Analysis of rayleigh step bearings biology essay

According to importance and application of slider bearings in industries, investigation and analysis of this type of bearings are significant and inevitable issue. A widely used bearing type is the slider bearings with application in many cases such as different types of engines, compressors, turbines, electric motors and electric generators. To ensure that no contact occurs between the opposing surfaces, the dimensions of the bearing surface are chosen, such that a lubricant film of sufficient thickness is available under all operating conditions. The classical theory of hydrodynamic lubrication assumes that the inertia forces in the fluid film are negligible. For large bearings using low kinematic viscosity lubricant or for high speed, the inertia forces could be important. So the inertia terms should be entered in the calculations. This increases the accuracy of obtained responses and closes them to reliable results.

Rayleigh bearing is designed in 1918 by Lord Rayleigh. He was first person who considered the concept of optimization design in lubrication applications and obtained an optimum design for an infinite-length stepped bearing by the use of a variation technique (Lord Rayleigh, 1918). Since then, there have been some studies on the characteristics of step bearings. Dowson (1962) introduced the generalized Reynolds equation, which allows for cross-film temperature variations. Then, this equation solved with realistic THD boundary conditions by Ezzat and Rohde (1973) using the finite difference method. Boncompain, et al. (1986) improved the numerical model by considering reverse flow, fluid-film rupture and elastic deformations (THD solution). Auloge et al. (1983) studied the optimum design of Rayleigh step bearing and determined the relationships between step location and height along with non-Newtonian lubricants. The same method was used by Fillon and Khonsari (1996) in tracing design charts for tilting-pad journal bearings. Jianming and Gaobing (1989) have presented the optimum design of one-dimensional Rayleigh step bearing with non-Newtonian lubricants. Tello (2003) has theoretically studied the regularity of the solution to the Reynolds equation in Rayleigh step type bearings for both compressible and incompressible fluids by employing a rigorous mathematical approach. Besides, there are many research works in which the well known Reynolds equation was solved by different numerical schemes in predicting the lubricant pressure field in step bearings (Hideki, 2005; Dobrica and Fillon, 2005). Rahmani et al. (2009) comprehensively studied the Rayleigh step slider bearing including the effect of variations of pressure at the boundaries on the optimum parameters. The bearing is also optimized considering the lubricant flow rate, friction force and friction coefficient.

In all of the above studies, the Reynolds equation was solve as the governing equation for calculation of lubricant pressure distribution in bearing lubricant flow. This equation is a simplified form of the momentum equation by neglection of fluid inertia terms. It is clear that under the condition of low lubricant viscosity and high runner surface velocity, this equation may lead to unreliable results. In the present study which a numeric one, the two-dimensional Navier-Stokes and energy equations are solved by CFD method with considering the variation of lubricant viscosity with temperature. By this technique the THD characteristics of Rayleigh slider bearings running under different steady conditions are explored.

2. PROBLEM DESCRIPTION

The schematic and coordinate system of Rayleigh slider bearing is shown in Fig. 1. The bottom wall of the step bearing moves with constant velocity U (runner velocity). The sudden change in film thickness generates a hydrodynamic pressure field that supports an applied load W. At the inlet section, the oil film is entered at 40oC with combination of Poiseuille and Couette flows. The total length of the bearing is and the film thicknesses before and after the step location are and , respectively.

D: payanameThermohydrodynamic with subroutin. mesh moteghayerPLOTSgeometryasli. wmf

Fig. 1: Sketch of problem geometry

Two important geometrical factors in step bearings are

(1)

(2)

In these explanations, and represents the bearing length ratio and the bearing height ratio, respectively, which are two important bearing geometrical factors.

3. THEORY

3. 1. Governing equations

For lubricant flow in bearings, the governing equations which are written for a two-dimensional, steady, incompressible, laminar and variable viscosity flow consist of the continuity, Navier-Stokes and energy equations. These equations in non-dimensional forms can be written as:

(3)

(4)

(5)

(6)

Where and represent the dimensionless viscous source terms:

(7)

(8)

And is the dimensionless viscous dissipation term:

(9)

Also the dimensionless oil viscosity based on Vogel equation can be calculated as follows:

(10)

In this expression, is the temperature-viscosity coefficient of the lubricant. The value of can is determined using two given viscosity values at and as follows (Khonsari and Booser, 2008):

(11)

In equations 3 to 9, the following non-dimensional groups are used:

(12)

In these definitions, is the thermal diffusivity of the lubricant and is the inlet lubricant viscosity.

The main physical quantities of interest in lubrication study are the load capacity and friction force that can be computed using the lubricant velocity and temperature fields.

The load capacity of the step bearing per unit width is obtained by further integration of lubricant pressure distribution on the runner surface as follows:

(13)

The friction force of the step bearing per unit width is calculated by the shear stress on the bottom wall as follows:

(14)

Where:

(15)

3. 2. Boundary conditions

The entire domain is fully flooded, such that oil pressure at the inlet and outlet sections of the bearing is set to zero atmospheric gauge pressure. Also the no-slip condition is employed on all boundary solid walls.

At the inlet section, oil enters into bearing with uniform temperature of and a specified velocity distribution which is a combination of the poiseulle and cuette flows whose pressure gradient is determined by numerical solution of the Reynolds equation. At the outlet section, zero axial gradients for all dependent variables are employed. Finally, the adiabatic condition is imposed on all of the bearing solid surfaces.

4. SOLUTION PROCEDURE

Finite difference forms of the continuity, momentum and energy equations were obtained by integrating over an elemental cell volume with staggered control volumes for the x- and y- velocity components. Other variables of interest were computed at the grid nodes. The nondimensionalized governing equations were discretized by using the hybrid scheme and numerically solved by the SIMPLE algorithm of Patankar and Spalding (Patankar and Spalding, 1972). Numerical solutions were obtained iteratively by the line-by-line method progressing in axial direction. The iterations were terminated when the sum of the absolute residuals was less than for each equation. Numerical calculations were performed by writing a computer program in FORTRAN.

mesh asli11

Fig. 2: A schematic of grid generation

As shown in figure 2, the computational domain is divided into three blocks, each having Nx points in x-direction and Ny points in y-direction. The mesh is non-uniform in x- and y- directions, because the grid refinement around the step is necessary to capture the occurrence of the recirculation and other flow changes due to the sudden change in geometry. As the result of grid tests for obtaining the grid-independent solutions, an optimum grid is determined in grid study. Five different numbers of grid size inside the total rectangular computational domain including the blocked-off region with their related numerical results are listed in Table 1. According to this grid study, an optimum grid of 640120 is used in all of the subsequent test cases.

Table 1: Grid independent study,

Grid size

Bearing friction force (KN/m)

Bearing load capacity (KN/m)

45080

0. 231

16. 45

540110

0. 217

17. 17

590110

0. 231

17. 48

640120

0. 239

17. 51

680140

0. 241

17. 52

5. VALIDATION OF NUMERICAL RESULTS

To test the validity of the present numerical results, computations were carried out for a test case and the computed results were compared with the theoretical findings by other investigators. The lubricant pressure distribution on the bottom wall and the temperature distribution on the top wall of the Hideki bearing (Hideki, 2005) are shown in Figs. 3 and 4, respectively.

D: payanameThermohydrodynamic with subroutin. mesh moteghayervalidation with Ogata(temprature)Plot validation with OgataCJS. wmf

Fig. 3: Lubricant pressure distribution on the bottom wall of the

Hideki bearing (Hideki, 2005),

The generated hydrodynamic pressure by the sudden contraction in flow domain is clearly seen in Fig. 3, such that at the entrance of narrow gap of the bearing, the maximum lubricant pressure occurs, and at the inlet and outlet sections, lubricant flow in at atmospheric pressure (zero gauge pressure).

D: payanameThermohydrodynamic with subroutin. mesh moteghayervalidation with Ogata(temprature)Plot validation with OgataCJS temp. wmf

Fig. 4: Temperature distribution on the top wall of the

Hideki bearing (Hideki, 2005),

Fig. 4 shows that the lubricant temperature increases along the flow direction because of the viscous dissipation in both domains upstream and downstream of the step. Such that, the rate of temperature increase in upstream region to the step is very greater than that is in downstream domain. It is due to this fact that the viscous dissipation in lubricant flow with small film thickness is high in comparison to lubricant flow with large film thickness. However, good consistencies are observed between the present numerical results with theoretical findings by Hideki (Hideki, 2005) about computations of both lubricant pressure and temperature distributions.

6. RESULT AND DISCUSSION

In this research work, the THD characteristics of Rayleigh step bearings are obtained by numerical solution of the Navier-Stocks and energy equations using the CFD technique. An attempt is made for obtaining the effects of important parameters including the runner surface velocity, bearing length ratio and bearing height ratio on thermal and hydrodynamic behaviors of Rayleigh step bearings. All of the subsequent figures are about a Rayleigh step bearing whose properties and geometrical parameters are given in Table 2.

Table 2: Bearing parameters and lubricant properties

Parameters

Units

Values in present work

b

m

0. 08-0. 12

h1

µm

480

U

m/s

10-30

Tin

40

ρ

860

Cp

2000

Kf

0. 13

µ at 40 C

0. 03

µ at 100 C

0. 0045

—

0. 28-0. 98

—

1. 2-2. 5

First the oil flow pattern inside the bearing is shown in figure 5 by plotting the fluid velocity vectors. The adverse pressure gradient in the upstream flow domain before the step location which leads to hydrodynamic pressure generation causes a concave shape for velocity distribution. Such that the velocity distribution changes to convex shape after the step where there is a favorable pressure gradient. Behind the step surface near to the stationary wall, a circulated flow domain happens which is due to the effects of both viscous friction and positive pressure gradient in this region. As another result that can be seen from Fig. 5, one can notice to almost stationary flow region in block 1 (see Fig. 2). Therefore, the lubricant average velocity across blocks 2 and 3 remains approximately constant.

C: UserszahraDesktopUntitled. png

Fig. 5: Velocity vectors in step bearing lubricant flow,

In Fig. 6, the lubricant pressure distributions along the bottom wall at five different values for the runner surface velocity are shown. It is seen that the velocity of moving surface has considerable effect on the value of generate hydrodynamic lubricant pressure, such that oil pressure has an increase trend by increase in velocity under a unique pattern.

D: payanameThermohydrodynamic with subroutin. mesh moteghayerPLOTScompare of PKcompare of speed runner. pkspeed runner. wmf

Fig. 6: Effect of runner surface velocity on lubricant pressure distribution

along the bottom wall,

A similar study is done for investigating the effect of runner velocity on thermal behaviour of step bearing in Fig. 7. It is seen in this figure that bearings with high runner surface velocity operate under high temperature condition. Besides, it is depicted in Fig. 7 that in both domains before and after the step, lubricant temperature increases along the flow direction because of the viscous dissipation. Also, it is seen that the oil temperature at the outlet section is affected strongly by the runner velocity, such that the bearings with high velocity have high temperature lubricant flow at their outlet sections.

D: payanameThermohydrodynamic with subroutin. mesh moteghayerPLOTScompare of TPcompare speed of runner. TpTshaftTshaft. wmf

Fig. 7: Effect of runner surface velocity on lubricant temperature distribution

along the bottom wall,

D: payanameThermohydrodynamic with subroutin. mesh moteghayerPLOTScompare of PKcompare of epsilonepsilon. wmf

Fig. 8: Effect of bearing length ratio on lubricant pressure distribution

along the runner surface,

The lubricant pressure distributions along the runner surface at four different values of the bearing length ratios are illustrated in Fig. 8. It is evident that the location of maximum pressure moves toward the downstream side by increasing in bearing length ratio, because the step location moves toward this sense when increases. Besides, it can be found from Fig. 8 that there is an optimum value for bearing length ratio to obtain the most value for lubricant maximum hydrodynamic pressure. It is depicted in Fig. 8 that this value for bearing length ratio in this test case is. Therefore, is an important parameter in step bearings that has great effects on lubricant pressure and consequently in bearing load capacity.

The effect of bearing length ratio of thermal behavior of step bearing is studied in Fig. 9 by plotting the lubricant temperature distributions on the runner surface for bearings with different length ratios. This figure shows that the effect of on temperature distribution is less than its effect of the hydrodynamic lubricant pressure. However, this figure depicts that bearings with length ratio greater that run cooler than the bearings with small less than.

D: payanameThermohydrodynamic with subroutin. mesh moteghayerPLOTScompare of TPcompare of epsilonCompare Tshaft of epsilonTshaft. wmf

Fig. 9: Effect of bearing length ration on lubricant temperature distribution

along the runner surface,

The variations of lubricant maximum pressure and temperature with bearing length ratio are presented in Fig. 10. This figure reveals the same trends for THD characteristics of step bearing those have been shown in the previous figures.

D: payanameThermohydrodynamic with subroutin. mesh moteghayerPLOTSplot Epsilon moteghayer(W. F. eta. etam)compare of ep. P. T. wmf

Fig. 10: Variations of lubricant maximum pressure and lubricant maximum temperature

with bearing length ratio,

In order to study more about the effect of bearing length ratio on THD characteristics of step bearings, the variations of bearing load capacity and friction force with are plotted in Fig. 11.

This figure presents that there is a maximum value for load capacity that takes place at

= 0. 718. Besides, it is revealed from Fig. 11 that in bearings with high length ratio, low friction force exists in comparison to bearings with small values for .

D: payanameThermohydrodynamic with subroutin. mesh moteghayerPLOTSplot Epsilon moteghayer(W. F. eta. etam)compare of ep. w. f. wmf

Fig. 11: Variations of load capacity and friction force with bearing length ratio,

Similar study is also done for investigating the effect of bearing height ratio on THD characteristics of step bearings by plotting the lubricant pressure and temperature distributions and also the variations of load capacity and friction force with various values of the parameter . According to Figs. 12 and 13, it is revealed that the values of lubricant pressure and temperature increase by increasing in bearing height ratio.

D: payanameThermohydrodynamic with subroutin. mesh moteghayerPLOTScompare of PKcompare of ksiksi. wmf

Fig. 12: Effect of bearing height ratio on lubricant pressure distribution

along the runner surface,

D: payanameThermohydrodynamic with subroutin. mesh moteghayerPLOTScompare of TPcompare of ksiCompare Tshaft of kesiTshaft. wmf

Fig. 13: Effect of bearing height ration on lubricant temperature distribution

along the runner surface,

This behavior is also presented by Fig. 14 in which the variations of maximum lubricant pressure and temperature are plotted with bearing height ratio. It is seen that both and have increasing trends with increase in the value of , such that the rate of increase in maximum temperature is greater than that is in maximum pressure.

D: payanameThermohydrodynamic with subroutin. mesh moteghayerPLOTSplot Kesi moteghayer(W. F. eta. etam)compare of ksi. P. T. wmf

Fig. 14: Variations of lubricant maximum pressure and lubricant maximum temperature

with bearing height ratio,

Fig. 15 shows a similar trend for bearing load capacity and friction force with the variation of height ratio. Such that it is seen in this figure that both load capacity and friction force increase with increasing in bearing height ratio.

D: payanameThermohydrodynamic with subroutin. mesh moteghayerPLOTSplot Kesi moteghayer(W. F. eta. etam)compare of ksi. W. F. wmf

Fig. 15: Variations of load capacity and friction force with bearing height ratio,

In the following figures, an attempt is made to verify the influences of bearing length, b, on the THD characteristics of step bearings.

D: payanameThermohydrodynamic with subroutin. mesh moteghayerPLOTScompare of PKcompare of length bearingcpmpare b. wmf

Fig. 16: Effect of bearing length on lubricant pressure distribution

along the bottom wall,

D: payanameThermohydrodynamic with subroutin. mesh moteghayerPLOTSplot Lenght moteghayer(W. F. eta. etam)TshaftTshaft. wmf

Fig. 17: Effect of bearing length on lubricant temperature distribution

along the bottom wall,

It is seen from Figs. 16 to 19 that in long bearings, the values of lubricant pressure and temperature and consequently the amounts of maximum pressure and temperature are high that leads to have high bearing load capacity and bearing friction force.

D: payanameThermohydrodynamic with subroutin. mesh moteghayerPLOTSplot Lenght moteghayer(W. F. eta. etam)compare of length. p, T. without point. wmf

Fig. 18: Variations of lubricant maximum pressure and lubricant maximum temperature with bearing length,

D: payanameThermohydrodynamic with subroutin. mesh moteghayerPLOTSplot Lenght moteghayer(W. F. eta. etam)compare of length. w, f. without point. wmf

Fig. 19: Variations of load capacity and friction force with bearing length,

7. CONCLUSION

This paper deals a numerical study for investigating the THD characteristics of Rayleigh step bearings running under different steady conditions. The set of governing equations consisting of the Navier-Stokes and energy equations is solved by the CFD technique and the variation of lubricant viscosity with temperature is also considered into account. This mathematical model and numerical method lead to more accurate numerical results in comparison to those obtained before by other investigation with numerical solution of the Reynolds equation that neglects the fluid inertia terms. It is found that the thermal and hydrodynamic behaviors of step bearing are affected considerably by the runner surface velocity and the bearing geometrical factors.

Nomenclature

B

bearing length

dimensionless velocity components

upstream bearing length

load capacity of bearing

downstream bearing length

horizontal and vertical coordinates

Cp

heat capacity

dimensionless coordinates

friction force of bearing

upstream film thickness

Greek symbols

downstream film thickness

friction coefficient

step height

modified friction coefficient

Kf

thermal conductivity

dynamic viscosity

width of bearing

dimensionless dynamic viscosity

origin of coordinate

μ1

dynamic viscosity at

pressure

μ2

dynamic viscosity at

pressure at the inlet

kinematic viscosity

pressure at the outlet

density

dimensionless pressure

shear stress

Pr

Prandtl number

Pe

Peclet number

Subscripts

Re

Reynolds number

fluid

Temperature

inlet

Tin

inlet temperature

maximum

dimensionless temperature

surface

runner velocity

velocity components