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EFFECT OF TEMPERATURE ON THE TRIBOLOGICAL BEHAVIOUR OF COPPER INFILTRATED POROUS JOURNAL BEARING USING NON NEWTONIAN FLUID

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ABSTRACT

In nature, the heat transfer in porous media is more complicated than that in the continuous because two media of widely different properties are conducting the heat simultaneously. The lubricant used with bearing system is usually non-Newtonian such as powdered graphite and the carrier fluid in the ethylene glycol. Considerable research studies have been carried out to explain the flow field in bearing system. Different methods such as experimental mathematical analysis or numerical simulation can be employed for studying the process fluid flow. The effects of isothermal shear theory and visco-elasticity needs to be considered. During the running condition viscous heat generated, that because the temperature rise in the system, which in turn may cause the fluid to behave in the non Newtonian manner. An attempt has been made to correlate the temperature function of the developed copper infiltrated bearing under running condition with the non-newtonicity of the lubricant, assuming suitable boundary condition with proposed methodology.

Keywords: Copper infiltration, journal bearing, non-newtonicity, power law fluid.

INTRODUCTION

Porous metal bearings can be used where other plain metal bearings are impractical due to lack of space or inaccessible of lubrication. These are also considerably cheaper than their equivalent externally lubricated plain bearings (Abdallaha and Guedouar, 2001). Impregnated oil bearing are widely used in industrial applications, in many cases they are more advantageous than non porous bearing because they do not need continuous lubrication therefore their structure is simple and they also reduce costs (Baka, 2002). The lubricant used with bearing system is usually non-Newtonian such as powdered graphite and the carrier fluid in the ethylene glycol. Most modern automotive lubricants contain polymeric additives that make lubricants slightly shear thinning and mildly viscoelastic (Zhang, 2002). In the case of hydrodynamic solution of the flow system in the bearing requires extensive computational effort, since the governing equations of flow are coupled due to the different species that exist in the system (Pakdemirli *et al*., 2004). However the assumptions of the uniform fluid with non Newtonian behaviour (Kumar, 1985) enable the solution of flow equation analytically. Various models are developed to account for the non Newtonian behaviour of the fluid flow. Numerical results of the modified Reynolds's equation show that non Newtonian lubricants load capacity is not always higher or lower than Newtonian lubricant and non Newtonian lubricant has flatter pressure profile at higher speed (Chen and Chen, 2005). The effect of isothermal shear theory and visco elasticity needs to be considered (Rastogi and Gupta, 1991). During the running condition viscous heat generated, that because the temperature rise in the system, which in turn may cause the fluid to behave in the non Newtonian manner. As reviewed by Raghunandan, (Raghunandan and Majumdar, 2001), many studies have been performed to evaluate the effect that non Newtonian lubricant may have on the performance of bearings. Non Newtonian temperature and pressure effect of graphite powder lubricant when added to a Newtonian carrier fluid and applied in a rotating hydrostatic step bearing was studied by Peterson *et al*. (1994). They showed that temperature increased with bearing rotational speed and compared favourably with the mathematical prediction. Heat transfer in porous media is more complicated as two different heat transport dynamics exist (Kheng *et al*., 1999) in the porous media as shown in figure 1. Consequently that indicates that the solid and liquid matrix in porous media no longer in thermal equilibrium. More over viscous heat generation is significant; heat transfer between the lubricant and the surroundings is also taken to be considered (Kozma, 1995). Then the property of the lubricant can be specified by non linear differential viscosity. This paper is about the calculation of hydrodynamic load carrying capacity of porous Journal bearings. Pressure functions were determined and compared to each other to show the differences of several simplifications, assumptions and boundary conditions (Baka, 2002). The porous material was assumed to be isotropic and homogeneous. Four

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pressure frictions were analyzed using the short bearing approximation. The load carrying capacity and the coefficient of friction were calculated and compared to one another. In this present paper an attempt has been made to describe the non newtonicity parameter with the change of working temperature both mathematically and experimentally.

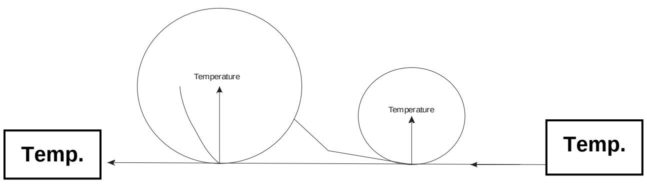

Fig. 1. Shows the temperature profile of the solid liquid matrix in a porous media.

THEORITICAL ANALYSIS

The hydrodynamic theory of lubrication of porous bearing originated with Cameron (1974) who obtained a solution for oil film pressure and load carrying capacity of finite, full bearing using short bearing assumption. He assumed that $\frac{\partial^2 p}{\partial y^2}$ = constant, and the same pressure is in the oil film and also in the porous material, Using these assumptions many solutions were achieved during the last century for static and also dynamic operating conditions, he wrote the Reynolds equation of porous bearings in the following form:

$$\frac{\partial}{\partial x}\left(h^3 \frac{\partial p}{\partial x}\right)+\frac{\partial}{\partial z}\left(h^3 \frac{\partial p}{\partial z}\right)=\left(6\eta U \frac{\partial h}{\partial x}+12\frac{\phi}{\eta}\frac{\partial p^*}{\partial y}\bigg|_{y=0}\right), \tag{1}$$

Where V = $\frac{\partial p^*}{\partial y}\frac{\Phi}{\eta}$ is the Darcy rule, which expresses the flow of lubricants into the porous materials. Using the clearance function h (θ) = C (1 + cosθ) of a Journal bearing and introducing the permeability parameter $\phi = \frac{HK_y}{C^3}$, he solved the Reynolds equation with the Somerfield boundary condition and got the following pressure function:

$$p(z)=\frac{3U\eta\varepsilon\sin\theta}{rC^3\left[(1+\varepsilon\cos\theta)^3+12\phi\right]}\left(\frac{L^2}{4}-z^2\right). \tag{2}$$

(p is marked the only function of z. This is only an agreement because p(z) was obtained from the Narrow Bearing Condition.)

Cameron (1974) also determined the friction force from the shear stresses in the lubricant:

$$F_s = \iint_A \tau dA, \tag{3}$$

and got the following formula to calculate the coefficient of friction:

$$f=\frac{C}{r}\left[\frac{4\pi^2 r^3\eta LN}{WC^2\sqrt{1-\varepsilon^2}}+\frac{\varepsilon}{2}\sin\beta\right]. \tag{4}$$

Roulea (1963) modified Morgan and Cameron (1957) conditions and assumed that the pressure in oil film and in porous material is different. He thought that Δ^2 p* (x, y, z) = 0, where p* is the pressure in the porous material. He solved the Reynolds equation using infinite series and got a pressure distribution in this form:

$$p=\frac{24\eta UL^2}{\pi^3 rC^2}\sum_{n=1}^{\infty}\frac{(-1)^{N+1}\varepsilon\sin\theta\cos\pi\beta_n\frac{z}{L}}{\beta_n^3\left[(1+\varepsilon\cos\theta)^3+12\phi\frac{L}{H\pi\beta_n}\tanh\pi\beta_n\frac{H}{L}\right]}, \tag{5}$$

Where β_n = (2N - 1).

Prakash and Vij (1974) also obtained a pressure function for short bearings with slip condition in a form of infinite series. They did not use such simplifications, but solved the two governing equations (Reynolds, Laplace) separately (Baka, 2002). The form of the pressure function in the porous bush p* and in the oil film (p) was similar:

$$p(\theta.z)=2\sum_{n=1}^{\infty}c_n\cos\lambda_n\cosh\left[\lambda_n H\right] \tag{6}$$

and

$$p^*(\theta.y.z)=2\sum_{n=1}^{\infty}c_n\cos\lambda_n\cosh\left[\lambda_n(y+H)\right] \tag{7}$$

Solving the governing equation the following expression can be got for pressure distribution in oil

$$p(z)=\frac{24\eta UL^2}{\pi^3 rC^2}\sum_{n=1}^{\infty}\varepsilon\frac{(-1)^{N+1}g(\theta)\cos\pi\beta_n\frac{z}{L}}{(2N-1)^3}, \tag{8}$$

And for the coefficient of friction

$$f=\frac{c}{r}\left[\frac{4\pi^2 r^3 LN\eta}{WC^{2\sqrt{(1+S)^2-\varepsilon^2}}}+\frac{\varepsilon}{2}\sin\beta+\frac{48n\eta L^3 r\left(1-2\alpha^2\right)S^2\varepsilon^2}{\pi^3 WC^2}\right] \tag{9}$$

$$\left[\sum_{i=1}^{5}\frac{1}{(2i-1)^4}\int_0^{\pi}\frac{\sin\theta g_{i(\theta)}}{(1+s+\varepsilon\cos\theta)^2}d\theta\right]$$

The above presented three pressure functions used the short bearing boundary condition. The newly developed copper infiltrated porous journal bearing under consideration assumed to be governed by the following equation under hydrodynamic lubrication using non Newtonian fluid. Figure 2 shows the schematic diagram of a hydrodynamic cu-infiltrated journal bearing under steady state operating condition.

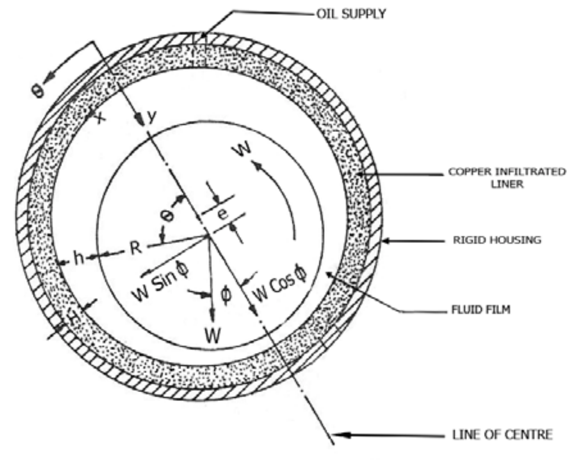

Fig. 2. Schematic diagram of a hydrodynamic cu-infiltrated journal bearing under steady state operating condition.

The Reynolds's Equation for the copper infiltrated bearing under consideration using power law fluids Kumar and Chandra (1981) can be written as follows:

$$\frac{n}{n+1} \quad \frac{1}{2(2n+1)}\left[\frac{\partial}{\partial x}\left(\frac{h^{2n+1}}{\mu}\frac{\partial p}{\partial x}\right)^{1/n}+\frac{\partial}{\partial z}\left(\frac{h^{2n+1}}{\mu}\frac{\partial p}{\partial z}\right)^{1/n}\right] = \frac{U}{2}\frac{\partial h}{\partial x}+\frac{k_y}{\mu}\frac{\partial p^*}{\partial y}\Big|_{y=0} \qquad (10)$$

Where n=Non-New tonicity of the lubricant.

Now applying the short bearing theory ($\frac{\partial p}{\partial z} >> \frac{\partial p}{\partial x}$), the equation reduced to

$$\frac{n}{n+1} \quad \frac{1}{2(2n+1)}\left[\frac{\partial}{\partial z}\left(\frac{h^{2n+1}}{\mu}\frac{\partial p}{\partial z}\right)^{1/n}\right] = \frac{U}{2}\frac{\partial h}{\partial x}+\frac{k_y}{\mu}\frac{\partial p^*}{\partial y}\Big|_{y=0} \qquad (11)$$

On subsequent integration of the above expression and using the boundary condition p=0 at z=$\pm\frac{L}{2}$ (the edges of the bearing) the equation yields to

$$P=\frac{1}{n+1}\,\frac{\left[\frac{U}{2}\frac{dh}{dx}\right]^n\left[z^{n+1}-\left(\frac{L}{2}\right)^{n+1}\right]}{\left[\frac{n}{2(n+1)(2n+1)}\left(\frac{h^{2n+1}}{\mu}\right)^{\frac{1}{n}}+\frac{k_y H}{\mu}\right]^n} \qquad (12)$$

For n=1 the equation reduced to the usual form of the Reynolds's Equation for Newtonian fluid in the film region (in isotropic condition).

$$X = R\vartheta, \quad C=\frac{(D-d)}{2}, \quad \varepsilon=\frac{e}{C}, \quad \dddot{U}=\frac{R\omega}{2}$$

$$h = C\,(1+\varepsilon\cos\theta), \quad \frac{dx}{d\theta}=R\theta,$$

$$\frac{dh}{d\theta}=-C\varepsilon\sin\theta; \quad \frac{dh}{dx}=-\frac{C\varepsilon\sin\theta}{R};$$

Since the journal axis is parallel to the bearing axis, the pressure distribution is symmetrical about the mid plane and it is sufficient to evaluate the pressure over half of the surface.

The steady state characteristics are calculated from the pressure distribution.

Load carrying capacity is given by

$$W_1 = -\int_0^{\pi}\int_{-L/2}^{L/2} PR\cos\theta \; d\theta \; dz.$$

$$W_2 = \int_0^{\pi}\int_{-L/2}^{L/2} PR\sin\theta \; d\theta \; dz.$$

The total load carrying capacity is given by

$$W=\sqrt{W_1+W_2}$$

Friction is given by $F_s=\iint_A \tau \, dA.$

Where $$\tau=\mu\left[\frac{U}{h}\pm\frac{n}{n+1}\left(\frac{h}{\mu}\frac{dp}{dz}\right)^{1/n}\right]^n$$

Coefficient of friction, f = Fs / w.

To compare the influence of the different assumptions and boundary conditions we calculated the load carrying capacity for all the four cases presented above. Besides load carrying capacity the co efficient of friction was also calculated. During the calculation we considered the changes of the absolute viscosity of the oil film due to temperature by means of Vogel's equation i.e. $\eta(T) = ae^{\frac{b}{T+c}}$ where a,b,c are three constants for the first three cases and for the last case non newtonicity re calculated for different temperature (Baka, 2002). In the first step of iteration the bearing temperature was assumed to be equal to the ambient temperature. In the next step the steady state bearing temperature was calculated taking only into consideration the heat transmission through the surface of the bearing house.

The set up for measurement of coefficient of friction and wear rate is illustrated in the figure 3, the set up consist of test journal which is connected to a shaft of the motor. The journal was press fitted and keyed to the motor shaft. The test bearing is fitted to a Mild Steel sleeve. There is a graduated brass lever brazed to the frame and a small

Table 1. Results according to CAMERON eq. (2)

ε	100 r.p.m	200 r.p.m	300 r.p.m
0.3	W=457.8 ,f=0.00771(41.6^{0}C)	W=714.6, f=0.00771(46.6^{0}C)	W=720.2, f=0.00771(55^{0}C)
0.7	W=1377.1, f=0.00357(42.4^{0}C)	W=1917.5, f=0.00357(49.6^{0}C)	W=1725.7,f=0.00357(61.6^{0}C)
0.9	W=1813.6, f=0.004(44^{0}C)	W=2065.5, f=0.004(55.8^{0}C)	W=1435.8,f=0.004(75.7^{0}C)

Table 2. Results according to ROULEAU eq.(5)

ε	100 r.p.m	200 r.p.m	300 r.p.m
0.3	W=464.3, f=0.00727(41.6^{0}C)	W=736.1, f=0.00727(46.3^{0}C)	W=747.5, f=0.00727(54.2^{0}C)
0.7	W=1430.9, f=0.00279(42^{0}C)	W=2146.4, f0.00279(47.7^{0}C)	W=2092.7, f=0.00279(57.3^{0}C)
0.9	W=1974.6, f=0.00311(43.1^{0}C)	W=2494.2, f=0.00311(52.4^{0}C)	W=2026.5, f=0.00311(68^{0}C)

Table 3. Results according to PRAKASH eq.(8)

ε	100 r.p.m	200 r.p.m	300 r.p.m
0.3	W=388, f=0.00784(41.4^{0}C)	W=626.4, f=0.00784(45.6^{0}C)	W=673.1, f=0.00784(52.6^{0}C)
0.7	W=1291.3, f=0.0026(41.6^{0}C)	W=2047.2, f=0.0026(46.3^{0}C)	W=2078.7, f=0.0026(54.2^{0}C)
0.9	W=1969.2, f=0.002(42^{0}C)	W=2953.8, f=0.002(47.8^{0}C)	W=2806.1, f=0.002(57.7^{0}C)

Table 4. Results according to eq.(12), taking n=1.3

ε	100 r.p.m	200 r.p.m	300 r.p.m
0.3	W=439.974, f=0.00446	W=590.874, f=0.00446	W=700.0966, f=0.00446
0.7	W=1242.49, f=0.00294	W=1335.72,f=0.00294	W=1262.764,f=0.00294
0.9	W=1886.217,f=0.00317	W=2413.145, f=0.00317	W=2782.006,f=0.00317

Table 5. Results according to eq.(12), taking n=1.35

ε	100 r.p.m	200 r.p.m	300 r.p.m
0.3	W=458.342, f=0.00479	W=658.532, f=0.00479	W=813.8438, f=0.00479
0.7	W=1752.461, f=0.00293	W=1918.044,f=0.00293	W=1815.79, f=0.00293
0.9	W=2287.673, f=0.0028	W=2831.339,f=0.0028	W=3080.84, f=0.0028

Table 6. Results according to eq.(12), taking n=1.4

ε	100 r.p.m	200 r.p.m	300 r.p.m
0.3	W=576.75,f=0.00421	W=730.33, f=0.00421	W=928.8405,f=0.00421
0.7	W=1822.776, f=0.00296	W=2698.993, f0.00296	W=2761.45, f=0.00296
0.9	W=2761.577, f=0.00248	W=4926.483, f=0.00248	W=4511.72, f=0.00248

spirit level is rigidly connected to the top surface of the frame with a thin layer of araldite. Normally the lever is in balanced condition and the spirit level shows the horizontality of the lever**.** Six holes are radially drilled in the bearing casing as well as each sample. Copper tubes are inserted through the holes in the casing. There is a set up of manometric glass tubes which are connected to the glass tubes through flexible rubber tubes. An inlet tube is also provided for insertion of lubricating oil in the clearance between journal and the shaft. Due to pressure variation for hydrodynamic effect the rise in the oil level in manometric tubes are different. The difference for each sample is to be compared.

Fig. 3. Experimental set-up for measuring coefficient of friction of copper infiltrated P/M journal bearing.

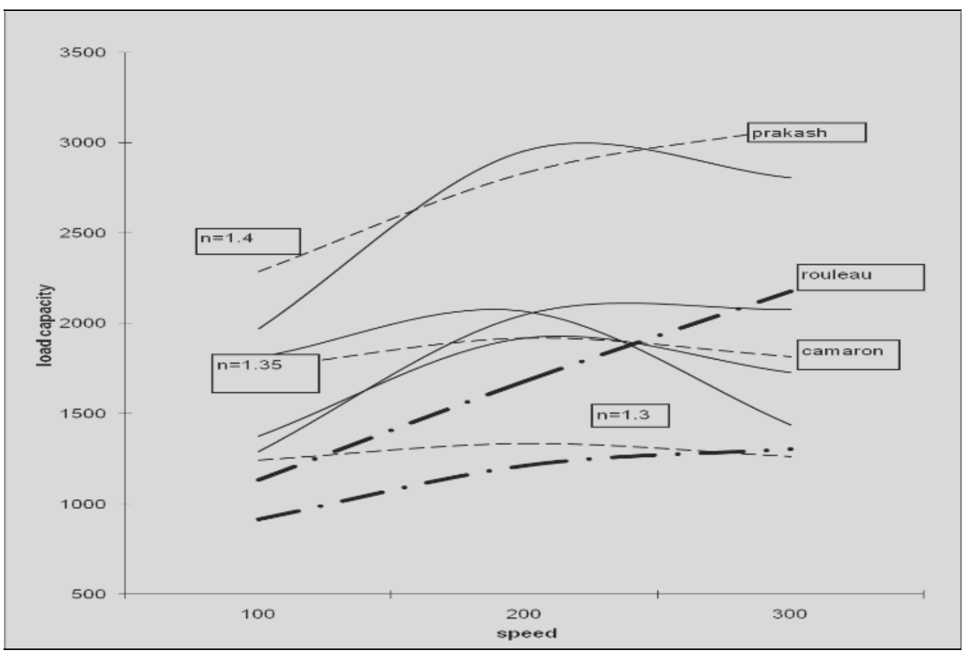

Fig. 4. Shows the variation of load capacity with the speed of the shaft.

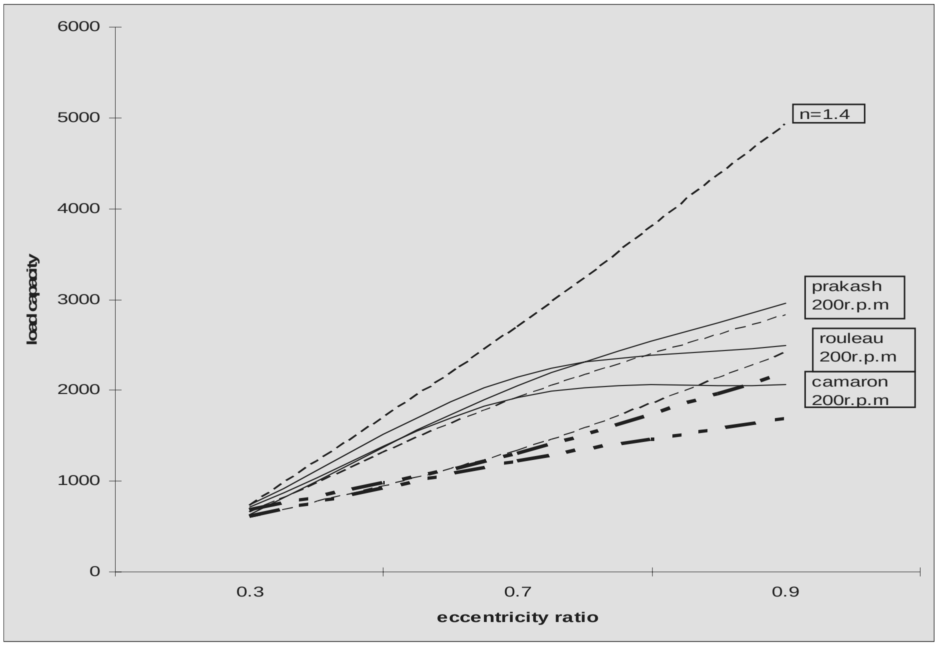

Fig. 5. Shows the load capacity at different eccentricity ratio.

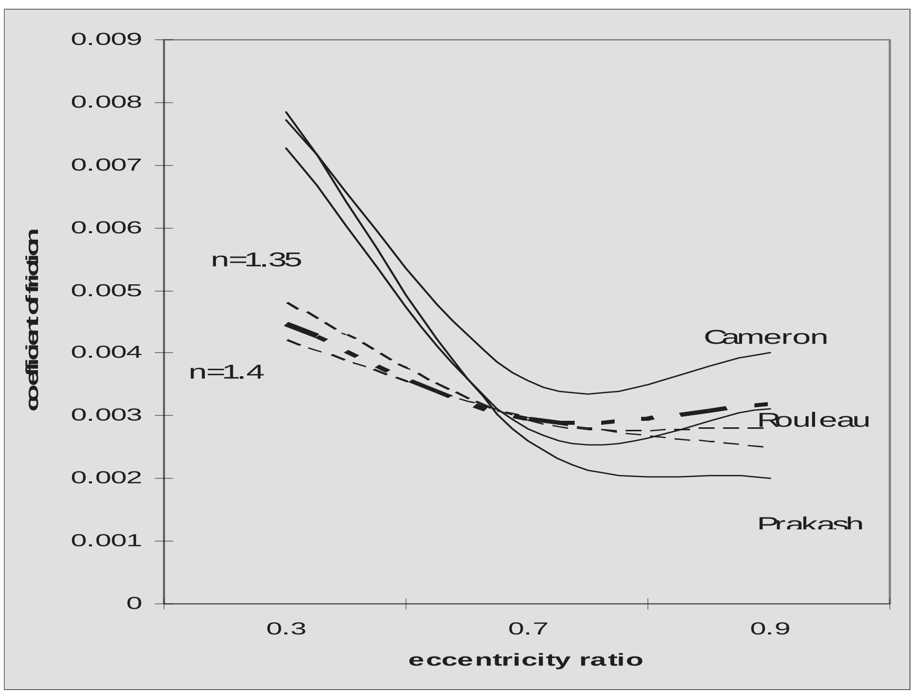

Fig. 6. Shows the co efficient of friction at different eccentricity ratio.

The co-efficient of friction f is calculated as follows: -

$f = (W_b .L)/ (W_d.D_j)$.

Where L = Distance from the centre of the lever to the point where the balance weight is placed, in mm.

W_b = Balance weight in Kg.

W_d =Dead weight in Kg.

D_j = Journal diameter in mm.

RESULTS AND DISCUSSION

The results of the calculations are summarized in the tables from 1-6. For the sake of better demonstration the difference between the results of calculation according to the above mentioned authors, the calculated load carrying

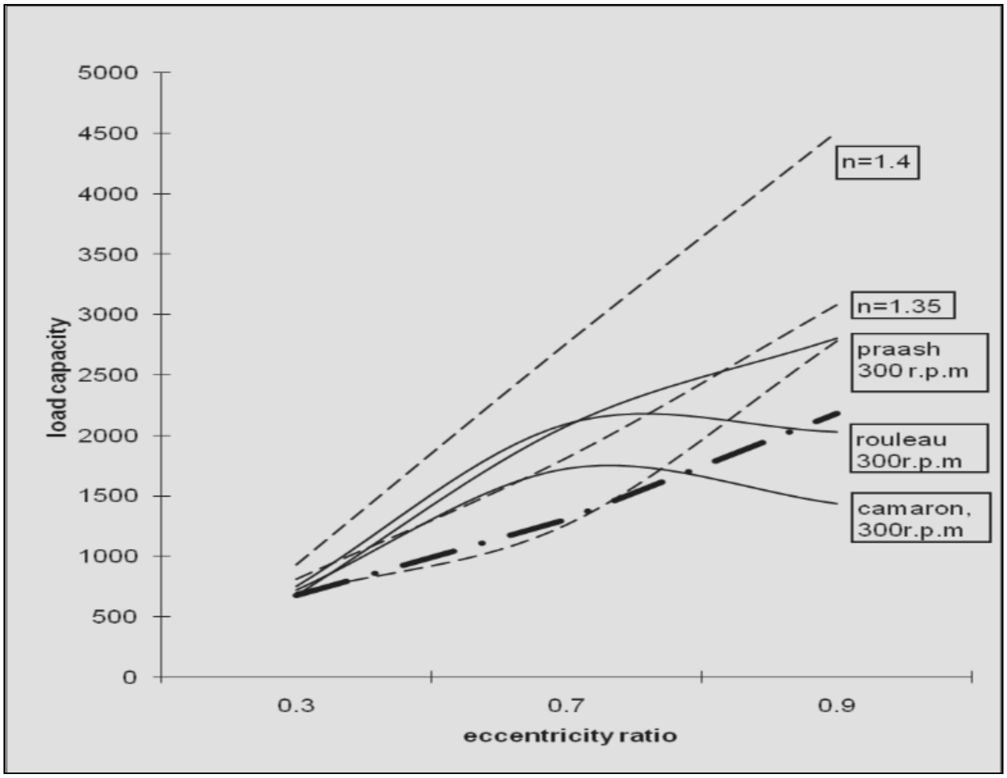

Fig. 7. Shows load capacity at 300 r.p.m for various eccn. Ratio.

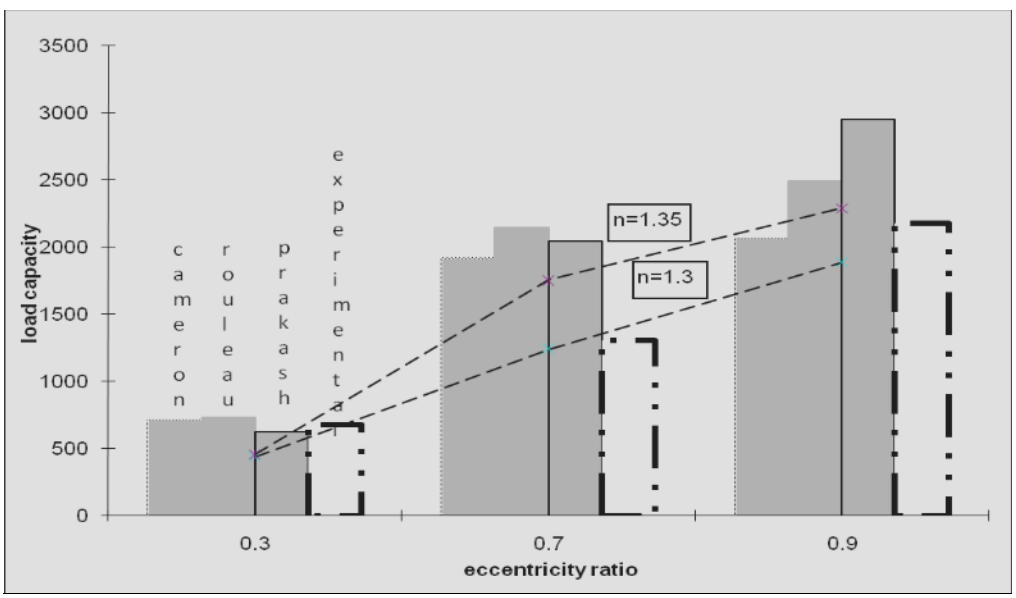

Fig. 8. Shows load capacity with the eccentricity ratio and compared with experimental data.

capacity at eccentricity ratio 0.3, 0.7, 0.9 and also the coefficient of friction in the figure 4 -7.

As it can be seen from the tables and those figures the differences between the calculated load carrying capacity at low temperature and low eccentricity ratio are in considerable. In the case of the last equation carrying non newtonicity parameter also fits well within the range (taking n=1.3 to1.4). For higher eccentricity ratio there are considerable differences between the calculated load carrying capacities. Using non Newtonian fluid for n=1.3 to 1.35 fits the calculated loads within the temperature range from 42^0C to 49^0C very well. As well as the

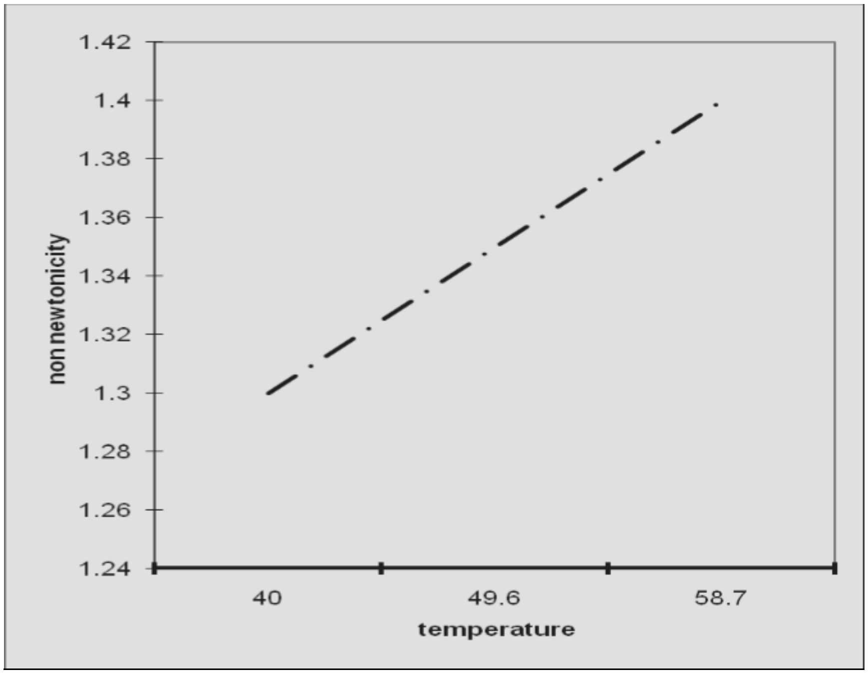

Fig. 9. Shows the change of non newtonicity with the change of temperature.

experimental data calculated using the above set up supports the results.

CONCLUSION

The non newtonicity of the fluid changes with the change of temperature of the bearing. once the viscosity of the working fluid is known for a particular temperature is known the other required parameter like load carrying capacity, co-efficient of friction can also be calculated by using the equation No. 12 and modifying the value of non newtonicity by equation $n = 1.253e^{0.037T}$ where n=non newtonicity of the fluid and T=the surrounding temperature of the bearing housing in ^{0}C, with respect to the working temperature without the aid of viscosity chart.

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NOMENCLATURE

C= radial clearance, m
D= bearing diameter, m
d = journal diameter, m
f = coefficient of friction
e = bearing eccentricity, m
F = shear force on journal surface, N
h = film thickness, m
N= revolution of the journal, rps
L= length of the bearing, cm
P= film pressure above ambient, N/m2
P* = Pressure in the porous matrix, above ambient, N/m2
R= shaft/bearing radius, m
U = shaft velocity (U = R ω), m/s
W = load on the bearing, N
X, y, z = circumferential, radial and axial Co-ordinates.
$\varepsilon =$ Eccentricity ratio
$\tau =$ Shear stress
K_X, K_y, K_z = permeability in x, y and z directions
θ= dimensionless circumferential coordinate X/R
n= Non-New tonicity of the lubricant.
H = Porous wall thickness, m
Φ= Permeability parameter ($\Phi = K_y H/C^3$)
▬ ▪ ▬ ▪ = experimental data

REFERENCES

Abdalha, A. Elsharkawy. and Lotfi, H Guedouar. 2001. Hydrodynamic lubrication of porous journal bearings using modified Brinkman-Extended Darcy model. Tribology International. 34:767-777.

Baka, E. 2002. Calculation of the hydrodynamic load carrying capacity of porous journal bearings. Periodica Polytechnica Ser. Mech. Eng. 46 (1):3-14.

Cameron, A. 1974. Principles of Lubrication. Longman.

Chen, H. and Chen, D. 2005. Modified Reynolds Equation for non Newtonian fluid with Rheological Model in Frequency domain. Journal of Tribology. October. 127.

Kheng, KA., Tan. and Torng, SAM. 1999. Simulations of The Onset Transient Convection in Porous Media Under Fixed Surface Temperature Boundary Condition. Second International Conference on CFD in Mineral and Process Industries, CSIRO, Melbourne, Australia.

Kumar, S. 1985. A Hand Book, Principles of Metal working. Oxford & IBH Publishing Co., Janapath, New Delhi.

Kumar, S. and Chandra, U. 1981. A analysis of Non-Newtonian hydrodynamic lubrication in plain strain forging process. Wear. 71:293-305.

Kozma, M. 1995. Load Carrying Capacity of dynamically Loaded Journal Bearing, Periodica Polytechnical. 40:39-44.

Morgan, VT. and Cameron, A. 1957. Mechanism of lubrication in porous metal bearings. Proc.Conference on Lubrication and wear, Inst. Mech. Engg. 89:151-157.

Pakdemirli, M., Yilbas, BS and Yurusoy, M. 2004. Entropy Generation in non Newtonian fluid flow in a slider bearing. Saanana. 29:629-640.

Peterson, J., Finn, WE. and Dareing, DW. 1994. Non Newtonian temperature and pressure effects of a lubricant slurry in a rotating hydrostatic step bearing. Tribol. Trans. 37:857-863.

Prakash, J. and Vij, S. 1974. Analysis of Narrow Porous Bearing Using Beavers Joseph Criterion of Velocity Slip. ASME Journal of Applied Mechanics. 348-353.

Raghunandana, K. and Majumdar, BC. 2001. Stability of flexible supported oil journal bearings using non Newtonian lubricant: linear perturbation analysis. ASME Journal of Tribology. 123:651-654.

Rastogi, A. and Gupta, RK. 1991. Accounting for lubricant shear thinning in design of short journal bearings. J. Rheol. 35:589-603.

Rouleau, WT. 1963. Hydrodynamic Lubrication of Narrow Porous Bearing. ASME Journal of Basic Engineering. 123-129.

Zhang, C. 2002. THED behaviuor of non Newtonian dynamically loaded journal bearings in mixed lubrication for direct problem, ASME J. Tribology. 124:178-185.