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The effect of couple stress and surface roughness on journal bearing using cavitation boundary conditions with respective load

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ABSTRACT

Theoretical field of study of combined effects of surface roughness and couple stress on journal bearing considering cavitations condition is analyzed. The modified Reynolds equations accounting for the couple stress and surface roughness, using deterministic theory. It is applied to study the effect of couple stress; Expressions for Load capacity are studied by evaluating them numerically, for various parameters. From the numerical computation of the result ,it is observed that the effect of cavitations point is increase the load carrying capacity decreases. The cavation point is inversely proportional to the load; A parameter characterizing different type of roughness is identified. It is also observed that load capacity diminishes, and cavation point for longitudinal increase for roughness pattern.

Keyword: Cavitations, couple stress, surface roughness, journal bearing, load.

INTRODUCTION

In general additives are added to the base lubricant to improve the bearing characterstics.Various theories have been proposed for this. These additives are generally long-chain organic compounds and they may form a high viscous layer near the surface .In most of the theoretical investigation of hydrodynamic lubrication, it is has been assumed that the bearing surface are smooth .This is an unrealistic assumption for the bearing operating with small film thickness. In the recent years, a considerable amount of tribology research has been devoted to the study of surface roughness on hydrodynamic lubrication, this is mainly because of the fact that all solid surfaces are rough to some extent and generally the height of roughness asperities is of the same order as the mean separation between lubricated contacts. Hence, the effect of surface roughness plays a significant role in the development of the science and technology of lubrication. In the deterministic approach, the effect of surface roughness is taken in to account in the usual Reynolds equation by considering that the film thickness is a function of surface roughness which may represent by a series of sine and cosine waves.(Dowson,1971) has applied this procedure to study the bearing characteristics of rollers; spiral groove bearing etc.(shukla 1978) gave deterministic theory to study the effects of surface roughness when the mean height of the asperities is of the same order of magnitude as the minimum film thickness.Interation between lubrication and technology has become important owing to increase demand for high speed highly loaded machines. Various materials in the form of solid particles and long-chain polymers are added to oils to enhance their lubricating properties. The prominent feature of such a lubricant is an increased effective viscosity, especially in thin films. Rheological abnormalities are mainly observed in lubricants with long-chain molecules, lubricants with additives and contaminated lubricants. Various theories have been postulated to describe rheological abnormalities. The Stokes micro continuum model which based on the classical conception of hydrodynamics is the simplest form and allows for inspection of polar effects, such as the existence of couple stresses, body couples and non symmetric tensor is adopted widely (stokes 1966) .The cavitations occurs in the lubrication of partial bearing which is considered by imposing Reynolds Boundary Conditions at the trailing end of the pressure curve.(Sinha et al.1981) studied the couple stress in journal bearing lubricant; (prawal Sinha and Chandan Singh 1981) analyzed the effect of couple stress in the lubrication of rolling contact bearings considering cavitation.(Ariticle et al.1985) improved the analysis of starved journal bearing including temperature and cavitation.(Vaidyanathan and Keith 1989) investigated the Numerical prediction of cavitation in noncircular bearings. (Mokhiammer et al.1999) studied the journal bearing lubricated by fluids with couple stress considering the elasticity of the linear. On the performance of dynamically loaded journal bearing lubricated with couple stress fluids is studied by (Xiao-L Wang et al.1997). (jaw-Ren Lin 2002) obtained the effects couple stresses the lubrication of finite journal bearing .Transverse roughness in short journal bearing under dynamic loading is presented by (Raj and Sinha 1983). (Raghavendra 2007) improved on roller bearing under lightly loaded conditions. (T.V.V.L.N.Rao 2005) has studied the stability analysis of a rough journal bearing considering cavitation effects. In this study the effect of couple stress and surface roughness on journal bearing using cavitation boundary conditions with respective load has been discussed.

MATHEMATICAL ANALYSIS AND METHOD OF SOLUTION

The problem is considered is that of the steady laminar flow of an incompressible fluid between two eccentric cylinders in uniform elative motion. The gap or clearance, (the difference between the two radii) between the two cylindrical surface is small compared with the radius of inner cylinder (journal).Below fig.1 shows a journal bearing operating with a constant external load ,W and speed U under the physical conditions imposed the journal operates at an eccentricity □.

The cavitations occurs in the lubrication of partial bearing which is considered by imposing Reynolds Boundary Conditions at the trailing end of the load curve.i.e., it is assumed that the load is positive and terminates to zero with a zero load gradient. Thus it is assumed that the film is continuous only in the region of positive pressure and that it capitates at some position $\theta = \theta^{*} > \pi, where\ L = \frac{dl}{d\theta} = 0$, forming a discontinuous mixture of air, vapors and lubricant in the cavitated region $(\theta^{*} < \theta < 2\pi)$. these conditions have been applied externally in journal bearing analysis (Sinha et al.1981)

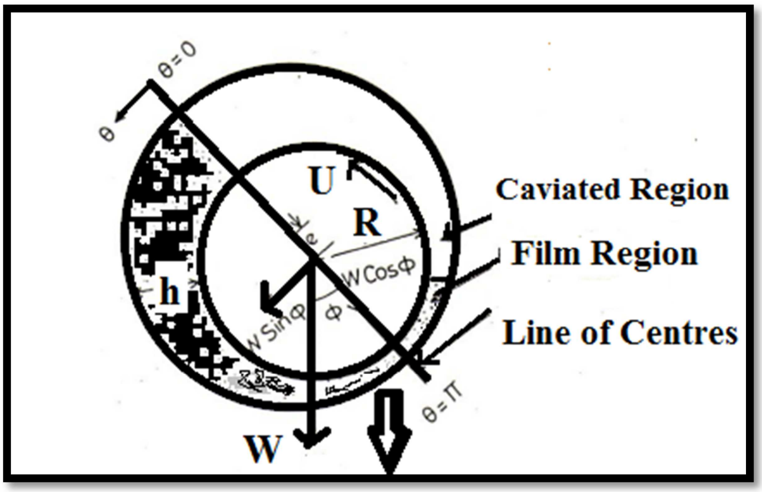

Fig.1 Schematic view of journal bearing operating with a constant external load.

The generalized Reynolds equation which accounts both surface roughness and couple stress property of the lubricant is given by (Ragavendra 2007).

If the bearing is assumed to be infinitely long in the axial direction, the one-dimensional Reynolds equation reduces to

$\frac{d}{dx}\left(\frac{hn^3F_2}{12\mu}\frac{dl}{dx}\right) = U\frac{d}{dx}(h_n + 2h_s)$………………….. (1)

$$x = R\theta, dx = Rd\theta$$

$\frac{d}{dx}\left(\frac{hn^3F_2}{12\mu}\frac{dl}{dx}\right) = U\frac{d}{dx}(1+\beta) \ldots\ldots\ldots\ldots..where\ \beta = \frac{2h_s}{h_n}$……………. (2)

Integrating equation (3)

$$\frac{d}{d\theta}\left(\frac{hn^3F_2}{12\mu}\frac{dl}{dx}\right) = UR(1+\beta)$$

$$\frac{dl}{d\theta} = \frac{12UR\mu(h_n + 2h_s)}{F_2hn^3} \ldots\ldots\ldots\ldots\ldots\ldots.(3)$$

$$\int\frac{dl}{d\theta} = \frac{12UR\mu}{F}\int_0^{\theta}\frac{h_n - {h_n}^*}{{h_n}^3}d\theta$$

$h_n = H, {h_n}^* = \overline{H},\quad F = \overline{F},\ \theta = \theta^*\ \ \overline{L} = \frac{L}{12U\mu R}$

$\overline{L} = \int_0^{\theta}\frac{H-\overline{H}}{H^3\overline{F}}d\theta \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(\ldots 4)$

$$\overline{L}(\theta^*) = 0\ in\ eq\ (4)$$

$0=\int_0^{\theta}\frac{H-\overline{H}}{H^3\overline{F}}d\theta$ Load Component

$$W_{\frac{\pi}{2}} = Wsin\varphi = \int_0^{\theta^*}Lsin\theta R\ d\theta$$

$$W_{\frac{\pi}{2}} = Wsin\varphi = \int_0^{\theta^*}\frac{h_n - {h_n}^*}{{h_n}^3F}12\mu URsin\theta R\ d\theta$$

$W_{\frac{\pi}{2}} = Wsin\varphi = 12\mu UR^2\int_0^{\theta^*}\frac{h_n-{h_n}^*}{{h_n}^3F}Cos\theta\ d\theta$

$\frac{W\pi/2}{12\mu UR^2}=\int_0^{\theta^*}\frac{h_n-{h_n}^*}{{h_n}^3F}Cos\theta\ d\theta$

$$\overline{W_{\frac{\pi}{2}}} = \int_0^{\theta^*}\frac{H-\overline{H}}{H^3\overline{F}}Cos\theta d\theta \ldots\ldots\ldots\ldots..(5)$$

$$W_0 = Wcos\emptyset = -\int_0^{\theta^*}Lcos\theta Rd\theta$$

$$W_0 = Wcos\emptyset = -12\mu UR^2\int_0^{\theta^*}\frac{h_n - {h_n}^*}{{h_n}^3F}cos\theta d\theta$$

$$W_0 = 12\mu UR^2\int_0^{\theta^*}\frac{h_n - {h_n}^*}{{h_n}^3F}sin\theta d\theta$$

$$\frac{W0}{12\mu UR^2} = \int_0^{\theta^*} \frac{H-\bar{H}}{H^3\bar{F}} sin\theta d\theta \ldots\ldots\ldots\ldots (6)$$

$$\overline{W_0} = \int_0^{\theta^*} \frac{H-\bar{H}}{H^3\bar{F}} sin\theta d\theta \ldots\ldots\ldots\ldots (D)$$

From Equation (B) and (D) we get
$W^* = (\overline{W_0}+\overline{W_{\frac{\pi}{2}}})^{1/2}$

$$W^* = \int_0^{\theta^*} \frac{H-\bar{H}}{H^3\bar{F}} sin\theta d\theta + \int_0^{\theta^*} \frac{H-\bar{H}}{H^3\bar{F}} Cos\theta d\theta$$

$W^* = \left[\frac{H-\bar{H}}{H^3\bar{F}}\right]^{\frac{1}{2}} \left\{\int_0^{\theta^*} sin\theta d\theta + \int_0^{\theta^*} cos\theta d\theta\right\}^{1/2}$ After simplification we get the result is where K=$\left[\frac{H-\bar{H}}{H^3\bar{F}}\right]^{\frac{1}{2}}$ and $\varepsilon = 2$
W^* =K$(2cos\theta^* + 1)^{1/2} \ldots\ldots\ldots\ldots\ldots (7)$

RESULTS AND DISCUSSION

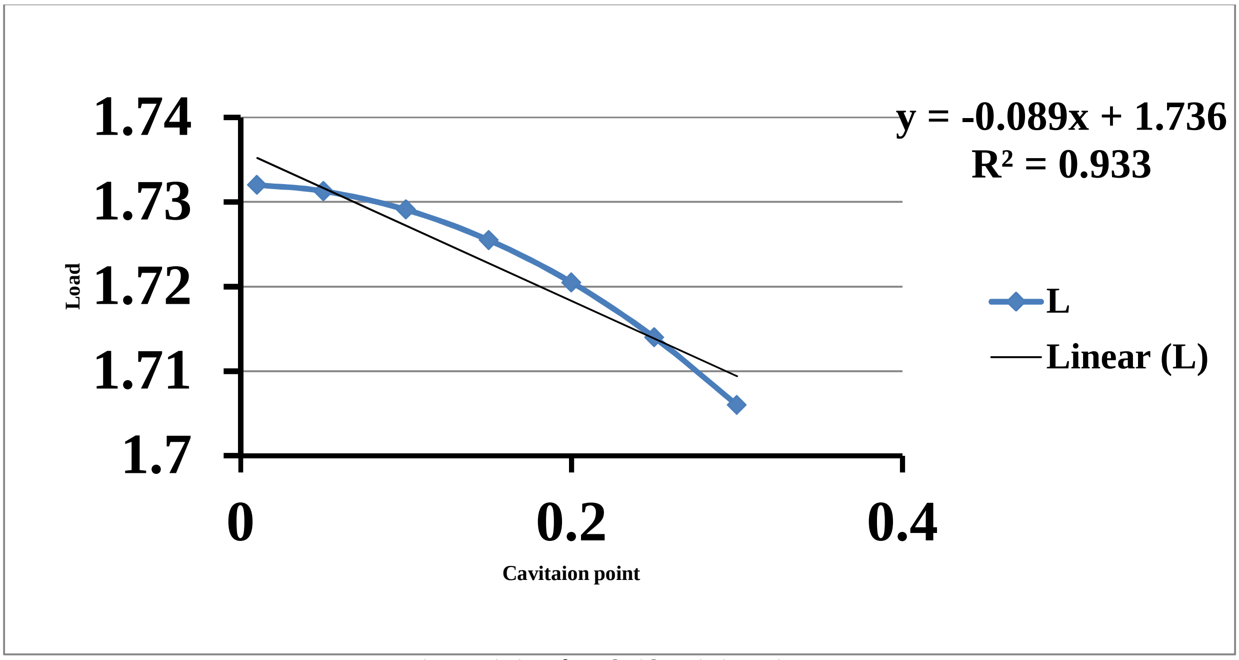

Fig.2 Variation of Load with cavitaion point.

The Observation of couple stress owing to the lubricating substance blending with some long chain additives and surface roughness on the performance characteristic of bearing system are evaluated on the idea of Stoke's theorem the couple stress impact is specified by the couple stress parameter L*.The dimensionless load carrying capacity L* with cavation point θ* at eccentricity ration ∈ the massive price of cavation point is that the film load obtained ,thus the couple stress impact lead to an improvement in load (W*) particularly at the cavation point is decreases then the load capacity increase, then the pressure is improve (shown in Fig.2). Mean whereas the roughness effects boosts the load capability over that of sleek matter

Table.1 Analytical Value of load and cavitaion point.

θ^*	L
0.01	1.73202
0.05	1.7313
0.1	1.7291
0.15	1.7255
0.2	1.7205
0.25	1.7140
0.3	1.7060

CONCLUSION

In this paper the generalized Sir Joshua Reynolds equation obtained for additive and surface roughness is applied to review the consequences of surface roughness and couple stress in bearing take into account cavation expression of load capability and purpose of cavation are derived and numerically estimated .The impact of couple stress ,eccentricity magnitude relation and mean height of roughness asperities on cavation is calculable .It is found that the impact of couple stress is to decrease that time of cavation and it decrease more because the couple stress parameter increase .The impact of surface roughness is to decrease cavation purpose for transversial roughness and increase for longitudinal roughness. The impact of couple stress parameter ,roughness parameter is mathematically derived on the load capability.

REFERENCES

[1]. Artiles A ,Heshmat H, ASME *Journal of LubricationTechnology*, **1985**, 1-13.
[2].Dowson D, Whomes T.L , *Wear* ,**1971,**18, pp. 129.
[3].Jaw-Ren Lin, *Wear*, **1997**,206, pp. 171-178.
[4].Mokhiammer V.M, Crosby W.A, El-Hamal W.A , *Wear* ,**1999,** 224, 194-201,
[5].Prawal Sinha and Chandan Singh , *Wear*, **1981,**67, 85-98,
[6].Raghavendra Rao R and Rajasekhar K , **2007**, *Indian Journal of Tribology2*(1), 11 – 25.
[7].Raj A ,Sinha P , *Tribology International* , ***1983***,16(5), Oct, 245.
[8].S.Rushma, V.Diwakar Reddy, G.Jayachandra Reddy,and Ramakrishna Prasad, *Int.J. of Appl.Math and Mech.* **2011**,7 -21/0: 85-98.
[9].Shukla JB, *Wear* **1978**, 49, 33.
[10].Sinha P, Singh C and Prasad KR ,*Wear*,**1981,** 67, 85.
[11].Vaidyanathan K, Keith T.G , STLE *Tribology Tran.* **1989,** 32(2), 215-224.
[12].Xiao-Li Wang, Ke-Qin Zhn and Shi-Zhu Wen ,*Tribology International*, **2002**, 35, 185-191.