PERIODICA POLYTECHNICA SER. MECH. ENG. VOL. 46, NO. 1, PP. 3–14 (2002)

CALCULATION OF THE HYDRODYNAMIC LOAD CARRYING CAPACITY OF POROUS JOURNAL BEARINGS

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Received: November 22, 1999

Abstract

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. The porous material was assumed to be isotropic and homogeneous. Four pressure functions were analysed using the short bearing approximation and one pressure distribution with the infinitely long bearing assumption. The load carrying capacity and the coefficient of friction were calculated and compared to one another.

Keywords: porous bearings, hydrodynamic, lubrication.

1. Introduction

Porous journal bearings impregnated with oils are widely used in industrial applications. In many cases they are more advantageous than the non-porous bearings because they do not need continuous lubrication therefore their structure is simple and they also reduce costs. Metal porous bearings can work in fluid lubrication where the rules of operation are determined by the hydrodynamic theory of lubrication. The hydrodynamic theory of porous bearings is not too old, it appeared about five decades ago. This theory is originated from the hydrodynamic theory of lubrication of solid bearings with several alterations.

2. The Hydrodynamic Theory of Lubrication

The hydrodynamic theory of lubrication of journal bearings is older than a century. In his famous experiment Tower has shown first the pressure distribution in the lubricating oil film in the clearance of journal bearings in 1883. Also in this year Petroff measured the friction torque of oil lubricated sliding bearings and created a formula to calculate it. Knowing the results of experiments made by Tower and Petroff Reynolds evolved the base equation of hydrodynamic theory of lubrication of journal bearings from the Navier-Stokes equations using many assumptions. The Reynolds equation cannot be solved in full form therefore it is necessary to make some simplifications to get a simple solution. There are two general simplifications: the *infinitely long bearing* ($b/d = \infty$) and the *short bearing assumption* $\left(\frac{\partial p}{\partial z} \gg \frac{\partial p}{\partial x}\right)$. In 1902 Sommerfeld solved the Reynolds equation making special boundary conditions for pressure distribution in tangential direction called according to him Sommerfeld conditions resulting a central symmetric solution. In the practice the often used boundary conditions are the following: the Sommerfeld (and Gümbel) conditions $p\big|_{\varphi=\pi} = 0$, the Reynolds conditions $\left.\frac{\partial^2 p}{\partial \varphi^2}\right|_{\varphi=\pi} = 0$. Using these assumptions many solutions were achieved during the last century for static and also dynamic operating conditions. Nowadays, numerical methods are often used for solving the Reynolds equation can be seen KOZMA [7].

3. The Hydrodynamic Theory of Lubrication of Porous Journal Bearings

The hydrodynamic theory of lubrication of porous journal bearings originated with CAMERON, [1] who obtained a solution for oil film pressure and load carrying capacity of finite, full bearings using the short bearing assumption. Later ROULEAU [2] modified CAMERON's assumptions and evolved another pressure function. The slip boundary condition was introduced by BEAVERS and JOSEPH [3]. MURTI [5] used this slip condition for infinitely short bearings and got an analytical pressure function with sly mathematical alterations. Later PRAKASH [6], also using the slip condition, solved the Reynolds equation and got a pressure function in form of infinite series.

The first pressure function of the oil film in a porous bearing was determined by CAMERON. 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. He wrote the Reynolds equation of porous bearings in the following form:

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

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 $h(\varphi) = \Delta r(1 + \cos\varphi)$ of a journal bearing and introducing the *permeability parameter* $\Theta = \frac{H\Phi}{\Delta r^3}$ he solved the Reynolds equation with the Sommerfeld boundary condition and got the following pressure function:

$$p(z) = \frac{3U\eta\varepsilon \sin\varphi}{r\Delta r^3\left[(1+\varepsilon\cos\varphi)^3 + 12\Theta\right]}\left(\frac{b^2}{4} - z^2\right). \qquad (2)$$

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

CAMERON also determined the friction force from the shear stresses in the lubricant:

$$F_s = \iint_A \tau \, \mathrm{d}A, \tag{3}$$

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

$$\mu = \frac{\Delta r}{r}\left[\frac{4\pi^2 r^3 \eta b n}{F \Delta r^2 \sqrt{1-\varepsilon^2}} + \frac{\varepsilon}{2}\sin\beta\right]. \tag{4}$$

ROULEAU modified CAMERON's conditions and assumed that the pressure in oil film and in porous material is different. He thought that $\Delta^2 p^*(x, y, z) = 0$, 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:

$$p = \frac{24\eta U b^2}{\pi^3 r \Delta r^2} \sum_{n=1}^{\infty} \frac{(-1)^{n+1} \varepsilon \sin\varphi \cos \pi\beta_n \dfrac{z}{b}}{\beta_n^3 \left[(1+\varepsilon\cos\varphi)^3 + 12\Theta \dfrac{b}{H\pi\beta_n} \tanh \pi\beta_n \dfrac{H}{b}\right]}, \tag{5}$$

where $\beta_n = (2n-1)$.

The coefficient of friction was calculated by ROULEAU using the formula derived by MORGAN and CAMERON.

Some years later BEAVERS and JOSEPH created a new boundary condition named *slip boundary condition*, a schematic view of which can be seen in the *Fig.1*.

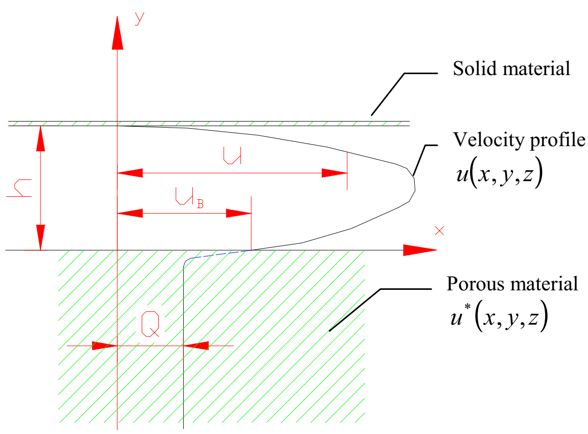


Fig. 1. The torn velocity profile in oil film

The slip boundary condition expresses that there is oil flow in tangential direction in the porous material. The slip condition can be written in the following form:

$$\left.\frac{\partial u(x,y,z)}{\partial y}\right|_{y=0} = \frac{\alpha}{\sqrt{\Phi}}[U_B - u^*]. \tag{6}$$

This slip philosophy altered the governing equations of hydrodynamic lubrication of porous bearings. MURTI applied the slip condition for porous bearings and obtained a pressure distribution for short bearings in analytical form. The velocity profiles from the Navier-Stokes equations, using the (6) boundary condition, in x direction

$$u(x,y,z) = \frac{1}{\eta}\left[\frac{1}{2}\frac{\partial p}{\partial x}(y-h)^2 + (y-h)\left\{\frac{1}{2}\frac{\partial p}{\partial x}h(1+\Sigma_1/3) - \frac{\eta U}{h}\Sigma_0\right\}\right] + \frac{Uy}{h}, \tag{7}$$

and z direction:

$$w(x,y,z) = \frac{1}{\eta}\left[\frac{1}{2}\frac{\partial p}{\partial x}(y-h)^2 + (y-h)\left\{\frac{1}{2}\frac{\partial p}{\partial x}h(1+\Sigma_1/3)\right\}\right], \tag{8}$$

where

$$\Sigma_1 = 3\frac{2\alpha + \frac{h}{\sqrt{\Phi}}}{\frac{h}{\sqrt{\Phi}}\left(1+\alpha\frac{h}{\sqrt{\Phi}}\right)} \quad \text{and} \quad \Sigma_0 = \frac{1}{1+\alpha\frac{h}{\sqrt{\Phi}}}.$$

So the Reynolds equation with slip condition has the following form:

$$\frac{\partial}{\partial x}\left[\frac{h^3}{12\eta}\frac{\partial p}{\partial x}(1+\Sigma_1)\right] + \frac{\partial}{\partial z}\left[\frac{h^3}{12\eta}\frac{\partial p}{\partial z}(1+\Sigma_1)\right] = \frac{U}{2}\frac{\mathrm{d}h}{\mathrm{d}x}(1+\Sigma_0) + \left.\frac{\Phi}{\eta}\frac{\partial p^*}{\partial y}\right|_{y=H}. \tag{9}$$

Solving this equation the pressure distribution can be obtained for short bearings, with the following assumption:

$$p = \frac{1}{H}\int_{y=0}^{y=H} p^*\,\mathrm{d}y. \tag{10}$$

The pressure function:

$$p(z) = \frac{3\eta U}{r\,\Delta r^2}\left(\frac{b^2}{4} - z^2\right) \times \frac{\varepsilon\sin\varphi\left[S^2 + (S+1+\varepsilon\cos\varphi)^2\right]}{(S+1+\varepsilon\cos\varphi)\left[(4S+1+\varepsilon\cos\varphi)(1+\varepsilon\cos\varphi)^3 + 12\Theta(S+1+\varepsilon\cos\varphi)\right]}, \tag{11}$$

where $S = \dfrac{1}{\alpha \dfrac{\Delta r}{\sqrt{\Phi}}}$ the slip coefficient. Using this pressure function the following equation was evolved by MURTI for the coefficient of friction:

$$\mu = \frac{\eta U b r}{\Delta r} \frac{2\pi}{F\sqrt{(1+S)^2 - \varepsilon^2}}. \tag{12}$$

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

$$p(\varphi, z) = 2\sum_{n=1}^{\infty} C_n \cos \lambda_n z \cosh[\lambda_n H]$$

and

$$p^*(\varphi, y, z) = 2\sum_{n=1}^{\infty} C_n \cos \lambda_n z \cosh[\lambda_n (y + H)], \tag{13}$$

where C_n are the unknown coefficients to be determined.

Solving the governing equations the following expressions can be got for the pressure distribution in oil film:

$$p(z) = \frac{24\eta U b^2}{\pi^3 r \Delta r^2} \sum_{n=1}^{\infty} \varepsilon \frac{(-1)^{n+1} g_n(\varphi) \cos\left(\pi \beta_n \dfrac{z}{b}\right)}{(2n-1)^3}, \tag{14}$$

where g_n contains the slip effect:

$$g_l(\varphi) = \frac{\sin\varphi[2S^2 + 2S(1+\varepsilon\cos\varphi) + (1+\varepsilon\cos\varphi)^2]}{(S+1+\varepsilon\cos\varphi)\left[(1+\varepsilon\cos\varphi)^2\left\{6\alpha^2 S^2 + 4S(1+\varepsilon\cos\varphi) + (1+\varepsilon\cos\varphi)^2\right\} + 12\Theta \dfrac{(S+1+\varepsilon\cos\varphi)\tanh\left(\pi\beta_n \frac{H}{b}\right)}{\pi\beta_n \frac{h}{b}}\right]} \tag{15}$$

and for the coefficient of friction:

$$\mu = \frac{\Delta r}{r}\left[\frac{4\pi^2 r^3 b n \eta}{F \Delta r^2 \sqrt{(1+S)^2 - \varepsilon^2}} + \frac{\varepsilon}{2}\sin\beta + \frac{48\eta n b^3 r(1-2\alpha^2) S^2 \varepsilon^2}{\pi^3 F \Delta r^2} \sum_{i=1}^{5} \frac{1}{(2i-1)^4} \int_0^{\pi} \frac{\sin\varphi\, g_i(\varphi)}{(1+S+\varepsilon\cos\varphi)^2} \mathrm{d}\varphi\right]. \tag{16}$$

The above presented four pressure functions used the *short bearing* boundary condition. In the following the governing equations of hydrodynamic lubrication of porous bearings will be solved using the *infinitely long bearing* boundary condition. To obtain the pressure distribution in oil film CAPONE [4] wrote the equations in the following forms:

The Reynolds equation:

$$\frac{1}{r}\frac{\partial}{\partial\varphi}\left(\frac{h^3}{r}\frac{\partial p}{\partial\varphi}\right) = 6\frac{\eta U}{r}\frac{\mathrm{d}h}{\mathrm{d}\varphi} + 12\Phi\left(\frac{\partial p}{\partial y}\right), \tag{17}$$

and the Laplace equation:

$$\Delta p(\varphi, y) = \frac{1}{r^2}\frac{\partial^2 p}{\partial\varphi^2} + \frac{\partial^2 p}{\partial y^2} = 0. \tag{18}$$

Having made integration and the simplifications he got a differential equation:

$$\frac{\mathrm{d}p}{\mathrm{d}\varphi} = 6\frac{\eta U r}{\Delta r^2}\left[\frac{1+\varepsilon\cos\varphi}{(1+\varepsilon\cos\varphi)^3 + 12\Theta} - \frac{h_0}{\Delta r}\frac{1+\varepsilon\cos\varphi}{(1+\varepsilon\cos\varphi)^3 + 12\Theta}\right]. \tag{19}$$

Overcoming many difficulties we solved this differential equation and obtained a pressure distribution in the following form:

$$\begin{aligned} p(\varphi) = \frac{-6\eta U r\varepsilon\sin\varphi}{\Delta r^2}&\left[\frac{1}{6a}\ln\frac{a^2 - a(1+\varepsilon\cos\varphi) + (1+\varepsilon\cos\varphi)^2}{(a+1+\varepsilon\cos\varphi)^2}\right. \\ &\left.+\frac{1}{a^2\sqrt{3}}\operatorname{arc\,tan}\frac{2(1+\varepsilon\cos\varphi)-a}{a\sqrt{3}}\right] \\ +\frac{6\eta U r\varepsilon\sin\varphi}{\Delta r^2}&\left[\frac{h_0}{\Delta r 6a^2}\ln\frac{(a+1+\varepsilon\cos\varphi)^2}{a^2 - a(1+\varepsilon\cos\varphi) + (1+\varepsilon\cos\varphi)^2}\right. \\ &\left.-\frac{h_0}{\Delta r a^2\sqrt{3}}\arctan\frac{2(1+\varepsilon\cos\varphi)-a}{a\sqrt{3}}\right], \end{aligned} \tag{20}$$

where $a = 12\Theta^{0.33}$.

4. The Calculations of the Load Carrying Capacity and the Coefficient of Friction

In order to determine the load carrying capacity of porous bearings we have to integrate the pressure function. To compare the influence of the different assumptions and boundary conditions we calculated the load carrying capacity for all the five cases presented above. Owing to their complexities the five pressure functions can be solved only with numerical methods. Beside the load carrying capacity the coefficient of friction was also calculated.

During the calculations we considered the changes of the absolute viscosity of the oil film due to temperature by means of the following Vogel equation:

$$\eta(T) = ae^{\frac{b}{T+c}}, \tag{21}$$

where a, b, c are constants. These constants were determined from three known viscosity values of the oils. The results of their calculation can be seen in *Table1*:

Table 1. Constants of Vogel equation for some viscosity grades

ISO-VG 46	ISO-VG 100	ISO-VG 150	ISO-VG 220
$\eta_{40} = 0.04$ [Pas]	$\eta_{40} = 0.09$ [Pas]	$\eta_{40} = 0.14$ [Pas]	$\eta_{40} = 0.2$ [Pas]
$\eta_{75} = 0.012$ [Pas]	$\eta_{75} = 0.02$ [Pas]	$\eta_{75} = 0.028$ [Pas]	$\eta_{75} = 0.037$ [Pas]
$\eta_{100} = 0.006$ [Pas]	$\eta_{100} = 0.0095$ [Pas]	$\eta_{100} = 0.013$ [Pas]	$\eta_{100} = 0.017$ [Pas]
$a = 2.27 \cdot 10^{-6}$	$a = 5.95 \cdot 10^{-5}$	$a = 1.1 \cdot 10^{-4}$	$a = 1.92 \cdot 10^{-4}$
$b = 2437.71$	$b = 991.36$	$b = 860.37$	$b = 758.31$
$c = 209.28$	$c = 95.39$	$c = 80.4$	$c = 69.13$

The calculations of the load carrying capacity and the coefficient of friction were carried out on the porous bearing having the following data:

$D1 = 44$ [mm]	$H = 4.5$ [mm]	$\Phi = 530 \cdot 10^{-16}$ [m^2] (measured value)
$D = 35$ [mm]	$\Delta r = 0.02$ [mm]	$\Theta = \Phi H/\Delta r$
$d = 34.96$ [mm]	$\nu = 100$ [mm^2/s]	$\alpha = 0.1$ (accepted value from literature)
$b = 40$ [mm]	$\eta_{40} = 0.09$ [Pas]	$T_{\text{ambient}} = 40$ °C.

4.1. *Method of Calculation*

The calculation of load carrying capacity was carried out by numerical integration of the pressure functions above presented. As lubricating oil was chosen in advance we have to use iteration to obtain the steady state operating temperature of the bearing.

In the first step of the iteration the bearing temperature was assumed to be equal to the ambient temperature. In the next calculation step the viscosity of the lubricating oil was chosen at the value of the bearing temperature determined in the earlier calculation step.

The steady state bearing temperature was calculated taking only into consideration the heat transmission through the surface of bearing house.

The friction power in the bearing can be calculated from this formula:

$$P_S = \mu F U.$$

The heat transmission through the house surface:

$$P_S = \delta A \Delta T = \mu F U,$$

where the coefficient of heat convection $\delta = 17.5$ [W/m^{2}K] and the heat convection surface

$$A = 17.5 \text{ bd}.$$

The calculations were performed at three eccentricity ratios: 0.3; 0.7 and 0.9. The results of the calculations are summarised in *Tables* 2–6 where T_w: working temperature.

Table 2. Results according to CAMERON (*Eq.* (2))

n / ε	$n = 100$ 1/min	$n = 200$ 1/min	$n = 300$ 1/min
$\varepsilon = 0.3$	$T_w = 41.6$ $\eta = 0.082$ $F = 457.8$ $\mu = 0.00771$	$T_w = 46.6$ $\eta = 0.064$ $F = 714.6$ $\mu = 0.00771$	$T_w = 55$ $\eta = 0.043$ $F = 720.2$ $\mu = 0.00771$
$\varepsilon = 0.7$	$T_w = 42.4$ $\eta = 0.079$ $F = 1377.1$ $\mu = 0.00357$	$T_w = 49.6$ $\eta = 0.055$ $F = 1917.5$ $\mu = 0.00357$	$T_w = 61.6$ $\eta = 0.033$ $F = 1725.7$ $\mu = 0.00357$
$\varepsilon = 0.9$	$T_w = 44$ $\eta = 0.072$ $F = 1813.6$ $\mu = 0.004$	$T_w = 55.8$ $\eta = 0.041$ $F = 2065.5$ $\mu = 0.004$	$T_w = 75.7$ $\eta = 0.019$ $F = 1435.8$ $\mu = 0.004$

Table 3. Results according to ROULEAU (*Eq.* (5))

n / ε	$n = 100$ 1/min	$n = 200$ 1/min	$n = 300$ 1/min
$\varepsilon = 0.3$	$T_w = 41.6$ $\eta = 0.082$ $F = 464.3$ $\mu = 0.00727$	$T_w = 46.3$ $\eta = 0.065$ $F = 736.1$ $\mu = 0.00727$	$T_w = 54.2$ $\eta = 0.044$ $F = 747.5$ $\mu = 0.00727$
$\varepsilon = 0.7$	$T_w = 42$ $\eta = 0.08$ $F = 1430.9$ $\mu = 0.00279$	$T_w = 47.7$ $\eta = 0.06$ $F = 2146.4$ $\mu = 0.00279$	$T_w = 57.3$ $\eta = 0.039$ $F = 2092.7$ $\mu = 0.00279$
$\varepsilon = 0.9$	$T_w = 43.1$ $\eta = 0.076$ $F = 1974.6$ $\mu = 0.00311$	$T_w = 52.4$ $\eta = 0.048$ $F = 2494.2$ $\mu = 0.00311$	$T_w = 68$ $\eta = 0.026$ $F = 2026.5$ $\mu = 0.00311$

For the sake of better demonstration of the differences between the results of calculations according to the above mentioned authors, the calculated load carrying capacities at eccentricity ratios 0.3 and 0.9 are presented in the *Figs.*2 and 3.

As it can be seen from these figures the differences between the calculated load carrying capacity at low eccentricity ratio are inconsiderable for short bearings.

Table 4. Results according to MURTI (*Eq.* (11))

ε \ n	$n = 100$ 1/min	$n = 200$ 1/min	$n = 300$ 1/min
$\varepsilon = 0.3$	$T_w = 41.4$ $\eta = 0.083$ $F = 388.5$ $\mu = 0.00809$	$T_w = 45.7$ $\eta = 0.066$ $F = 610.5$ $\mu = 0.00809$	$T_w = 53$ $\eta = 0.047$ $F = 652.2$ $\mu = 0.00809$
$\varepsilon = 0.7$	$T_w = 41.8$ $\eta = 0.081$ $F = 1247.6$ $\mu = 0.003$	$T_w = 47.1$ $\eta = 0.062$ $F = 1910$ $\mu = 0.003$	$T_w = 56$ $\eta = 0.041$ $F = 1894.6$ $\mu = 0.003$
$\varepsilon = 0.9$	$T_w = 42.3$ $\eta = 0.08$ $F = 1912.9$ $\mu = 0.00255$	$T_w = 49.3$ $\eta = 0.056$ $F = 2678.1$ $\mu = 0.00255$	$T_w = 61.1$ $\eta = 0.033$ $F = 2367.3$ $\mu = 0.00255$

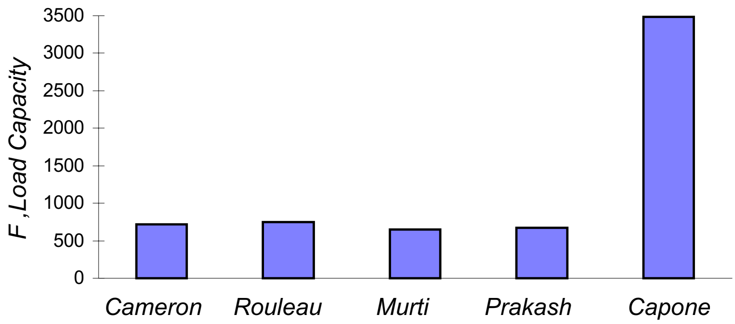
Table 5. Results according to PRAKASH (*Eq.* (14))

ε \ n	$n = 100$ 1/min	$n = 200$ 1/min	$n = 300$ 1/min
$\varepsilon = 0.3$	$T_w = 41.4$ $\eta = 0.083$ $F = 388$ $\mu = 0.00784$	$T_w = 45.6$ $\eta = 0.067$ $F = 626.4$ $\mu = 0.00784$	$T_w = 52.6$ $\eta = 0.048$ $F = 673.1$ $\mu = 0.00784$
$\varepsilon = 0.7$	$T_w = 41.6$ $\eta = 0.082$ $F = 1291.3$ $\mu = 0.0026$	$T_w = 46.3$ $\eta = 0.065$ $F = 2047.2$ $\mu = 0.0026$	$T_w = 54.2$ $\eta = 0.044$ $F = 2078.7$ $\mu = 0.0026$
$\varepsilon = 0.9$	$T_w = 42$ $\eta = 0.08$ $F = 1969.2$ $\mu = 0.002$	$T_w = 47.8$ $\eta = 0.06$ $F = 2953.8$ $\mu = 0.002$	$T_w = 57.7$ $\eta = 0.038$ $F = 2806.1$ $\mu = 0.002$

Table 6. Results according to CAPONE (*Eq.* (20))

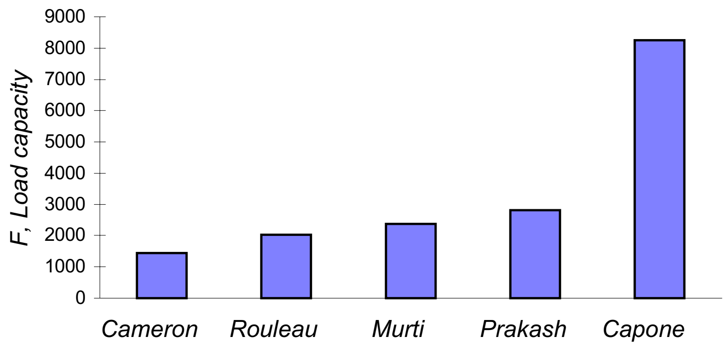
ε \ n	$n = 100$ 1/min	$n = 200$ 1/min	$n = 300$ 1/min
$\varepsilon = 0.3$	$T_w = 42.7$ $\eta = 0.078$ $F = 3238$ $\mu = 0.00173$	$T_w = 51$ $\eta = 0.051$ $F = 4234$ $\mu = 0.00173$	$T_w = 65$ $\eta = 0.028$ $F = 3487$ $\mu = 0.00173$
$\varepsilon = 0.7$	$T_w = 43$ $\eta = 0.076$ $F = 7780$ $\mu = 0.00071$	$T_w = 52$ $\eta = 0.049$ $F = 10032$ $\mu = 0.00071$	$T_w = 66$ $\eta = 0.027$ $F = 8292$ $\mu = 0.00071$
$\varepsilon = 0.9$	$T_w = 44$ $\eta = 0.072$ $F = 9897$ $\mu = 0.00073$	$T_w = 55.5$ $\eta = 0.042$ $F = 11546$ $\mu = 0.00073$	$T_w = 75$ $\eta = 0.02$ $F = 8247$ $\mu = 0.00073$

At the same time at high eccentricity ratio (0.9) there are considerable differences between the calculated load carrying capacities. Taking into consideration that the pressure functions in the oil film and in the porous material are different that increased the load carrying capacity (ROULEAU). Further increasing of load carrying capacity results the using of slip boundary condition (MURTI). PRAKASH and VIJ obtained the pressure function for short bearings with slip boundary condition in a form of infinite series and solved the Reynolds and Laplace equations separately,



ε=0.3, n=300 [1/min]

Fig. 2.



ε=0.9, n=300 [1/min]

Fig. 3.

which led higher load carrying capacity.

Figs. 2 and 3 also show the enormous differences between the load carrying capacities using the *short bearing* and the infinitely *long bearing* assumptions. We mention that the hydrodynamic load carrying capacity of sliding bearings with 'the *finite width* boundary condition' is between the two values. The calculated hydrodynamic load carrying capacities are relatively low, at eccentricity ratio 0.9 because of the *permeability*. The average bearing pressure is 2 [N/mm^2].

5. Conclusions

The solutions presented above of the equations of the hydrodynamic lubrication for porous bearings and the calculated load carrying capacities show that:

- The solutions of equations of hydrodynamic lubrication for porous bearings are more complicated than for the solid sliding bearings.
- There are many new boundary conditions taken into consideration with the influence of the porous bearings on the hydrodynamic effect.
- The short bearing assumption gives more simpler solution than the infinite long bearing assumption.

Acknowledgement

These studies are a part of research programs OTKA T019804 and OTKA F022179 sponsored by the Hungarian Government. For the support of the research programs we would like to express our thanks from this place, too.

Nomenclature

x – tangential coordinate [m]
y – radial coordinate [m]
z – axial coordinate [m]
r – radius of shaft
u – x velocity [m/s]
v – y velocity [m/s]
w – z velocity [m/s]
n – number of revolution [1/s]
d – shaft diameter [m]
b – width of bush [m]
$\Delta r = (D - d)/2$ [m]
D – inner diameter of bush [m]
D_1 – outside diameter of bush [m]
H – wall thickness of bush [m]
ε – eccentricity ratio [-]
φ – circumferential coordinate
η_{xx} – absolute viscosity of oil [Pas]
ν – viscosity of oil [mm^2/s]
μ – coefficient of friction [-]
U – circumferential velocity
T – temperature [°C]
F – load carrying capacity [N]
ϕ – permeability [m^2]

Θ	– permeability parameter [-]
α	– slip coefficient [-]
$S = \frac{\sqrt{\Phi}}{\alpha \Delta r}$	– slip parameter
$p(x, z)$	– pressure in oil film
$p^*(x, y, z)$	– pressure in porous bush
u, v, w	– velocity in oil film
u^*, v^*, w^*	– velocity in porous bush
$h(\varphi) = \Delta r(1 + \cos\varphi)$	– clearance function
h_0	– minimum film thickness

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