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SHORT PLANE BEARINGS LUBRICATION APPLIED ON CHAIN JOINTS

Radu VELICU[1], Lenard JURJ[2]
[1]Transylvania University of Brașov, rvelicu@unitbv.ro
[2] Transylvania University of Brașov, jurj.lenard@unitbv.ro

***Abstract*—**The aim of this paper is to describe the theoretical pressure distribution and loading capacity of chain joints. The lubricating contact between pin and bush (bush chains) or pin and tooth cleat (silent chains) is modelled with Ocvirk's short bearing solution which is based on the assumption that the contact surface between the two elements on width direction is significantly smaller than on length direction. This paper is presenting the steps to obtain the final formula for pressure distribution. Numerical calculations are applied in the case of a bush chain. The loading capacity is calculated depending on lubricant thickness and speed in order to predict the type of lubrication that is present between the pin and bush.

***Keywords*—**chain joint, lubrication, Ocvirk, pressure distribution

I. INTRODUCTION

WORLDWIDE an increased interest is given for performance improvements of automobile engine systems, which can generate a lot of benefits: reduced fuel and oil consumption; increased engine power and torque output; reduction in harmful exhaust emissions; improved, reliability and engine life and longer service intervals [1]–[3].

The object of the present paper is the study of the theoretical pressure distribution and loading capacity of the joints of timing chains. The timing chains are parts of automotive internal combustion engines used to transmit rotational movement from the crankshaft to the camshaft, oil pump and high-pressure pump. The final purpose of the research is the evaluation of the lubrication type in the joints, depending on dimensions, speed and loading.

The pin-bush joint (in the case of bush chains) or pin-toothed cleat joint (in the case of silent chains) is a typical case of plane bearings [4]. The lubrication theory [5]–[8] dealing with plane bearings is so vast, with several models based on different assumptions and simplifications. But, all theoretical models of lubrication, describing the process of hydrodynamic pressure generation can be described mathematically with Reynolds equation.

The Reynolds equation is a partial differential equation describing the pressure distribution of thin viscous fluid film bearings, within bodies are fully separated by thin film of fluid. The general equation is

$$\begin{aligned}&\frac{\partial}{\partial x}\left(\frac{\rho h^3}{12\eta}\frac{\partial p}{\partial x}\right)+\frac{\partial}{\partial y}\left(\frac{\rho h^3}{12\eta}\frac{\partial p}{\partial y}\right)=\\&=\frac{\partial}{\partial x}\left(\frac{\rho h(u_a+u_b)}{2}\right)+\frac{\partial}{\partial x}\left(\frac{\rho h(\upsilon_a+\upsilon_b)}{2}\right)+\\&+\rho(w_a-w_b)-\rho u_a\frac{\partial h}{\partial x}-\rho\upsilon_a\frac{\partial h}{\partial y}+h\frac{\partial \rho}{\partial z}.\end{aligned} \tag{1}$$

The parameters from (1) are: p – the fluid film pressure; x and y are the two axes describing bearing length and width; z fluid film thickness coordinate; h fluid film thickness; η is the fluid viscosity; ρ is the fluid density; u, υ and w are the bounding body velocities in x, y, z directions a and b the top and bottom boundaries. The assumptions considered in Reynolds equation: the fluid is Newtonian; the viscous forces dominate over the fluid inertia forces; the fluid body forces are negligible; the variation of pressure across the fluid film is small; the fluid film thickness is much less than the length and width.

II. APPLIED OCVIRK'S SHORT BEARING SOLUTION

A. *Theoretical model*

Ocvirk's short bearing solution [5] is a simplified model of Reynolds equation. This model is applicable for bearings with relatively reduced width where the lubricant loss plays a big role. In this case the pressure peak is obtained on a reduced longitudinal section of the bearing, due to the incapacity to maintain the lubricant between the friction surfaces. It is based on the assumption that the contact surface on x direction is so reduced compared to the y direction that it can be neglected; the lubricant thickness tends to 0, can be also neglected.

The model can be applied to bush chain joints where providing the necessary oil in the pin bush space is the main focus for improvement. This mean holes in the bush through which oil flows in, but also, easily out. This diminishes the capacity to create a fluid film in the direction of rotation (*x*), while lubricant losses at the edges of the bush are important. A similar situation is the case of toothed chains (silent chains) where the cylindrical joint between the pin and mobile cleat has a short width and is subject to the same important lubricant losses at the edges.

Fig. 1. presents the diagram of a general plane bearing lubrication.

In this case the pressure distribution is varying within an angle φ from 0 to 180 degrees.

The unidirectional velocity approximation of Reynold's equation is

$$\frac{\partial}{\partial x}\left(\frac{h^3}{12\eta}\cdot\frac{\partial p}{\partial x}\right)+\frac{\partial}{\partial y}\cdot\left(\frac{h^3}{12\cdot\eta}\cdot\frac{\partial p}{\partial y}\right)=u\cdot\frac{\partial h}{\partial x}+\frac{\partial h}{\partial t} \tag{2}$$

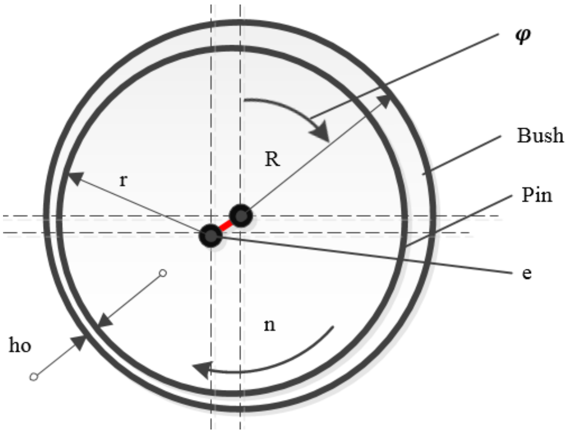

Fig. 1. Cylindrical joint with minimum lubricant thickness

Calculus starts with the following mathematical assumptions:

$$\frac{\partial}{\partial x}\left(\frac{h^3}{12\eta}\frac{\partial p}{\partial x}\right)<<\frac{\partial}{\partial y}\left(\frac{h^3}{12\eta}\frac{\partial p}{\partial y}\right); \tag{3}$$

$$\frac{\partial h}{\partial t}\rightarrow 0\,. \tag{4}$$

The Reynold's unidirectional velocity approximation (1) becomes Ocvrik's equation

$$\frac{\partial}{\partial y}\left(\frac{h^3}{12\eta}\cdot\frac{\partial p}{\partial y}\right)=u\frac{\partial h}{\partial x}\,. \tag{5}$$

Based on the diagram from Fig. 1., the following relations can be drawn:

$$u=\frac{\pi n}{30}R;\quad x=R\cdot\varphi;\quad c=R-r;\quad e=R-r-h_0;\quad h=c+e\cdot\cos\varphi\,. \tag{6}$$

u is the relative average speed between bush and pin in rotational movement, e is the eccentricity between the pin and bush and h_o is the minimum lubricant thickness.

After integration, (5) becomes

$$\frac{h^3}{12\eta}\frac{\partial p}{\partial y}=u\frac{\partial h}{R\partial\varphi}+C_1 \tag{7}$$

Applying the boundary condition

$$\frac{\partial p}{\partial y}=0,\ \text{at}\ y=0\ , \tag{8}$$

it results $C_1=0$ and

$$\frac{\partial p}{\partial y}=\frac{12\eta u}{h^3 R}\frac{\partial h}{\partial\varphi}y\,. \tag{9}$$

After the second integration, the equation of pressure is

$$p=\frac{12\cdot\eta\cdot u}{h^3\cdot R}\cdot\frac{\partial h}{\partial\varphi}\cdot\frac{y^2}{2}+C_2 \tag{10}$$

By applying the boundary conditions for the pressure, at the limits of width of the bush, pressure *p* at $\pm\frac{B}{2}$ is equal to 0, where *B* is the width of the bush, the constant C_2 becomes

$$C_2=-\frac{12\eta u}{h^3 R}\frac{\partial h}{\partial\varphi}\frac{B^2}{8} \tag{11}$$

and (10) becomes

$$p=\frac{6\eta u}{h^3 R}\frac{\partial h}{\partial\varphi}\left(y^2-\frac{B^2}{4}\right). \tag{12}$$

Integrating *h* from (6), the pressure distribution is

$$p=\frac{6\eta u}{R}\frac{e\sin\varphi}{(c+e\cos\varphi)}\left(\frac{B^2}{4}-y^2\right). \tag{13}$$

expressing a function *p* (φ, *y*) on intervals φ ϵ [0, 180], $y\ \epsilon\left[-\frac{B}{2},+\frac{B}{2}\right]$.

The loading capacity *F* of the bearing is

$$F=\iint_{\varphi\,y} p\,dy\,d\varphi\,. \tag{14}$$

The type of lubrication represents also a point of interest.

The dimensionless film thickness ratio Λ [6], indicating the type of lubrication is computed as

$$\Lambda=\frac{h_0}{\sqrt{R_{apin}^2+R_{abush}^2}}\,, \tag{15}$$

where: R_a is the average surface roughness of the pin and bush.

The range of values of Λ [6] indicates:

$\Lambda < 1$ - boundary lubrication;
$1 < \Lambda < 5$ - mixed lubrication;
$3 < \Lambda < 10$ - elasto-hydrodynamic lubrication (EHL);
$5 < \Lambda < 100$ -hydrodynamic lubrication(HDL).

B. Results – Pressure Distribution

The mathematical model for bearing lubrication presented above has been applied to the case of a bush chain joint, using as input parameters three values for minimum lubricant thickness h_0 (0.75 μm, 1 μm, 2 μm). and three values of the rotational speed of the chain sprocket n (1000 rpm, 3000 rpm, 5000 rpm). Table I presents the numerical results for the case of minimum loading: h_0 = 2 μm (maximum from the three values) and rotational speed n = 1000 rpm (minimum of the three values). For the same case, Fig. 2. shows the pressure distribution and Fig. 3. shows the peak pressure at the middle of the bush.

TABLE I
BEARING PARAMETERS FOR h_0 = 2 μm, n = 1000 rpm

Symbol	Unit	Value
h_0	m	2.00E-06
n	rpm	1000
B	m	0.0076
R	m	0.00163
r	m	0.00158
c	m	0.00005
η	Pas	0.06
ω	rad/s	104.7197551
u	m/s	0.164933614
e	m	4.80E-05
R_a	m	4.00E-07
Λ		3.52
F	N	10.467

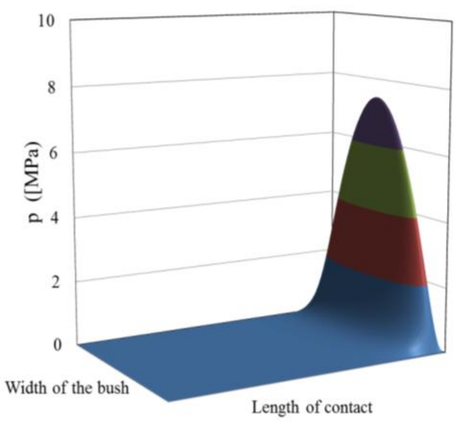

Fig. 2. Pressure distribution for n =1000 rpm, h_0 = 2 μm.

Table II presents the numerical results for the case of maximum loading: h_0 = 0.75 μm (minimum from the three values) and rotational speed n = 5000 rpm (maximum of the three values). For the same case, Fig. 4. shows the pressure distribution and Fig. 5. shows the peak pressure at the middle of the bush.

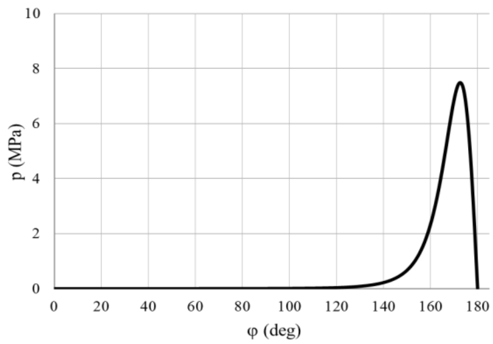

Fig. 3. Peak pressure at the middle of the bush for n = 1000 rpm at h_0 = 2 μm.

TABLE I
BEARING PARAMETERS FOR h_0 = 0.75 μm, n = 5000 rpm

Symbol	Unit	Value
h_0	m	0.75E-06
n	rpm	5000
B	m	0.0076
R	m	0.00163
r	m	0.00158
c	m	0.00005
η	Pas	0.06
ω	rad/s	523.5987756
u	m/s	0.824668072
e	m	4.925E-05
R_a	m	4.00E-07
Λ		1.276
F	N	616.5

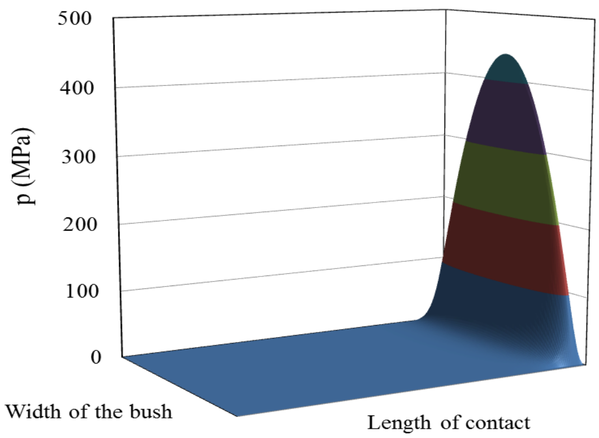

Fig. 4. Pressure distribution for n = 5000 rpm at h_0 = 0.75 μm.

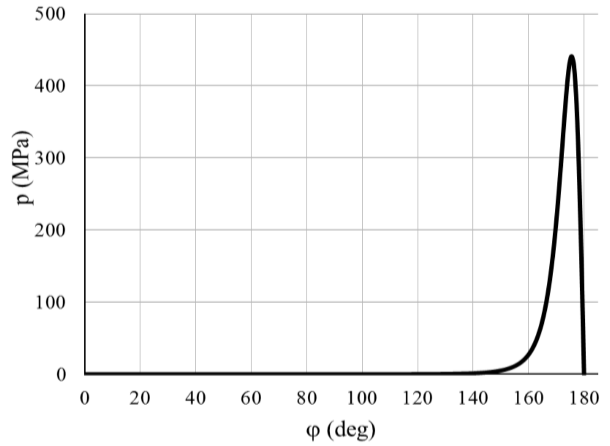

Fig. 5. Peak pressure at the middle of the bush for n = 5000 rpm, h_0 = 0.75 µm.

C. Results – Type of Lubrication

The analysis of the type of lubrication must involve the dependence between minimum lubricant thickness h_0, loading capacity F and rotational speed n. Fig. 6. shows the loading capacity depending on rotational speed, for three values of minimum lubricant thickness.

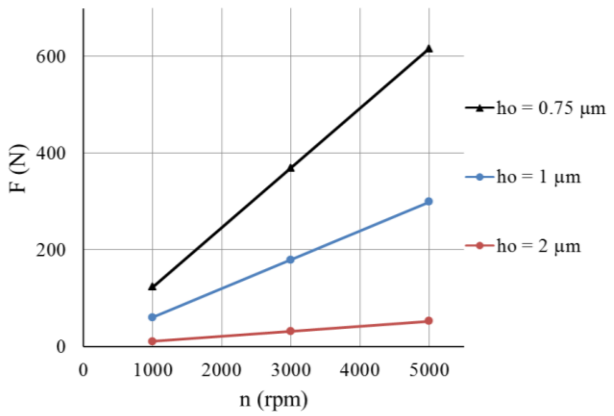

Fig. 6. Loading capacity depending on rotational speed for constant minimum lubricant thickness.

Fig. 7. presents the calculated dimensionless film thickness ratio Λ for average surface roughness of pin and bush R_a = 0.4 µm depending on load, for three values of rotational speed.

The results of the theoretic analysis for the pin bush contact parameters determination, based on the lubrication theory show that:

1) The maximum pressure values are obtained at the areas very near to the minimum lubricant thickness;

2) Looking at the values of the maximum pressure, limited to 1000 MPa, we can conclude that the theory of hydrodynamic lubrication is applied correctly; there is no need to consider elastohydrodynamic lubrication;

3) The normal condition of working for such joints is in the range [9]: rotational speed n from 500 to 5000 rpm, normal force F from 100 to 1500 N; The small values of the lubricant film thickness indicate that it is nearly impossible to obtain conditions for hydrodynamic lubrication. The conditions show mostly mixed lubrication or boundary lubrication.

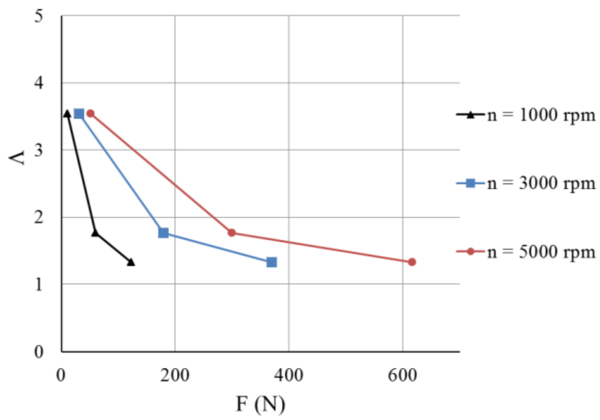

Fig. 7. Film thickness ratio Λ depending on rotational speed for constant rotational speed.

III. CONCLUSION

The main conclusion from theoretical analyse of the lubrication conditions of chain joints is that hydrodynamic lubrication is almost impossible to achieve.

However, this conclusion must be proved by experimental testing.

REFERENCES

[1] K. Holmberg, P. Andersson, A. Erdemir, “Global energy consumption due to friction in passenger cars,” *Tribology International* 47 (14), 2012.

[2] T. Hyakutake, M. Inagaki, M. Matsuda, N. Hakamada, Y. Teramachi, “Measurement of friction in timing chain,” *JSAE Review* 22:5, 2001.

[3] M. Schwaderlapp, F. Koch, J. Dohmen, “Friction Reduction - the Engine’s Mechanical Contribution to Saving Fuel,” *Seoul 2000 FISITA World Automotive Congress*, Seoul, Korea, 2000.

[4] M. T. Lates, “Bush chains design process,” *Annals of the Oradea University, Fascicle of Management and Technological Engineering*, vol. XI (XXI), nr.2, pp.2.51-2.55, 2012.

[5] P.K. Goenka, K.P. Oh, “An Optimum Short Bearing Theory for the Elastohydrodynamic Solution of Journal Bearings,” *Transaction of ASME Series F, Journal of Tribology* 108, pp. 294-299, 1986.

[6] J. Williams, *Engineering tribology*, Ed. Cambridge University Press, New York, 2011.

[7] W. Shizhu, H. Ping, *Principles of tribology*, Ed. Tsinghua University Press, Singapore, 2012.

[8] G. W. Stachowiak, A.W. Batchelor, *Engineering tribology*, Ed. Elsevier, 3rd ed., Burlington, 2005.

[9] M. T. Lates, C. Gavril. “Friction phenomenon in polyamide – steel plate front face type contacts,” *Annals of the Oradea University, Fascicle of Management and Technological Engineering*, vol. XIII (XXIII), ISSUE 1, pp.75-78, 2014.