TRANSIENT ANALYSIS OF 3-LOBE BEARINGS AT 80000 RPM FOR A GAS TURBINE

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ABSTRACT

In this paper the various parameters of the oil flow in a multi lobe bearing are calculated using unsteady k-epsilon turbulence model. For realizing the problem a 3 lobe bearing was selected which had the lobes placed at a distance of 120 degrees. The rotation speed of the shaft was considered to be 80000 rpm. The results show a strong affinity of the oil property to segregate to critical values at elevated rotational speeds. Thus the present study could lead towards the formulation of new bearing oil which corresponds to higher performance indices. The results show the presence of lobes highly effect the performance of the multilobe bearing as the critical quantities developed here are comparatively lesser to the other zones in the bearing.

KEYWORDS

Multi-lobe, viscosity, turbulence, dissipation, pressure, wall shear stress.

1. INTRODUCTION

Nearly all heavy industrial turbo machines use fluid film bearings of some type to support the shaft weight and control motions caused by unbalance forces, aerodynamic forces, and external excitations from seals and couplings. The two primary advantages of fluid film bearings over rolling element bearings are their superior ability to absorb energy to dampen vibrations, and their longevity due to the absence of rolling contact stresses. The damping is very important in many types of rotating machines where the fluid film bearings are often the primary source of the energy absorption needed to control vibrations. Fluid film journal bearings also play a major role in determining rot dynamic stability, making their careful selection and application a crucial step in the development of superior rotor-bearing systems.

Fixed-geometry bearings differ from tilting pad bearings in that the fixed-geometry bearing has no moving parts, making the lobes or arcs stationary around the shaft. As the shaft is forced from its centered position under the downward load, the bearing clearance becomes a converging-diverging wedge. Oil is supplied through two axial grooves located diametrally opposite each other at the bearing horizontal split line. After entering the arc leading edge, the oil is drawn by shaft friction into the converging radial clearance where it is compressed to a much higher pressure, giving the bearing its load carrying capability. Notice that the shaft does not move vertically downward under the vertical load but, rather, also moves in the horizontal (positive X) direction as well. This is because of the cross-coupling effects that are inherent to fixed-geometry journal bearings. These effects can contribute to rotor dynamic instability in some applications.

1.1 Objective

The objective of the present work is to design 3 lobe bearing and to analyze the various flow parameters arising due to the motion of the shaft at rpm of 80000. The design of the 3 lobe model was done using GAMBIT and its subsequent analysis and simulation was carried out using FLUENT.

1.2 Theory On Multilobe Bearing

Distributed across the entire shaft diameter, there are as many individual hydrodynamic carrying forces directed at the centre of the shaft as there are lobes. The strength of the individual hydrodynamic force is, among other things, dependent on the width of the wedge gap. The vector total of all the individual carrying forces represents the effective load capacity of the MF bearing towards the outside. This results in a strong centering effect being applied to the shaft which produces good concentricity and generates a defined shaft position. By matching the lubricant viscosity to the shaft´s peripheral speed and the wedge gap shape, the degree of the hydrodynamic carrying force and the bearing friction can be varied to meet individual requirements. The same wedge principle is applied to thrust bearings. In combination with a journal bearing, a specific number of taper land faces is created on one or both faces of the MF bearing. Stanislaw Strzelecki [1] worked on “Effect of lobe profile on the load capacity of 2-lobe journal bearing”. His main findings are the following. The results of calculations of load capacity of 2-lobe journal bearing characterized by different profiles of upper and bottom lobe. The load capacity of combined 2-lobe journal bearing type 2-LCOF is smaller than the load capacity of 2-lobe and Offset-Halves one. At assumed bearing type and bearing aspect ratio an increase in lobe relative clearance causes the decrease in load capacity of combined and another considered 2-lobe bearings. Except of 2-lobe bearing with offset upper half and cylindrical bottom one, the largest load capacity shows the 2-lobe journal bearing, particularly in the range of larger relative eccentricities. All considered 2-lobe bearings show small differences in the values of load capacity for the lower range of relative eccentricities of bearings.

Stanislaw Strzelecki and Sobhy M. Ghonheam [2] worked on “Dynamically loaded cylindrical journal bearing with recess” .Their main findings are given below. The profile of the journal centre trajectory changes with the presence of recess in it. They considered two types of bearing load one characterized by internal combustion engine and other by needle punching machine. They also calculate the journal centre trajectory with and hence found out various parameters oil film pressure distribution and oil film resultant force. They eventually found out that the trajectory is affected by the presence of recess. The presence of recess on the peripheral position of the bearing affects the trajectory too; hence this method could be subsequently applied to the study of multi- lobe bearing.

Sobhy M. Ghoneam and Stanislaw Strzelecki [3] worked on “Thermal problems of multilobe journal bearing tribosystem” .Their main findings are cited below. They found an approximate method for finding the condition of the lubricating oil film temperature. Oil film temperature was obtained from the basis of the known quantities like Reynolds’s number and viscosity equation based on empirical calculations and theoretical data. It could help in solving the problems related to 4-lobe bearing with known parameters. The oil film temperature distribution and maximum oil film temperature have been obtained from the numerical solution of bearing geometry, Reynolds, energy and viscosity equations.

Edmund A. Memmott and Oscar De Santiago [4] worked on “A classical sleeve bearing instability in an overhung compressor” .Their main findings are described below.

They enumerated the use of 2- lobe lemon bore bearings to solve the problem. As sleeve bearings are incapable to solve the various conditions required to extensively increasing speed and vibration the introduction of the lemon bore bearing with suitable lubricating oil have been proposed. The bearing that was installed on the high-pressure side of the compressor actually worked as a seal. They also proposed the solution of the problem with a bearing seal arrangement with 4- lobe bearing system with moderate preload. They also suggested that substitution of this bearing by a tilt pad seal/bearing solved the root problem and allowed satisfactory operation of the compressor.

Dr G Bhushan, Dr S Rattan, Dr N P Mehta [5] worked on “Effect of Pressure Dams and Relief-tracks on the Performance of a Four-lobe Bearing” .Their main findings is the following. The presence of pressure dams and relief cracks on the performance of an ordinary four lobe bearing. The generation of pressures and their circumferential variation in the upper half of a bearing primarily affect the stability of a rotor bearing system. In qualitative terms, the proportion of hydrodynamic load generated in upper half with respect to load generated in lower half is one of the deciding factors as to how stable a bearing would be. The magnitudes and pressures generated in the lobes of the four-lobe bearing without and with dam indicate that the latter would provide a relatively smoother operation of the bearing. A four-lobe pressure dam bearing operates in the higher range of eccentricity ratios compared to an ordinary four-lobe bearing. There is a marginal increase in the dimensionless friction coefficient when pressure dams are incorporated in an ordinary four-lobe bearing. The stability of an ordinary four-lobe bearing increases when pressure dams and relief-tracks are incorporated in it.

F.A Martin and A.V. Ruddy [6] worked on “The effect of manufacturing tolerances on the stability of profile bore bearings” .Their main findings is presented below. The introduction of new quantities of speed independent of the clearance and clearance independent of speed. They give a more precise analysis to problem than quantities like M’ and W’ which arise due to various factors and are not independent of machining allowances. The method could be well implemented for 4-lobe bearing. The tighter bearing tolerances results to higher instability at increased condition of speed and turbulence as there is no chance of loss of thermal quantities over them. They categorized the tolerances in two distinct parts like tolerances on the shaft and the tolerances on the bearing itself. Both these clearances have a distinct role in the instability in the bearings caused at very high speeds. The importance of considering the tolerances is based on the fact that tighter tolerances result in the higher instability like vibrations, overheat and wear and tear.

Raghunandana. K. [7] worked on “Inverse Design Methodology for the Stability Design of Elliptical Bearings Operating with Non-Newtonian Lubricants” .Their main findings are described below. The lubrication being considered Newtonian in nature incidentally allows in error in calculation of various critical parameters. This study provides steady state results for different L/D and eccentricity ratios in the form of empirical equations, hence the simulation with the various data and with the aid of computational methods various factors like oil film density and oil film viscosity could be found out for various NON-NEWTONIAN fluids and for BINGHAM plastics too.

J.D Knight and L.E. Barrett [8] worked on “An Approximate Solution Technique for Multilobe Journal Bearings Including Thermal Effects, with Comparison to Experiment”. Their main findings are cited below. They proposed an approximate solution method for multilobe journal bearings that includes thermal effect. Comparison of solutions obtained by the variable viscosity method to effective viscosity solutions after Lund and Thomsen illustrates discrepancies in operating eccentricity and stiffness coefficients between the two approaches. They also derived a very good co-relation between the variable viscosity solutions and experimental measurements reported by Tonnesen and Hansen of eccentricity, pressures, and temperatures in a two-axial groove bearing.

2. METHODOLOGY

In this part we aim towards the formulation of the problem and realization of constraints and pre and post defining the problem. The main objectives in this stage were:

- To find the pressure distribution across the various parts of the oil media as well as the shaft in an unsteady condition.
- To find the temperature distribution across the oil media and the shaft body in an unsteady condition.
- To find the various other quantities across the oil media and the shaft body in an unsteady condition.

2.1 Mathematical Formulation:

Mathematical model can be defined as the combination of dependent and independent variables and relative parameters in the form of a set of differential equations which defines and governs the physical phenomenon. In the following subsections differential form of the governing equation are provided according to the computational model and their corresponding approximation and idealizations.

2.2 Governing Equations

The steady, conservative form of Navier-Stokes equations in two dimensional forms for the incompressible flow of a constant viscosity fluid are as follows:

Continuity:

$$\frac{\partial U}{\partial X} + \frac{\partial V}{\partial Y} = 0$$

(1)

X- momentum:

$$\frac{\partial (UU)}{\partial X} + \frac{\partial (VU)}{\partial Y} = -\frac{\partial P_n}{\partial X} + \frac{1}{R_e}\left(\frac{\partial^2 U}{\partial X^2} + \frac{\partial^2 U}{\partial Y^2}\right)$$

(2)

Y- momentum:

$$\frac{\partial (UV)}{\partial X} + \frac{\partial (VV)}{\partial Y} = -\frac{\partial P_n}{\partial X} + \frac{1}{R_e}\left(\frac{\partial^2 V}{\partial X^2} + \frac{\partial^2 V}{\partial Y^2}\right)$$

(3)

where,

$$X=\frac{x}{D}\ , Y = \frac{y}{D}\ , P_n = \frac{p}{\rho u_\infty{}^2}\ , U = \frac{u}{u_\infty}, V = \frac{v}{u_\infty}\ ,\quad R_e=\frac{\rho u_\infty D}{\mu}$$

2.3 Transport Equation For The Standard K-E Model

The simplest and most widely used two-equation turbulence model is the standard k-ε model that solves two separate transport equations to allow the turbulent kinetic energy and its dissipation rate to be independently determined. The transport equations for k and ε in the standard k- ε model are:

$$\rho \frac{DK}{Dt} = \frac{\partial}{\partial x_i}[(\mu + \frac{\mu_t}{\partial_k})\frac{\partial k}{\partial x_i}] + G_k + G_b - \rho\varepsilon - Y_M$$
(4)

$$\rho \frac{D\varepsilon}{Dt} = \frac{\partial}{\partial x_i}[(\mu + \frac{\mu_t}{\partial_\varepsilon})\frac{\partial \varepsilon}{\partial x_i}] + C_{1_\varepsilon}\frac{\varepsilon}{k}(G_k + C_{3_\varepsilon}Gb) - C_{2_\varepsilon}\rho\frac{\varepsilon^2}{k}$$
(5)

where turbulent viscosity,

$$\mu_t = \rho C\mu \frac{k^2}{\varepsilon}$$

In these equations, G_k represents the generation of turbulence kinetic energy due to the mean velocity gradients. G_b is the generation of turbulence kinetic energy due to buoyancy. σ_k and σ_ϵ are the turbulent Prandtl numbers for k and ε, respectively. All the variables including turbulent kinetic energy k, its dissipation rate ε are shared by the fluid and the volume fraction of each fluid in each computational volume is tracked throughout the domain.

In the present study, a three-dimensional numerical study of unsteady, static pressure distribution and temperature distribution across the various parts of the oil media as well as the shaft of the 3-lobe bearing.

2.4 Exporting the Numerical Details

The 3 D modeling scheme was adopted in gambit and the various parts were analyzed using fluent. The following model was generated using GAMBIT.

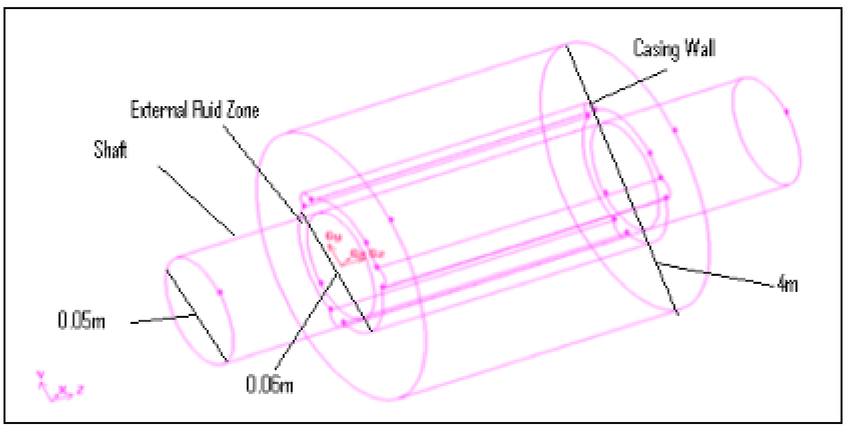

Fig 1: The wireframe diagram of the physical model

3. Defining The Physical Model

For purpose of defining the physical model we used the following values for the shaft and the bearing surface.

1. The bearing of .08 m was selected and the diameter was selected to be 0.06m.

2. The 3 lobes were placed 120 degrees apart whose diameter was 0.004 m.

The surface which holds the oil was assumed to be present between the shaft and the bearing surface area. The gambit model was drawn and consequently the consequently the various walls were selected. The walls were defined and the continuum was supposed to exist in the fluent state. The rest of the model continuum was supposed to be solid walls.

3.1 Meshing In Gambit

The part of the oil flooded region is meshed using GAMBIT. The model is exported to fluent for post analysis and results.

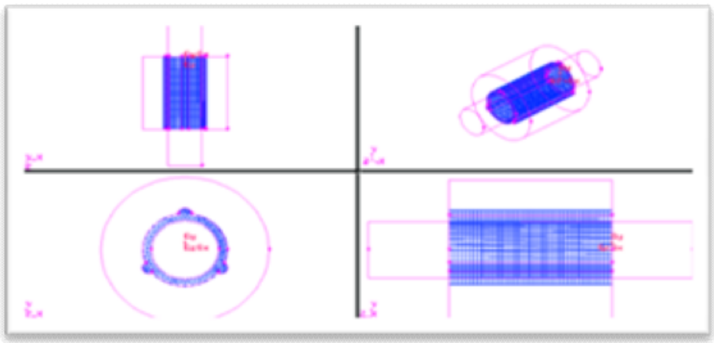

Fig 2: The 4-d view of the meshed part

3.2 Generation Of The Computational Domain

It involved transforming the generated physical domain into a mesh (structured/unstructured) with number of node points depending on the fineness of the mesh. The various flow properties were evaluated at these node points. The extent of accuracy of the result depended to a great extent on the fact that how fine the physical domain was meshed. After a particular refining limit the results changed no more. At this point it was said that grip independence was achieved. The results obtained particularly for this mesh were considered to be the best. This mesh formation was done with GAMBIT.

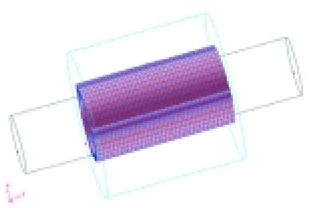

Fig 3: The meshed surface for analysis

4. GEOMETRY AND GRID ARRANGEMENT

The mesh file obtained from the gambit was exported to fluent for subsequent analysis. The mesh file was read using fluent and subsequently its grid checking was done the grid was checked with no error and the formation of one default surface at the boundary of the shaft and oil surface. The rest of the surfaces were defined in the similar manner. The pictorial representation of the various grids are shown here.

The following conditions were assigned to the various components of the exported file:-

Table 1: Defining the various walls and interfaces

Zone	Type
Fluid Wall Interface	Interior
Fluid	Fluid
Wall 1	Inlet pressure
Wall 2	Wall
Wall 3	Wall

4.1 Defining The Various Boundary Types

- **For Fluid**

The property of the fluid was defined in the following way:-
The lubricating fluid was considered to be SAE-50. The properties of the fluid were defined in the following way.

Table 2: Thermal property of the fluid SAE 50 For wall 1

Property	Value
Cp (kg-k)	2250
Thermal conductivity (W/m-k)	0.22
Viscosity (kg/m-s)	0.004
Density (kg/m^3)	839

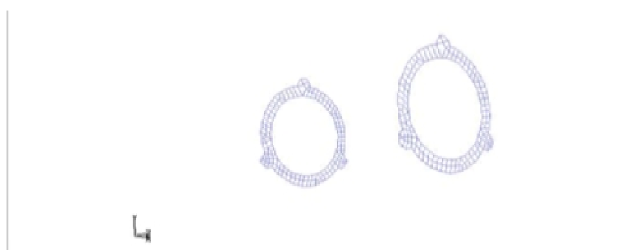

Fig 4: Grid display for wall 1

The nature of the wall surface was taken as inlet vet type. The various parameters considered are given below:

Table 3: Defining the boundary conditions for wall 1

Property	Value
Gauge Total Pressure	101325
Supersonic Pressure	0
Direction Specification Method	Normal to the boundary
Temperature	300

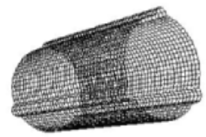

Fig 5: Grid display for wall 2

The wall was considered to be stationary with no slip condition and Marangoni stress. The wall thickness was considered to be negligible and the roughness constant was 0.5. The thermal conditions are illustrated below.

Table 4: Defining conditions for wall 2

Property	Value	Nature
Temperature(K)	300	CONSTANT
Heat Generation Rate(W/m3)	0	CONSTANT

The material for the wall 2 was considered to be copper and the various properties of copper used are as follows.

Table 5: Properties for wall 2 for wall 3

Property	Value
Density(kg/m3)	8030
Specific heat (j/kg-k)	502.48
Thermal Conductivity(W/m-k)	16.27

The wall 3 is also the shaft wall. The material for the shaft was chosen as steel. The various properties for the copper were defined as follows.

Table 6. Defining the material for wall 3

Property	Value
Density(kg/m3)	8030
Specific Heat (j/kg-k)	502.48
Thermal Conductivity (W/m-k)	16.27

FIG 6: Grid display for wall 3

The analysis was to be carried out for 80000 rpm. Thus the wall was defined as a rotational body having rpm of 80000.

Post Processing and Analysis

This involves the analysis of various contours and plots obtained from the analysis of fluent. A comparative analysis of the performance of multilobe bearing was carried at this various rpms and the results were displayed and analyzed using the FLUENT software.

5. RESULTS AND DISCUSSION

Analysis For Pressure Static Pressure

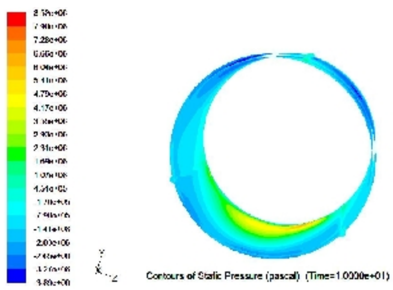

FIG-7(a)

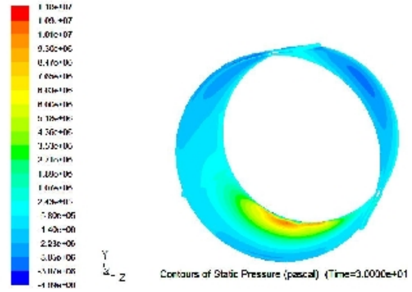

FIG-7(b)

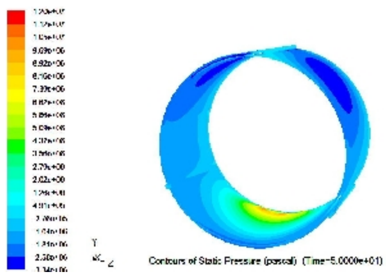

FIG-7(c)

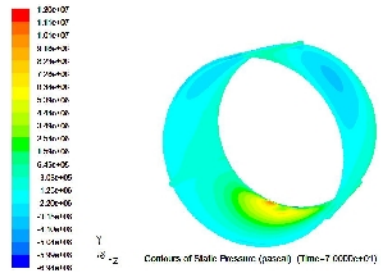

FIG-7(d)

FIG 7: Contours of Static pressure @ 80000 rpm

The distribution of static pressure in this case exists mainly on the top layer whereas the pressure almost remains constant on the inner side of the oil zone. The minimum value is the same i.e. 101325 whereas the maximum value is 8.81e+06 Pascal.

Analysis for Temperature

Static Temperature

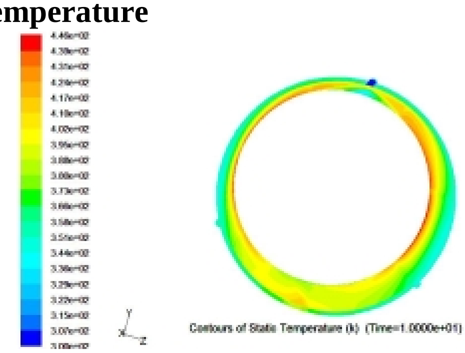

FIG-9(a)

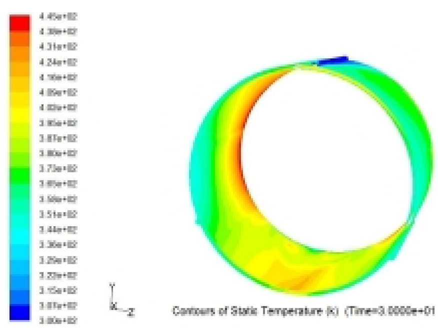

FIG-9(b)

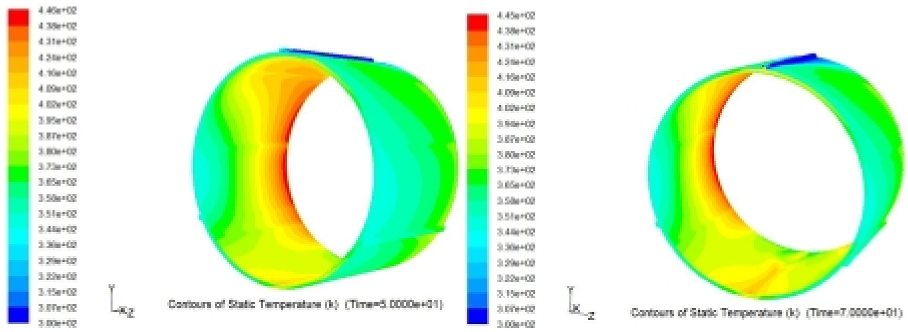

FIG-9(c) FIG-9(d)

FIG-9: Contours of Static temperature @ 80000 rpm

The distribution of static temperature is maximum near the middle of the shaft region. The rise in temperature is also contributed by the rotational speed of the shaft The contour is intensifies near the minimum middle of the shaft. The minimum and maximum values are 300K and around 445 K respectively.

6. CONCLUSION

As predicted the results tend to segregate to critical values at comparatively higher rotational speeds. The counters exhibit distinct pattern to give critical values of temperature and pressure near the interface of the wall and the surface of the shaft. Comparatively the afore said values are lower in the lobes which fulfils the justification of incorporating lobes in the ordinary bearings for very high speed applications. Though the values are comparatively on the higher side the paradox could be easily explained because the analysis were carried at practically at very high speed. Due to the steep rise in the temperature and pressure the oil could easily detain its lubricating properties. Hence the selections of proper lubricating oil as well as the material for the shaft and the bearing design have to be done judiciously. The present project could thus be suitably platform for carrying on this kind of studies in the future.

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