Thermal Characteristics of Offset-Halves Bearing With Offset Factor

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Abstract- The Reynold's equation and energy equation has been solved using finite difference method to study the effect of variation in offset factor on the performance characteristics such as oil-film temperatures, thermal pressures, load carrying capacity, power loss, and Sommerfeld's number of offset-halves journal bearing. Numerical solution of Energy equation is achieved using Parabolic Temperature Profile Approximation technique for faster computation. It has been observed that while keeping the radial clearance constant, an increase in offset factor results in a decrease in oil-film temperature, thermal pressure, load carrying capacity and an increase in Sommerfeld's number. Finally, it has been concluded that the offset factor may be kept equal to 0.4 while designing offset-halves journal bearing, and should be preferred to get the desired thermal characteristics.

Index Terms— Offset-halves bearing**,** oil-film temperature, thermal pressure, Offset factor, load carrying capacity

I. Introduction

A hydrodynamic journal bearing finds an extensive use in high speed rotating machinery. The journal bearings operate under hydrodynamic lubrication regime and in this regime, a thick film of lubricant separates the bounding surfaces. Under the normal operating conditions, hydrodynamic journal bearings usually experiences a considerable variation in oil-film temperature due to viscous heat dissipation. This significantly affects the bearing performance as lubricant viscosity is a strong function of temperature and, consequently, leads to failure of the bearing. Computation of oil-film temperature is of great importance to predict bearing performance parameters. The non-conventional journal bearings, like lobed bearings and tilting pad bearings have a common feature that these bearings operate with more than one active oil film which accounts for superior stiffness and damping characteristics of these bearings as compared to the conventional circular bearings. The offset-halves journal bearing is commonly used as a lobed bearing in which two lobes are obtained by orthogonally displacing the two halves of a cylindrical bearing (Fig. 1) and frequently used in gear boxes connecting turbine and generator in power generation industries. In the design of offset-halves halves bearing, offset factor play an important role and is defined as the ratio of the minimum clearance to the radial clearance.

Offset-halves journal bearings show stiffness and damping properties which permit light loads at high rotational speeds [1]. The importance of thermal effects in hydrodynamic journal bearings has been long recognized, but very limited study about thermal effects in lobed bearing especially offset-halves bearing has been reported in literature. A. Chauhan et al. [2] has carried out a comparative study for rise in oil-film temperatures, thermal pressures and load capacity for three different commercially available grade oils have been carried out.

Manuscript received March 05, 2013;
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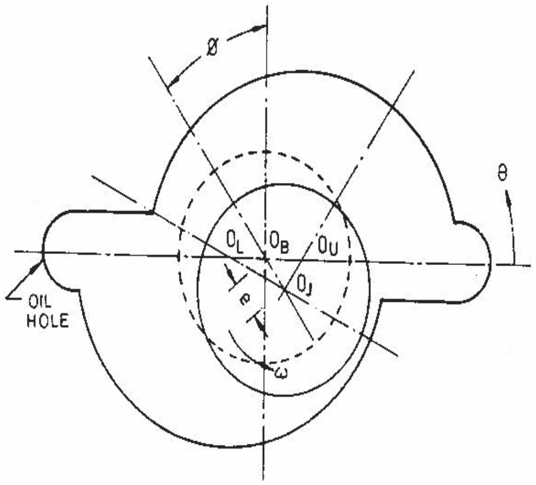

Fig. 1 Schematic diagram of offset-halves journal bearing

The authors have reported that with increase in speed, oil-film temperature, thermal pressure and load carrying capacity rises for all grade oils under study. Also, A. Chauhan et al. [3] have analyzed thermal performance of elliptical and offset-halves bearings by solving energy equation while assuming parabolic temperature profile approximation across the fluid film. Authors have been found that offset-halves journal bearing runs cooler with minimum power loss and good load capacity. An attempt was made by T. Suganami and A. Z. Sezri [4] to formulate a thermohydrodynamic model of film lubrication which is valid in both laminar and superlaminar flow regimes. The authors stated that energy equation retains heat conduction in direction of sliding motion, and is applicable even at large eccentricities. R. Boncompain et al. [5] have presented a general thermohydrodynamic theory. The authors have solved generalized Reynolds equation, energy equation in film and heat transfer equation in bush and shaft simultaneously. T. P. Indulekha et al. [6] have solved three dimensional momentum and continuity equations, and three dimensional energy equations to obtain pressure, velocity and temperature field in the fluid of a hydrodynamic circular journal bearing using finite element method. Authors have computed attitude angle, end leakage and power loss, for a wide range of eccentricity ratios. A. Hussain et al. [7] have predicted temperature distribution in non-circular journal bearings (namely two-lobe, elliptical and orthogonally displaced). The work is based on a two-dimensional treatment following Mc Callion's approach (an approach in which Reynolds and energy equations in oil-film are decoupled by neglecting all pressure terms in energy equation). R. Sehgal et al. [8] have presented a comparative theoretical analysis of three types of hydrodynamic journal bearing configurations namely: circular, axial groove, and offset-halves. It has been observed by the authors that the offset-halves bearing runs cooler than an equivalent circular bearing with axial grooves. A computer-aided design of hydrodynamic journal bearing is provided considering thermal effects by D. S. Singh and B. C. Majumdar [9]. In this design, Reynolds equation has been solved simultaneously along with energy equation and heat conduction equations in bush and shaft to obtain steady-state solution and a data bank is generated that consists of load, friction factor and flow rate for different L/D and eccentricity ratios. R. K. Sharma and R. K. Pandey [10] have carried out a thermohydrodynamic lubrication analysis of infinitely wide slider bearing assuming parabolic and Legendre polynomial temperature profile across film thickness. Authors have observed that temperature profile approximation across the film thickness by Legendre Polynomial yields more accurate results in comparison to Parabolic Temperature Profile approximation.

Evaluation of oil-film temperatures, thermal pressures, load carrying capacity, power loss, and Sommerfeld's number has been carried out using Parabolic Temperature Profile Approximation (PTPA) in the fluid film of an offset-halves journal bearing for the Mak Multigrade oil. Mak Multigrade oil is recommended for use in heavy duty commercial vehicles, light commercial vehicles and multi-utility vehicles fitted with high speed naturally aspirated or turbo charged diesel engines operating at low speed and high torque conditions [2].

II. GOVERNING EQUATIONS

Reynolds equation:

For steady-state and incompressible flow, Reynolds equation is [7]:

$$\frac{\partial}{\partial x}\left(\frac{h^3}{\mu}\frac{\partial p}{\partial x}\right)+\frac{\partial}{\partial z}\left(\frac{h^3}{\mu}\frac{\partial p}{\partial z}\right)=6U\frac{\partial h}{\partial x} \qquad (1)$$

ISBN: 978-988-19252-9-9
ISSN: 2078-0958 (Print); ISSN: 2078-0966 (Online)

This equation is then set into finite differences by using central difference technique. The variation of viscosity with temperature and pressure has been simulated using following relation:

$$\mu = \mu_{ref} e^{\alpha P - \gamma (T - T_0)} \quad (2)$$

Energy Equation:

The energy equation for steady-state and incompressible flow is given as [10]:

$$\rho C_P \left(u \frac{\partial T}{\partial x} + w \frac{\partial T}{\partial z} \right) = \frac{\partial}{\partial y} \left(K \frac{\partial T}{\partial y} \right) + \mu \left[\left(\frac{\partial u}{\partial y} \right)^2 + \left(\frac{\partial w}{\partial y} \right)^2 \right] \quad (3)$$

The term on left hand side in above equation represents energy transfer due to convection, whereas first term on right hand side represents energy transfer due to conduction and second term on right hand side represents energy transfer due to dissipation. The variation of temperature across film thickness in equation (3) is approximated by parabolic temperature profile for faster computation of temperatures [10]. The temperature profile across film thickness is represented by a second order polynomial as:

$$T = a_1 + a_2 y + a_3 y^2 \quad (4)$$

In order to evaluate constants appearing in eqn. (4), following boundary conditions are used:

At $y = 0, T = T_L$,

At $y = h, T = T_U$, $T_m = \frac{1}{h} \int_0^h T dy$

Thus, temperature profile expression (written in eqn. (4)) takes the following form:

$$T = T_L - \left(4T_L + 2T_U - 6T_m \right) \left(\frac{y}{h} \right) + \left(3T_L + 3T_U - 6T_m \right) \left(\frac{y}{h} \right)^2 \quad (5)$$

Where, T_L, T_U, and T_m represent temperatures of lower bounding surface, upper bounding surface, and mean temperature across film respectively.

Substitution of '*u*', '*w*', and '*T*' expressions (Eqns. (4), (5), and (6)) into energy equation (3) and subsequently integrating energy equation across film thickness from limit '0' to 'h' yields following form of energy equation.

$$6T_L + 6T_U - 12T_m - \frac{\rho C_P h^4}{120 K\mu} \frac{\partial P}{\partial x} \left(\frac{\partial T_L}{\partial x} + \frac{\partial T_U}{\partial x} - 12 \frac{\partial T_m}{\partial x} \right) - \frac{\rho C_P h^4}{120 K\mu} \frac{\partial P}{\partial z} \left(\frac{\partial T_L}{\partial z} + \frac{\partial T_U}{\partial z} - 12 \frac{\partial T_m}{\partial z} \right) - \frac{\rho C_P h^2 (u_L + u_U)}{2K} \frac{\partial T_m}{\partial x} - \frac{\rho C_P h^2 (u_U - u_L)}{12K} \left(\frac{\partial T_U}{\partial x} - \frac{\partial T_L}{\partial x} \right) + \frac{h^4}{12 K\mu} \left[\left(\frac{\partial P}{\partial x} \right)^2 + \left(\frac{\partial P}{\partial z} \right)^2 \right] + \frac{\mu (u_U - u_L)^2}{K} = 0 \quad (6)$$

The film thickness (*h*) equations for offset-halves journal bearing are given as [8]:

$$h = c_m \left[\left(\frac{1+\delta}{2\delta} \right) + \left(\frac{1-\delta}{2\delta} \right) \cos\theta - \varepsilon \sin(\phi - \theta) \right]$$

$(0 < \theta < 180)$ (7)

$$h = c_m \left[\left(\frac{1+\delta}{2\delta} \right) - \left(\frac{1-\delta}{2\delta} \right) \cos\theta - \varepsilon \sin(\phi - \theta) \right]$$

$(180 < \theta < 360)$ (8)

III. COMPUTATIONAL PROCEDURE

Numerical solution of Reynold's and energy equations has been obtained for offset-halves journal bearing through finite difference approach. The temperature of upper and lower bounding surfaces have been assumed constant throughout and have been set equal to oil inlet temperature for first iteration. A suitable initial value of attitude angle is assumed. During solution of Reynolds's equation over relaxation with relaxation factor of 1.7 has been taken for error convergence whereas under relaxation factor of 0.7 has been taken in numerical solution of energy equation for error convergence. The load carrying capacity is obtained by applying Simpson's rule to pressure distribution. In computation, wherever reverse flow arises in domain, upwind differencing has been resorted. The boundary conditions used in the solution of governing equations are same as reported in A. Chauhan et al. [2]

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IV. RESULTS AND DISCUSSION

Input parameters and properties of oil used in computer simulations are given in Table 1. From computer simulation, various performance characteristics have been obtained and are discussed as:

1. The variation of oil-film temperature and thermal pressure with offset factor (which is defined as the ratio of minimum clearance to the radial clearance) has been presented in Fig. 2 and Fig. 3 respectively. From Fig. 2, it has been observed that oil-film temperatures and thermal pressures decrease with increase in offset factor while keeping radial clearance constant which can be well explained with help of Fig. 1. However, oil-film temperatures and thermal pressures have been obtained same with variation in offset factor while keeping minimum clearance constant. Further, values of oil-film temperature and thermal pressures has been observed nearly same when offset factor is 0.4 for both the cases. Hence, in further analysis the radial clearance has been kept constant only.

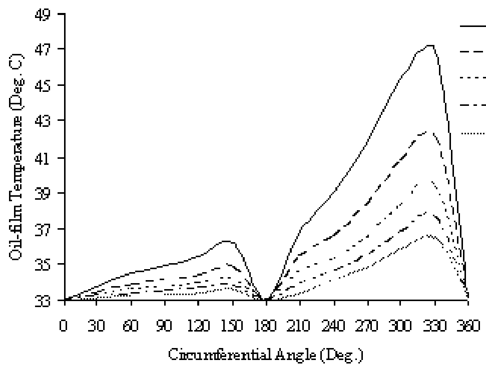

Fig. 2 Variation of oil-film temperature with circumferential angle for Mak Multigrade oil with different offset factor at speed=5000rpm for offset-halves profile bearings

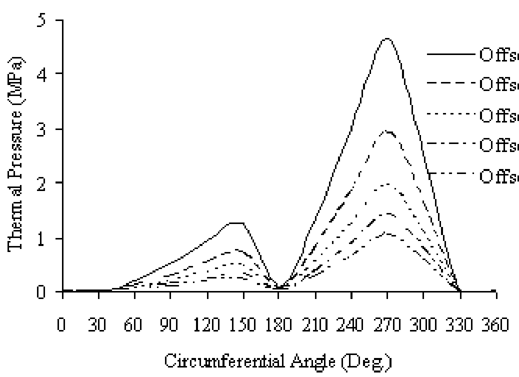

Fig. 3 Variation of thermal pressure with circumferential angle for Mak Multigrade oil with different offset factor at speed=5000rpm for offset-halves profile bearings.

2. The load carrying capacity (Fig.4) and power losses (Fig. 5) have been observed of decreasing nature with an increase in offset factor while keeping radial clearance constant. The Sommerfeld's number has been found of increasing nature with an increase in offset factor while keeping radial clearance or minimum clearance constant (Fig. 6). Further, values of load carrying capacity and power losses has been observed same when offset factor is 0.4 while keeping the radial clearance or minimum clearance constant and a slightly higher Sommerfeld's number has been observed when radial clearance is kept constant.

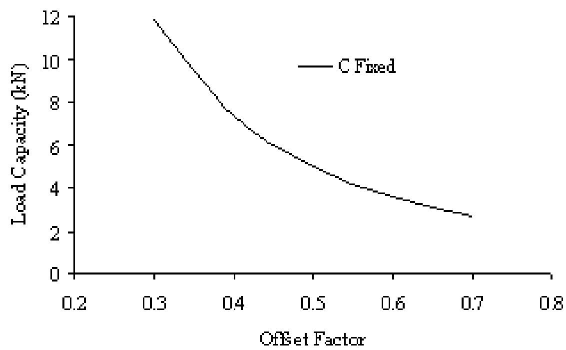

Fig. 4 Variation of load capacity with offset factor for Mak Multigrade oil at speed=5000rpm for offset-halves profile bearings

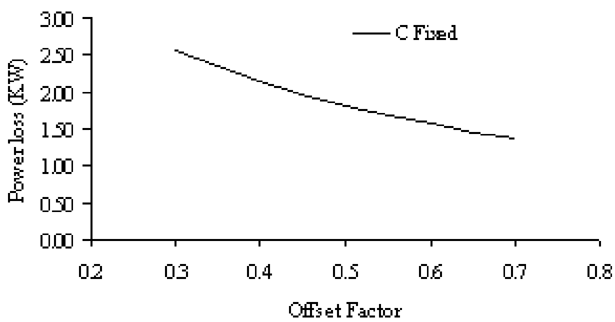

Fig. 5 Variation of Power Loss with offset factor for Mak Multigrade oil at speed=5000rpm for offset-halves profile bearings.

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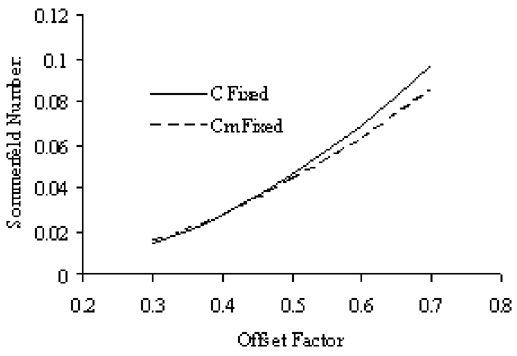

Fig. 6 Variation of Sommerfeld's Number with offset factor for Mak Multigrade oil at speed=5000rpm for offset-halves profile bearings

V. CONCLUSION

The study of the effects of variation in offset factor on various performance characteristics of offset-halves journal bearing has been carried out. From the results and discussion, the oil-film temperatures and thermal pressures are found to decrease with increase in offset factor. However, oil-film temperatures and thermal pressures have been obtained same with variation in offset factor while keeping the minimum clearance constant. Keeping the radial clearance constant, the load carrying capacity and power losses have been observed of decreasing nature with an increase in offset factor. The Sommerfeld's number has been found of increasing nature with an increase in offset factor while keeping the radial clearance or minimum clearance constant. Finally, it can be concluded that while designing offset-halves journal bearing, preferred offset factor may be kept equal to 0.4 to get desired characteristics.

Nomenclature

e Eccentricity, m

O_B Bearing centre

O_j Journal centre

O_L Lower-lobe centre

O_U Upper-lobe centre

P Film pressure, Pa

R Journal radius, mm

r Bush radius, mm

T Lubricating film temperature, ^{0}C

U Relative velocity between journal and bearing surface, m/s

u, w Velocity components in X- and Z-directions, m/s

u_L Velocity of lower bounding surface, m/s

u_U Velocity of upper bounding surface, m/s

θ Angle measured from horizontal split axis in direction of rotation

ω Angular velocity of shaft, rad/s

φ Attitude angle

ε Eccentricity Ratio

Table 1: Input parameters

Outer diameter of bearing, OD	85mm
Inner diameter of bearing, ID	65mm
Length, l	65mm
Radial Clearance, C	500μm
Minimum Clearance, C_m	200μm
Oil inlet temperature, T_0	33^{0}C
Ambient Temperature, Ta	30^{0}C
Viscosity, μ	0.200Pas
Density of oil, ρ	885Kg/m^3
Barus viscosity-pressure index, α	2.3e-8
Temperature viscosity- coefficient, γ	0.034

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