LOAD BEARING CAPACITY ANALYSIS IN SHORT AND LONG JOURNAL BEARINGS WITH PARAMETERS VISCOSITY AND HEAT EFFECT

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ABSTRACT: This research uses the Reason-Narang Rapid approach to examine the squeeze film lubrication of a finite journal bearing while taking heat effect and viscosity change into account. The generalized Reynolds equations are developed before. Load capacity expressions for squeeze film lubricated short and long journal bearings are derived. The load capacity is determined using the Reason-Narang Rapid approach, and the effects of temperature and viscosity fluctuation are numerically examined. Plots of graphs for different parameters are made.

KEY WORDS: Viscosity, thickness of the peripheral layer, Nonconformity, heat-related effect, Ability to load

1.1 INTRODUCTION:

The term "squeeze film lubrication" refers to the phenomena that occurs when two lubricated surfaces move toward one another at a regular speed. The design of the surface, the fluid's characteristics, and the load being applied all affect how long it takes to squeeze out the lubricant.

Tribologists have worked very hard in recent years to improve the effectiveness of lubricants. Because the flow characteristics of the lubricants are stabilized, it has been shown that adding modest amounts of long-chain polymer solutions to Newtonian fluids results in the best suited lubricants. The susceptibility of lubricants to shear rate charges is reduced by the application of additives. In particular, the additives can be utilized as rust inhibiters (amino phosphates), corrosion inhibiters (sulphurized olefins), and fire resistant (halogenated hydrocarbons) to improve the properties of the base oil. There are several experimental studies that make this observation. Because of the additives added to the basic oil, there is less wear and friction and more support for heavier loads.

Using the Reason-Narang Rapid approach, the generalized Reynolds equation developed in this chapter is used to examine the squeeze film lubrication of finite journal bearings while taking heat effects and viscosity change into consideration. We establish an expression for the load capacity of squeeze film lubricated short and long journal bearings. The load capacity equations are obtained using the Reason-Narang Rapid approach, and viscosity change and temperature effects are numerically examined.

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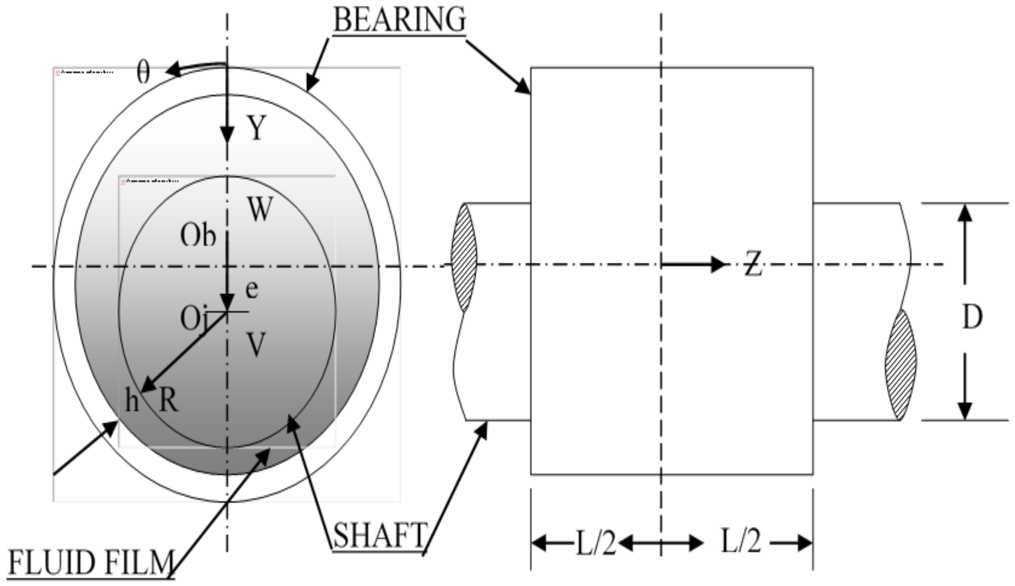

R = Radius of the Shaft. L = Length of the Bearing.

W = Load on the Shaft. V = Approach Velocity of the Shaft.

Ob = Center of the Bearing Oj= Center of the Shaft.

e = Eccentricity. h = Fluid Film Thickness.

Fig (1.1): A squeeze film journal bearing's geometry

1.2 GOVERNING EQUATION:

the lubricant flow in the journal bearing as a two-layer fluid, as depicted in fig. (1.1). The following is the equation for the fluid flow in the bearing

$$\frac{\partial}{\partial x}\left[\frac{h^3}{12\mu}F\frac{\partial p}{\partial x}\right]+\frac{\partial}{\partial z}\left[\frac{h^3}{12\mu}F\frac{\partial p}{\partial z}\right]=U\frac{dh}{dt} \qquad (1.1)$$

Here $F = \frac{(1-\frac{a}{h})^3(k-1)+1}{k}$, (1.2)

h is the film thickness is given by $h = c(1+\varepsilon\cos\theta)$ (1.3)

$$\frac{dh}{dt} = c\frac{d\varepsilon}{dt}\cos\theta \quad (1.4)$$

Where c = R-r is the clearance width and $\varepsilon = \frac{e}{c}$ is the eccentricity ratio as shown fig (1.1)

$$\frac{\partial}{\partial x}\left[\frac{h^3}{12\mu}F\frac{\partial p}{\partial x}\right] + \frac{\partial}{\partial z}\left[\frac{h^3}{12\mu}F\frac{\partial p}{\partial z}\right] = Uc\frac{d\varepsilon}{dt}\cos\theta \quad (1.5)$$

Where $F = \frac{(1-\frac{a}{h})^3(k-1)+1}{k}$

Considering the thermal effect and the viscosity, μ can be taken as

$$\mu = \mu_0(\frac{h}{h_0})^q \quad (1.6)$$

Where q is thermal factor

When (1.4) is used in place of (1.5), the updated Reynolds equation is obtained as

$$\frac{\partial}{\partial x}\left[\frac{h^3}{12\mu_0(\frac{h}{h_0})^q}F\frac{\partial p}{\partial x}\right] + \frac{\partial}{\partial z}\left[\frac{h^3}{12\mu_0(\frac{h}{h_0})^q}F\frac{\partial p}{\partial z}\right] = Uc\frac{d\varepsilon}{dt}\cos\theta \quad (1.7)$$

Substituting non-dimensional parameters

x=Rθ,dx=$Rd\theta$,z=$\bar{z}$, $\bar{h} = \frac{h}{h_0}{}^q$ (1.8)

The modified Reynolds equation may therefore be expressed in a non-dimensional manner as

$$\frac{\partial}{\partial\theta}\left[\bar{h}^3 h^{-q}\bar{F}\frac{\partial p}{\partial\theta}\right] + \frac{1}{4\lambda^2}\frac{\partial}{\partial\bar{z}}\left[\bar{h}^3 h^{-q}\bar{F}\frac{\partial p}{\partial\bar{z}}\right] = 12\frac{\mu_0 R^2 U}{c^2}\frac{d\varepsilon}{dt}\cos\theta \quad (1.9)$$

The non-dimensionless pressure is given by

$$\bar{p} = \frac{pc^2}{U\mu_0 R^2 \frac{d\varepsilon}{dt}} \tag{1.10}$$

By substituting (1.10) in (1.9), it becomes

$$\frac{\partial}{\partial\theta}\left[\bar{h}^{(3-q)}\bar{F}\frac{\partial\bar{p}}{\partial\theta}\right] + \frac{1}{4\lambda^2}\frac{\partial}{\partial\bar{z}}\left[\bar{h}^{(3-q)}\bar{F}\frac{\partial\bar{p}}{\partial\bar{z}}\right] = 12\cos\theta \tag{1.11}$$

Where $\bar{F} = \frac{(1-\frac{a}{h})^3(k-1)+1}{k}$

The boundary conditions for the equation (1.10) are

$$\bar{p} = 0, at\ \theta = \frac{\pi}{2}, \frac{3\pi}{2} \quad , \bar{p} = 0\ at\ \bar{z} = \pm\frac{1}{2}, \frac{\partial\bar{p}}{\partial\bar{z}} = 0\ at\ \bar{z} = 0 \tag{1.12}$$

Where 'θ' the circumferential angle, z is bearing axis parallel to the shaft axis

SHORT BEARING ANALYSIS:

If λ≤ 0.5 it is called short bearing or narrow bearing. Neglecting pressure variation in 'x' Direction, the modified Reynolds equation reduces to

$$\frac{\partial}{\partial\bar{z}}\left[\bar{h}^{(3-q)}\bar{F}\frac{\partial\bar{p}}{\partial\bar{z}}\right] = 48\lambda^2\cos\theta \tag{1.13}$$

Integrating the above equation (4.13) twice, the equation becomes

$$\bar{p} = \frac{48\lambda^2\cos\theta\bar{z}^2}{2\bar{h}^{(3-q)}\bar{F}} + \frac{c_1}{\bar{F}\bar{h}^{(3-q)}}\bar{z} + c_2 \tag{1.14}$$

Where c_1 and c_2 are constants of integration and evaluated using boundary conditions given in equation (1.12), we get

$$c_1 = 0, c_2 = \frac{48\lambda^2\cos\theta}{\bar{h}^{(3-q)}\bar{F}}\left(\frac{1}{8}\right) \tag{1.15}$$

Substituting equation (1.15) in (1.14), the pressure for a short bearing becomes

$$\bar{p} = \frac{24\lambda^2 \cos\theta}{\bar{h}^{(3-q)}\bar{F}}\left(\bar{z}^2 + \frac{1}{4}\right) \quad (1.16)$$

Pressure at the Centre line of the bearing is $\bar{z}=0$ then

$$\bar{p} = \frac{6\lambda^2 \cos\theta}{\bar{h}^{(3-q)}\bar{F}} \quad (1.17)$$

Therefore the non-dimensional pressure for short bearing is

$$\bar{p}_s = \frac{6\lambda^2 \cos\theta}{\bar{h}^{(3-q)}\bar{F}} \quad (1.18)$$

$$W_s = 2\int_{\frac{\pi}{2}}^{\frac{3\pi}{2}} \cdot \int_{0}^{\frac{L}{2}} PR\,\mathrm{Cos}\,\theta\, dz\, d\theta \quad (1.19)$$

$$W_s = 2\frac{\mu_0 UR^3 \frac{d\varepsilon}{dt} L}{c^2}\int_{\frac{\pi}{2}}^{\frac{3\pi}{2}} \frac{6\lambda^2 Cos^2\theta}{\bar{h}^{(3-q)}\bar{F}}\, d\theta \quad (1.20)$$

And the dimensionless load carrying capacity is given by

$$\bar{W}_s = \frac{W_s c^2}{\mu_0 UR^3 \frac{d\varepsilon}{dt} L} = 6\lambda^2 \int_{\frac{\pi}{2}}^{\frac{3\pi}{2}} \frac{Cos^2\theta}{\bar{h}^{(3-q)}\bar{F}}\, d\theta \quad (1.21)$$

Where $\bar{F} = \frac{(1-\frac{a}{h})^3(k-1)+1}{k}$

LONG BEARING ANALYSIS:

If λ > 2, it is called long bearing that is the bearing is infinitely long in axial direction

and pressure is constant in that direction. Thus neglecting the $\frac{\partial p}{\partial z}$, the modified Reynolds equation reduces to

$$\frac{\partial}{\partial\theta}\left[\bar{h}^{(3-q)}\bar{F}\frac{\partial\bar{p}}{\partial\theta}\right]=12\cos\theta \tag{1.22}$$

Integrating (1.22) with respect to θ, We get

$$\bar{h}^{(3-q)}\bar{F}\frac{\partial\bar{p}}{\partial\theta}=12\sin\theta+B \tag{1.23}$$

Where B is the integral constant

The boundary condition is $\frac{\partial\bar{p}}{\partial\theta}=0\ at\ \theta=\pi$ (1.24)

Applying the boundary condition in (1.23)

Then constant B=0 we get

$$\frac{\partial\bar{p}}{\partial\theta}=\frac{12\sin\theta}{\bar{h}^{(3-q)}\bar{F}} \tag{1.25}$$

Again integrating (1.25) and applying the boundary conditions

$$,\bar{p}=0\ \text{at}\ \bar{z}=\mp\frac{1}{2}\ ,\frac{\partial\bar{p}}{\partial\bar{z}}=0\ at\ \bar{z}=0 \tag{1.26}$$

Then we get dimensionless pressure as

$$\bar{p_l}=12\int_0^{\frac{\pi}{2}}\frac{\sin\theta}{\bar{h}^{(3-q)}\bar{F}}d\theta \tag{1.27}$$

Load capacity for the long bearing is

$$W_l=2L\int_0^{\pi}PR\sin\theta d\theta \tag{1.28}$$

$$W_l=\frac{\mu_0 UR^3\frac{d\varepsilon}{dt}L}{c^2}\int_0^{\pi}\frac{24\sin^2\theta}{\bar{h}^{(3-q)}\bar{F}}d\theta \tag{1.29}$$

And the dimensionless load capacity is

$$\overline{W}_l = \frac{W_l c^2}{\mu_0 U R^3 \frac{d\varepsilon}{dt} L} = 24 \int_0^{\pi} \frac{\sin^2\theta}{\overline{h}\,(3-q)\,\overline{F}} d\theta \qquad (1.30)$$

Where $F = \frac{(1-\frac{a}{h})^3 (k-1) + 1}{k}$

FINITE BEARING ANALYSIS:

For finite bearings, the two dimensional Reynolds equation is solved using Rapid-Narang technique. If p, $p_s$andp$_l$ are the pressure in finite, short and long bearings respectively, then the relationship between them is given by

$$\frac{1}{p} = \frac{1}{p_s} + \frac{1}{p_l} \qquad (1.31)$$

The finite bearing pressure is

$$p = \frac{p_s p_l}{p_s + p_l} \qquad (4.32)$$

LOAD CARRYING CAPACITY:

As the load is proportional to the pressure, the load carrying capacity for the finite bearing is *Wf*in relation with W_s andW_l

$$\frac{1}{W_f} = \frac{1}{W_s} + \frac{1}{W_l} \qquad (1.33)$$

$$W_f = \frac{W_s W_l}{W_s + W_l} \qquad (1.34)$$

By substituting the short and long bearing load equations (1.21) and (1.30) in the above equation (1.34) the finite bearing load carrying capacity in non-dimensional form is

$$\overline{W} = \frac{W_f c^2}{\mu U R^3 \frac{d\varepsilon}{dt} L} = \frac{W_s W_l}{W_s + W_l} \tag{1.35}$$

Equation (1.35) is solved numerically and graphs have been plotted for different values of various parameters.

1.3 RESULTS AND DISCUSSION:

For various values of "k," the load bearing capacity is displayed with "a" in Fig. (1.2). It has been noted that for k>1 the load capacity drops and rises with increasing 'a'. In the case of a single layer with k=1, the load capacity is unaffected as 'a' rises.

For various values of the eccentricity ratio "epc," the load bearing capacity is displayed with letter "a" in figure (1.3). It has been shown that for 'k'>1, the load capacity improves with rising values of 'epc' and with increasing values of 'a'.

The load bearing capacity for various values of 'a' is displayed with 'q' in fig. (1.4). It has been noted that at 'k'>1, the load capacity increases for rising values of 'a' and decreases for increasing values of 'q'.

In fig. (1.5), the load bearing capacity is shown with 'q' for various values of 'k'. The load capacity is seen to decrease for rising values of q' and rise for increasing values of 'k'.

For various levels of eccentricity, or "epc," the load bearing capacity is shown with the letter "q" in Fig. 1. It has been found that for 'k'>1, the load capacity increases for rising values of 'epc' and decreases for increasing values of 'q'.

1.4 GRAPHS:

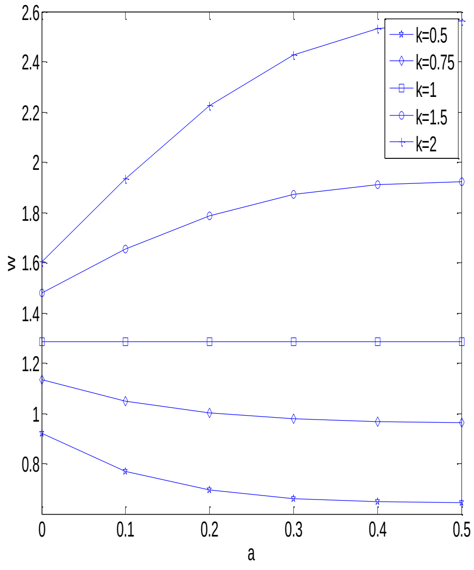

Fig (4.2): The dimensionless load with peripheral thickness 'a'

for different values of 'k' at q=0, epc=0.4, λ=0.2

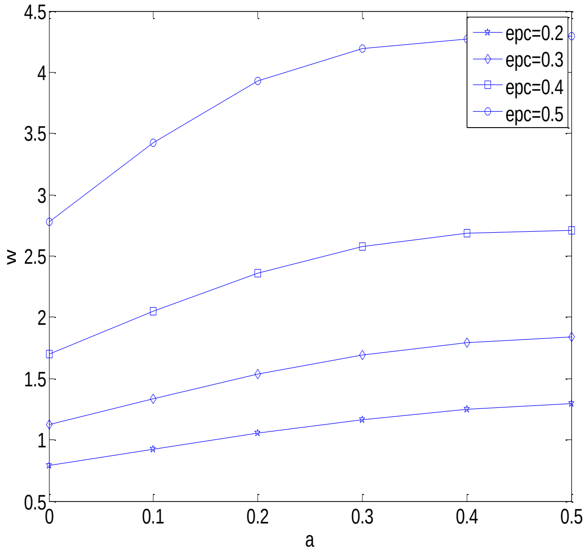

Fig (4.3): The dimensionless load with peripheral thickness 'a' for different

values of 'epc'at **k=2,q=0.1,λ=0.2**

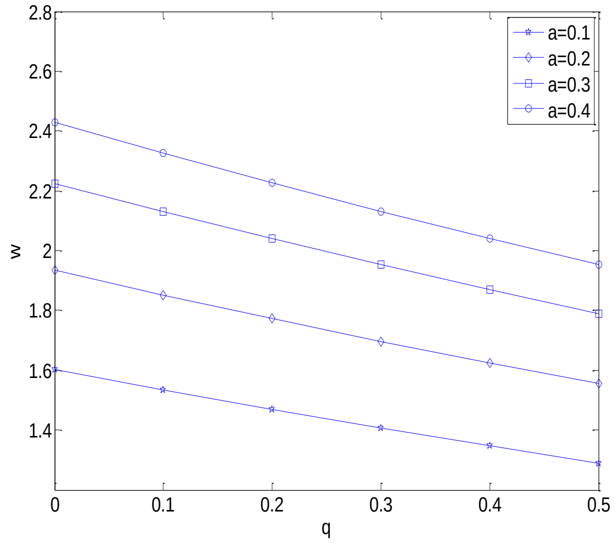

Fig (4.4): The dimensionless load with thermal effect 'q' for different values of 'a'at k=2,epc=0.4,λ=0.2

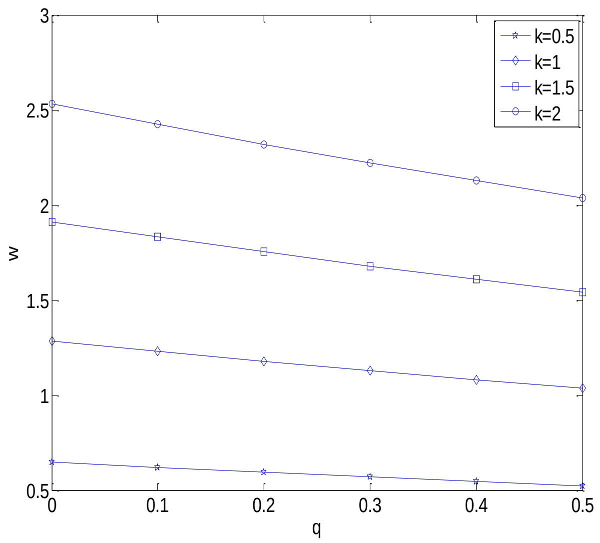

Fig (4.5): The dimensionless load with the thermal effect 'q' for different values of 'k' at a=0.5,epc=0.4,λ=0.2

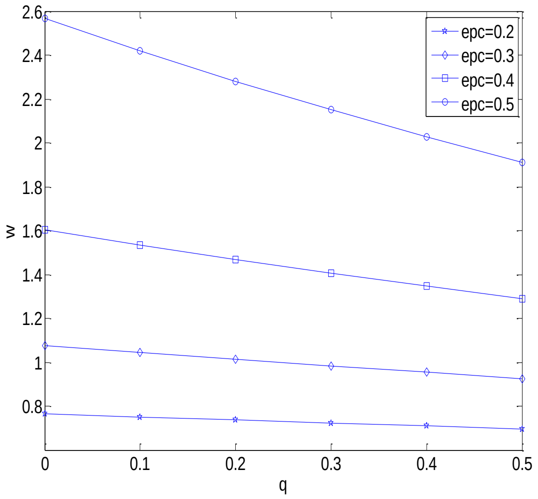

Fig (4.6): The dimensionless load with the heat effect 'q' for different values of 'epc' at k=2,a=0.1,λ=0.2

1.5. CONCLUSIONS:

- Using the Reason-Narang Rapid approach, this work studies the squeeze film lubrication of finite journal bearings while taking heat effects and viscosity fluctuation into consideration.
- A squeeze film lubrication expression for load capacity in short and long journal bearings is derived.
- It has been demonstrated that a high viscous layer at the perimeter causes an increase in load capacity whereas a low viscous layer causes a drop.
- For a variety of parameters, it is demonstrated that the load capacity falls down as the heat factor rises.
- It is determined that the presence of lubricating additives causes the particles to adhere to surfaces, produce a highly viscous layer close to the perimeter, and improve the load capacity.

NOMEMCLATURE

a	The mean height of surface asperities in the symmetric roughness case
h	Film thickness

c	Radial clearance
e	Eccentricity
ε	Eccentricity ratio
μ	Viscosity of the base lubricant
p	Hydrodynamic pressure
W	Load carrying capacity
R	Radius of the shaft
r	Radius of the bearing
k	Ratio of the viscosities near the surface to the purely hydrodynamic zone
U	Velocity of the surfaces in the case of one-dimensional form
x,*y*,*z*	Cartesian coordinates
μ	Viscosity of the base lubricant
q	Thermal factor

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