially. Three dimensional energy equation is used to obtain the temperature distribution in the fluid film.

Index Terms- Journal Bearing, Eccentricity Ratio, Pressure distribution, Thermal analysis, Temperature distribution, CFD, Fluent

I. Introduction

The increasing trend towards higher-speed, higher-performance but smaller-size machinery has pushed the operating conditions of bearings towards their `limit design'. Hence, for reliable prediction of the performance of such bearings, a model which accounts for all the operating conditions is becoming increasingly important. Since, the lubricant viscosity strongly depends on temperature, the usual classical assumptions of constant viscosity or effective viscosity become untenable. The temperature variation and hydrodynamic pressure of lubricant in journal bearings depend strongly on the lubricant flow through the entire bearing. Thereby, prediction of a bearing performance based on a thermohydrodynamic (THD) analysis generally requires simultaneous solution of the equations governing the flow of lubricant, the energy equation for the flow field, the heat conduction equations in the bearing and the shaft and an equation describing the dependence of the lubricant viscosity upon temperature. Further, factors such as the complex geometrical shape of the bearing assembly, the regime of flow which may be laminar, transitional or complete turbulence, the type of flow in the cavitated region and the nature of mixing of the supplied lubricant with the recirculating streamers within the supply recess introduce difficulties in the numerical THD analysis of journal bearings. It is not surprising that research into THD bearing performance is still incomplete. Therefore, different simplifying assumptions, some of which may be based on the experimental observations, are usually made to obtain approximate THD characteristics of journal bearings.

II. THEORETICAL BACKGROUND

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

$$\rho \frac{Du}{Dt} = -\frac{\partial p}{\partial x} + div(\eta \ grad\ u) + S_{Mx} \qquad (1.a)$$

$$\rho \frac{Dv}{Dt} = -\frac{\partial p}{\partial y} + div(\eta \ grad\ v) + S_{My} \qquad (1.b)$$

$$\rho \frac{Dw}{Dt} = -\frac{\partial p}{\partial \mathfrak{z}} + div(\eta \ grad\ w) + S_{M_\mathfrak{z}} \qquad (1.c)$$

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.

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

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.

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

and the equations of state i.e.

$p = \rho RT$ and $i = C_v T$.

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-

1) Inertia and body force terms are negligible as compared to viscous and pressure forces.
2) There is no variation of pressure across the fluid film i.e. $\frac{\partial p}{\partial y} = 0$.
3) There is no slip in the fluid-solid boundaries (as shown in the figure below).
4) No external forces act on the film.
5) The flow is viscous and laminar (as shown in the figure below).
6) 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 (10^{-3}).

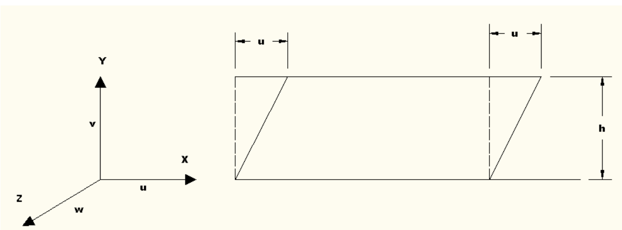

Fig 1.0: Fluid film depicting the Shear

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

$$\frac{\partial p}{\partial x} = \frac{\partial}{\partial y}\left(\eta \frac{\partial u}{\partial y}\right) \qquad (4.a)$$

$$\frac{\partial p}{\partial z} = \frac{\partial}{\partial y}\left(\eta \frac{\partial w}{\partial y}\right) \qquad (4.b)$$

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.

$$\frac{\partial u}{\partial y} = \frac{1}{\eta}\frac{\partial p}{\partial x} y + C_1 \qquad (5.a)$$

$$\frac{\partial w}{\partial y} = \frac{1}{\eta}\frac{\partial p}{\partial z} y + C_2 \qquad (5.b)$$

Where C1 and C2 are constants

Integrating above equations once more we get-

$$u = \frac{1}{2\eta}\frac{\partial p}{\partial x} y^2 + C_1 y + C_3 \qquad (6.a)$$

$$w = \frac{1}{2\eta}\frac{\partial p}{\partial z} y^2 + C_2 y + C_4 \qquad (6.b)$$

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 $y = h, u = u_a, w = w_a$

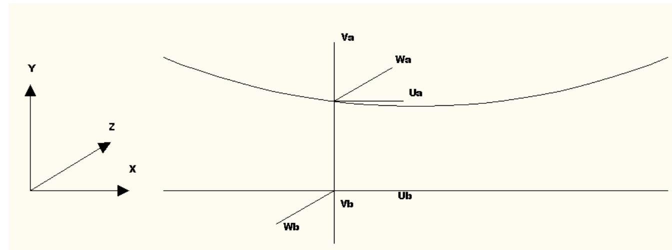

Fig 2.0: Fluid film depicting the velocity components

Imposing above boundary conditions we get-

$$u = \frac{1}{2\eta}\frac{\partial p}{\partial x} y(y-h) + \left(\frac{h-y}{h}\right) u_b + \frac{y}{h} u_a \qquad (7.a)$$

$$w = \frac{1}{2\eta}\frac{\partial p}{\partial z} y(y-h) + \left(\frac{h-y}{h}\right) w_b + \frac{y}{h} w_a \qquad (7.b)$$

Now using above expressions of velocity components in continuity equation i.e. Eqn (2) we get-

$$\frac{\partial(\rho v)}{\partial y} = -\frac{1}{2}\left\{\frac{\partial}{\partial x}\left[\frac{\rho}{\eta}\frac{\partial p}{\partial x} y(y-h)\right] + \frac{\partial}{\partial z}\left[\frac{\rho}{\eta}\frac{\partial p}{\partial z} y(y-h)\right]\right\} - \frac{\partial}{\partial x}\left[\rho\left\{\left(\frac{h-y}{h}\right)u_b + \frac{y}{h}u_a\right\}\right] - \frac{\partial}{\partial z}\left[\rho\left\{\left(\frac{h-y}{h}\right)w_b + \frac{y}{h}w_a\right\}\right] - \frac{\partial \rho}{\partial t} \qquad (8)$$

Now imposing the boundary conditions-

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

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

$$\rho(v_a - v_b) = -\frac{1}{2}\left\{\int_0^h \frac{\partial}{\partial x}\left[\frac{\rho}{\eta}\frac{\partial p}{\partial x} y(y-h)\right] dy + \int_0^h \frac{\partial}{\partial z}\left[\frac{\rho}{\eta}\frac{\partial p}{\partial z} y(y-h)\right] dy\right\} - \int_0^h \frac{\partial}{\partial x}\left[\rho\left\{\left(\frac{h-y}{h}\right)u_b + \frac{y}{h}u_a\right\}\right] dy - \int_0^h \frac{\partial}{\partial x}\left[\rho\left\{\left(\frac{h-y}{h}\right)u_b + \frac{y}{h}u_a\right\}\right] dy - h\frac{\partial \rho}{\partial t} \qquad (9)$$

Integrating the Eqn (9) we get-

$$\frac{\partial}{\partial x}\left(\frac{\rho h^3}{12\eta}\frac{\partial p}{\partial x}\right) + \frac{\partial}{\partial z}\left(\frac{\rho h^3}{12\eta}\frac{\partial p}{\partial z}\right) = \frac{\partial}{\partial x}\left[\frac{\rho(u_a - u_b)h}{2}\right] + \frac{\partial}{\partial z}\left[\frac{\rho(w_a - w_b)h}{2}\right] + \rho(v_a - v_b) - \rho u_a \frac{\partial h}{\partial x} - \rho w_a \frac{\partial h}{\partial z} + h\frac{\partial \rho}{\partial t} \qquad (10)$$

The two terms of left hand side of the Eqn (10) is due to pressure gradient and first two terms of the right hand side of the Eqn (10) is due to surface velocities. These are called Poiseuille and Couette terms respectively.

Now if we impose the following boundary conditions-

$$w_a = w_b = 0$$

$$v_a = u_a \frac{\partial y}{\partial x}$$

$$U = \frac{u_a + u_b}{2} \text{ in equation (3.7)we get–}$$

$$\frac{\partial}{\partial x}\left(\frac{\rho h^3}{12\eta}\frac{\partial p}{\partial x}\right)+\frac{\partial}{\partial \mathfrak{z}}\left(\frac{\rho h^3}{12\eta}\frac{\partial p}{\partial \mathfrak{z}}\right)=U\frac{\partial}{\partial x}(\rho h) \qquad (11)$$

If the fluid property ρ does not vary, as in the case of incompressible lubricant we can write Eqn (11) as follows-

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

If we assume the bearing is of infinite length, then $\frac{\partial p}{\partial \mathfrak{z}}=0$ and the Eqn (12) becomes-

$$\frac{\partial}{\partial x}\left(h^3\frac{\partial p}{\partial x}\right)=12\eta U\frac{\partial h}{\partial x} \qquad (13)$$

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 (13) the governing equation of the journal bearing becomes-

$$\frac{\partial}{\partial x}\left(h^3\frac{\partial p}{\partial x}\right)=6\eta U\frac{\partial h}{\partial x} \qquad (14)$$

Using polar coordinates-

$x = R\theta \qquad and \qquad dx = R\,d\theta$

The equation (14) 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

$$p=\frac{6\eta UR}{C^2}\left[\int\frac{d\theta}{(1+\varepsilon\cos\theta)^2}-\frac{h_m}{C}\int\frac{d\theta}{(1+\varepsilon\cos\theta)^3}\right]C_1 \qquad (15)$$

Where C_1 is a constant.

Now putting the boundary conditions-

i) At $p=0, \theta=0$

ii) At $p=0, \theta=2\pi$, we get from equation (3.14)—

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

Now load carrying capacity becomes:

$$W=L\sqrt{\left(\int_0^{2\pi}p\cos\theta\,R\,d\theta\right)^2+\left(\int_0^{2\pi}p\sin\theta\,R\,d\theta\right)^2} \qquad (16)$$

In the above equation 'p' can be substituted from Eqn (3.15). But here a problem may be raised. Value of φ and Є depends on the configuration, loading condition and lubricant. So for any research work or validation of any design modification we usually adapt numerical method rather than analytical method.

CFD is a process to solve a flow problem with the help of numerical methods. In this method we firstly identify the transport equation for the problem and then impose boundary conditions on it. The general expression of transport equation is actually derived from generalized Navier-Stokes equation. This transport equation may be expressed generally in the following form-

$$\frac{\partial(\rho\alpha)}{\partial x}+div(\rho\alpha\vec{\mathbf{V}})=div(\Gamma\ grad\ \Phi)+S_{\Phi} \qquad (17)$$

Here we have considered 'α' as any property of the flowing fluid.

After identifying the correct transport equation, we would discretize the fluid flow domain into a number of parts. This process is called 'meshing'. After meshing, we identify different boundary of the flow domain with some easy understandable name under different pre-defined category. Now we impose properly the flowing fluid property and also take decision whether energy conservation equation has to be considered or not.

Next we have to identify properly the other boundary conditions to complete the model definition stage. After completing the definition the software is instructed to solve the problem and the software solve the problem by constructing a matrix and solving it with a predefined algorithm like 'Semi-Implicit Method for Pressure Linked Equation' (SIMPLE) algorithm.

Once solution is completed by the software we can get many outputs as a part of post-processing stage. The outputs which we may get are like pressure distribution, velocity distribution, stress distribution, path line display of the flow, plotting of graphs between different quantities etc.

Here lies the utility of a CFD Software. If we wanted to investigate the above mentioned outputs manually we must have gone for physical testing. But many a disadvantages are associated with physical testing. It requires more financial investment and needs more time to be validated. Ultimately the idea of development loses its economical viability in this age of vast competitive market. On the other hand a numerical method can solve a fluid flow problem not only with a negligible error but also with minimum effort.

A number of simulation software on CFD is available in market. Fluent is the most popular and widely used amongst them. The software used in this project work to investigate the influence of surface texture on a Journal bearing is Fluent 6.3.26.

III. LITERATURE SURVEY

The current trend of modern industry is to use machineries rotating at high speed and carrying heavy rotor loads. In such applications hydrodynamic journal bearings are used. When a bearing operates at high speed, the heat generated due to large

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shearing rates in the lubricant film raises its temperature which lowers the viscosity of the lubricant and in turn affects the performance characteristics. Thermohydrodynamic (THD) analysis should therefore be carried out to obtain the realistic performance characteristics of the bearing. In the existing literature, several THD studies have been reported. 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. First a remarkable work on Thermohydrodynamic study of journal bearing was done by Hughes et al. (Ref. [1]) in the year of 1958. In their paper Hughes and his colleague found out a relation between viscosity as a function of temperature and pressure of the lubricant inside the journal bearing. In this work investigation of Hughes et al have been used to predict perfectly the pressure distribution on journal surface of Journal Bearing with dimension as per S Cupillard, S Glavatskih, and M J Cervantes (Ref. [7]) by simulating a 3-dimensional journal bearing model in Fluent 6.3.26. In the year of 2007 S. A. Gandjalikhan Nassab and M S Moayeri did a thermal analysis on a axially grooved journal bearing and showed the importance of thermohydrodynamic analysis of bearing. Besides this so many other scientists proved the inevitable importance of thermohydrodynamic study of journal bearing like Prakash Chandra Mishra's work (Ref. [4]) in the year of 2007 and in the same year Wei Wang, Kun Liu & Minghua Jiao did a remarkable work in this field. In the year of 2008 K.P. Gertzos, P.G. Nikolakopoulos & C.A. Papadopoulos investigated journal bearing performance with a Non-Newtonean fluid ie Bringham fluid considering the thermal effect. Recently in the year of 2010, Ravindra R. Navthar et al investigated stability of a Journal Bearing Themohydrodynamically.

IV. JOURNAL BEARING MODELING

In the process of model verification, first a smooth bearing of the following dimensions have been analyzed then two types of dimple have been considered as mentioned in reference [7].
The smooth journal bearing which have been analyzed first is having following dimensions as referred in [7]—

TABLE 1: INPUT DATA FOR BEARING ANALYSIS

Length of the bearing (L)	133mm
Radius of Shaft (R_s)	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)	0.0127 Pas $\left[\frac{Kg}{m.Sec}\right]$

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

I. Radius of Bearing (R_b) : (R_s + C) = 50.145mm
II. Attitude angle (φ): 68.4^0. (as per reference [7])
III. Eccentricity (e) : (ε × C) = (0.61 × 0.145) =0.08845mm.

Now details for cavitation model are as follows as per reference [7].

TABLE 2: PARAMETERS FOR CAVITATION MODEL.

Lubricant vapour saturation pressure	20 Kpa.
Ambient pressure	101.325 Kpa.
Density of lubricant vapour	1.2 kg/m3
Viscosity of lubricant vapour	2×10-5 Pas.
Assumed vapour bubble dia	1×10-5 m

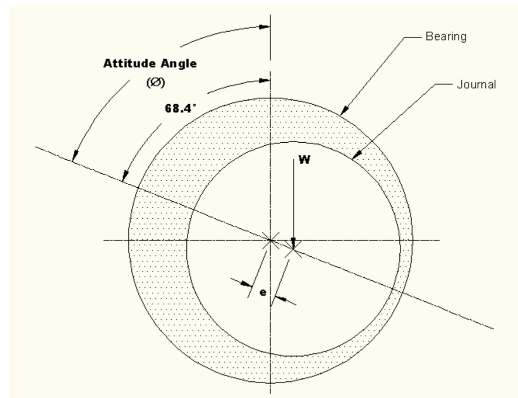

Fig 3.0: Schematic diagram of a smooth journal bearing

In their paper or work S Cupillard, S Glavatskih, and M J Cervantes have analyzed the above journal bearing without considering the temperature effect. In this work a plain journal bearing has been analyzed with the effect of temperature. After analyzing plain journal bearing, a textured journal bearing as per dimensions mentioned in reference [7].

To proceed in this analysis, first a 3-dimensional bearing has been generated in GAMBIT 2.3.16. Figures below show the 3d-geometry and meshed geometry in GAMBIT.

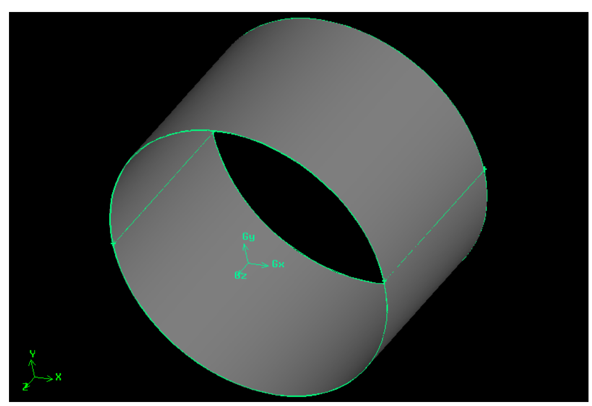
Fig 4.0: 3-dimensional representation of a smooth journal bearing in GAMBIT

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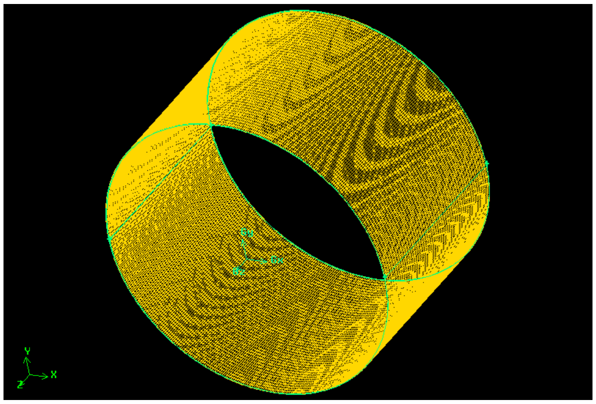

Fig 5.0: Meshed volume of a smooth journal bearing in GAMBIT

After generating meshed volume in GAMBIT next following boundary conditions have been fixed.

TABLE 3: NAME AND TYPES OF BOUNDARIES OF THE FLOW REGION.

SL NO.	BOUNDARY NAME	BOUNDARY TYPE
1	Middle cross-sectional plane	SYMETRY
2	End plane of the bearing	PRESSURE OUTLET
3	Journal surface	WALL
4	Bearing surface	WALL

V. MATHEMATICAL MODEL VALIDATION

After assigning boundary name and types of the flow region the file has been exported as '.msh' and then has been imported to the 'Fluent' software for CFD simulation.

In Fluent, data regarding chemical and physical properties of lubricant oil and properties of lubricant vapor, which have been mentioned in table1 and table 2, have been fed into the software. Here, mathematical parameters have also been set in the software.

TABLE 4: MATHEMATICAL PARAMETERS FOR CFD SIMULATION.

Pressure-Velocity Coupling	Discretization Methods			
	Pressure	Density	Momentum	Vapor
SIMPLE	PRESTO	Second order	Second Order	First order

After simulation pressure distribution on journal surface has been found out as contour representation. The pressure contour has been shown in figure below.

There are two stress distribution have been shown below. First figure depicts the stress distribution starting from the mid plane that is plane of symmetry of the bearing. Next figure expresses the pressure distribution starting from a cross-sectional plane at a distance of 10% of total bearing length from the plane of symmetry.

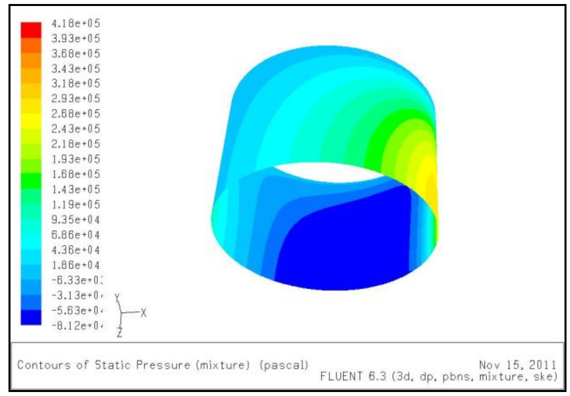

Fig 6: Pressure contour on Journal surface starting from plane of symmetry.

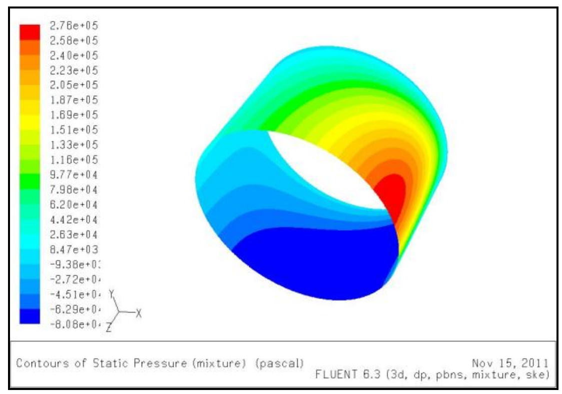

Fig 7.0: Pressure contour on Journal surface starting from a plane 10% of length.

The above pressure distribution on Journal surface of a Journal Bearing has been generated without considering the effect of temperature. The above result is very much in compliance with the work of S Cupillard, S Glavatskih, and M J Cervantes presented in reference [7]. But in their work Cupillard et. al. simulated a journal bearing with 2-Dimensional flow region. So, their work does not say about the pressure distribution along the length of bearing. In this work simulation has been done in 3-Dimensional flow region representing the actual lubricant flow of inside the bearing. So, the work presented in this thesis depicts more accurate pressure distribution in all 3-Dimensions.

In next section it will be shown that value of maximum pressure in pressure distribution on journal surface becomes less if we consider temperature effects.

VI. THERMAL CONSIDERATION

In previous section pressure distribution of a Journal Bearing has been shown without considering the effect of temperature on the properties of lubricating oil. In this section effect of temperature has been included and then pressure variation on the journal surface of the bearing has been evaluated.

To include the effect of temperature on the properties of the bearing oil in ANSYS a very beautiful mechanism is there in ANSYS software. This mechanism is known as "UDF" method. Full form of UDF is 'User Defined Function'. By this method one can append a governing function which would control the variation of any property of the fluid with respect to pressure or temperature or both. Here in this project following relation has been used to control the viscosity as a function of temperature and pressure. This equation has been adapted from the reference [1].

$$\mu = \mu_0 e^{\alpha(p-p_o)} e^{\beta(t-t_o)} \qquad (18)$$

The above equation has been appended to the ANSYS Fluent software through a C-Program with a ‘udf’ header file. The program has been shown below.

```
#include "udf.h"
DEFINE_PROPERTY(cell_viscosity,c,t)
{
real mu_lam;
real temp = C_T(c,t);
real pr = C_P(c,t);
mu_lam = 0.0127*exp(0.000000213345*(pr-101345))*exp(0.029*(temp-293));
```

Fig10: UDF program for controlling viscosity as function of temperature.

After appending this program to Fluent and analyzing it we get the following pressure distribution.

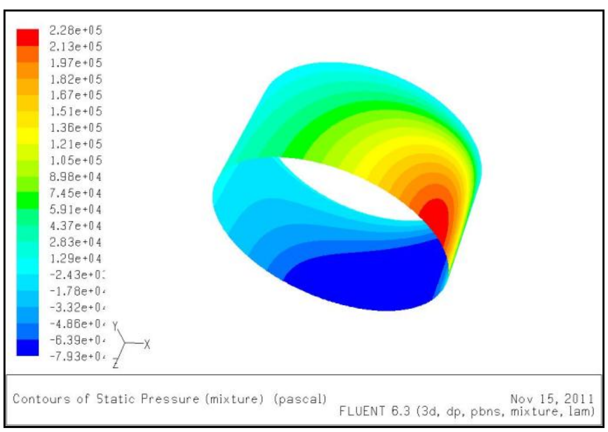

Fig 9 : Pressure distribution on journal surface considering temperature effect.

In above program we have used two terms α and β which are the pressure and temperature coefficient of viscosity and value of these quantities are 21.3345x10-8 m2/kg and 0.029/°K.

VII. RESULT AND DISCUSSION

From the above result it is clear that temperature created from the frictional force increases decreases the viscosity of the lubricant and lesser viscosity decreases the maximum pressure of the lubricant inside the bearing. For this reason it is recommended that when any analysis of journal bearing is done to measure its performance always thermohydrodynamic analysis should be used. Because considering the thermal effect on lubricant property actual value of performance parameters can only be obtained.

Now when the thermal analysis is done on the journal bearing temperature distribution has been obtained along the journal surface. Figure below represents the temperature variation of oil along the journal surface.

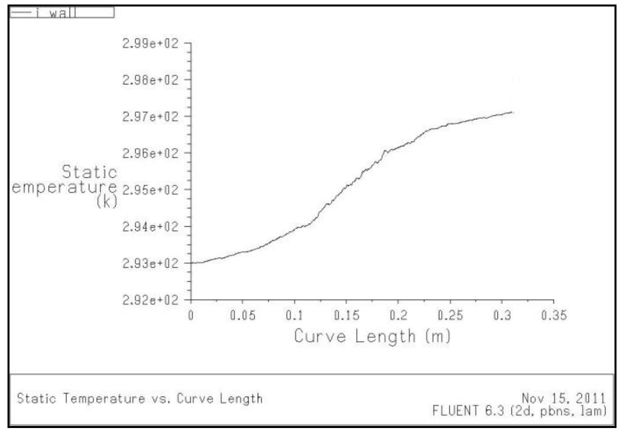

Fig 11.0: Temperature distribution on journal surface

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AUTHORS

First Author – Mukesh Sahu, mukeshsahu@yahoo.com
Second Author – Ashish Kumar Giri, agiri031@gmail.com
Third Author – Ashish Das, ashishdas.1110@gmail.com