Proceedings of the 15th International Conference on Manufacturing Systems – ICMaS
Published by Editura Academiei Române, ISSN 1842-3183

"Politehnica" University of Bucharest, Machine and Manufacturing Systems Department
Bucharest, Romania, 26–27 October, 2006

QUASI-DYNAMIC EFFECTS IN BALL BEARINGS WITH 2, 3 OR 4 CONTACT POINTS

Daniel REZMIRES, Cezar RACOCEA

Abstract: *The paper presents a unitary model that is proposed to describe the quasi-dynamic behavior of ball bearings either with 2, 3 or 4 contact points (4PCBB). The model has five degrees of freedom and allows to: (i) predict the internal load distribution and calculating the rigidity matrix; (ii) determine the ball and cage kinematics by the principle of power minimization; and (iii) describe the interactions between various rolling bearing elements in terms of normal and tangential loading, and lubricant film thickness.*

Key words: *ball bearings kinematics, cage speeds, sliding speeds, quasi-dynamic effects, 4PCBB.*

1. INTRODUCTION

Hamrock, 1973, [4] presented a mathematical model for ball bearings with arched outer race that corresponds to the case of 3 contact points bearings.

Also, most of the existing ball bearing analysis used the assumption that balls roll with spin on one raceway only [5]. In spite of this, recent experimental studies on high-speed lubricated ball bearings have revealed that the power loss by ball spinning is well shared between the inner and the outer race [9, 11].

In many cases the bearing raceways geometry and the load distribution generate the Non-Hertzian contacts responsible for the bearing premature failure.

The bearing elements kinematics has been solved considering a mixed control of balls between bearing's raceways.

The cage-balls interactions, as well as the cage-ring interactions, (the short bearing effect), have been considered to obtain angular rotational speed of the cage.

The traction forces involved in the dynamic equilibrium have been obtained using a Non-Newtonian rheological, model when the asperity effects have also been considered.

2. QUASI-DYNAMIC ANALYSIS

Quasi-dynamic analysis is needed to attain more accurate values for the kinematics parameters and further, to evaluate the power losses in the 4PCBB structure.

In this purpose the lubricant characteristics at the working temperature and the quasi-static parameters of the structure have to be known.

2.1. Lubricant viscosity and piezoviscosity factor

The WLF mathematical model, modified according to Yasotomi [12], is used to evaluate the lubricant dynamic viscosity. In the WLF model the dynamic viscosity, denoted $W_{LF}(T, p)$ depends on temperature and pressure.

2.2. Isothermal lubricant film thicknesses

Hamrock and Dowson formulae [3] for isothermal steady state full flooded conditions and elliptical point contact geometry are applied for the contact (j, idx) to obtain the film thicknesses $h_{min}(i\mathrm{d}x, j)$ and $h_{cen}(i\mathrm{d}x, j)$.

The thermal reduction factor Φ_t and the mathematical formulae are presented by Olaru in [9].

2.3. The lubricant shear stress

The model proposed by Houpert [6] has been used to evaluate the lubricant shear stress in the point $P(x, y)$ of contact area (j, idx). The mathematical model is shown also by Olaru in [10] and is used to use for evaluation of the tangential traction forces.

2.4. Angular rotational speeds

The quasi-dynamic equilibrium developed for 4PCBB structures establishes the angular rotational speeds for balls and cage considering both, the normal and the tangential forces which act between bearing's elements. The ball and cage kinematics are functions of ball angle according to [14–16].

2.5. Ball's equilibrium equations

The normal and tangential forces accounted for ball's equilibrium equations are schematically presented in Fig. 1, where: F_H is the hydrodynamic force, F_R is the rolling force, F_L is the viscous friction force on the rolling direction, F_A is the friction force due to the asperity tractions, F_{AL} is the air-lubricant resistant forces.

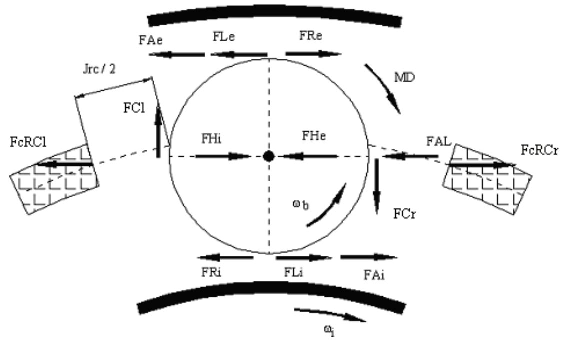

Fig. 1. Forces accounted for ball's quasi-dynamic equilibrium.

The general form of ball's quay-dynamic equilibrium equation is:

$$M(\omega_{b,j}, \omega_c) = 0, \tag{1}$$

$$\sum F(\omega_{b,j}, \omega_c) = F_{cRC,l,r}, \tag{2}$$

where: indices l and r indicate the contact in the left part or right part of the ball respectively, Fig. 1; F_{cRC} is the contact load between ball j and cage; the $F_{CBC}(j)$ vector contains all F_{cRC} contact forces between balls and cage.

The sense of the ball-cage contact force $F_{CBC}(j)$ is presented in Fig. 2.

Considering the previously mentioned forces the ball's equilibrium equations become:

$$\sum M(\omega_{b,j},\omega_c) = \frac{d_w}{2}\left[S_{Fi} + S_{Fe} - F_{cRC}\mu_{RC}\right] = 0, \tag{3}$$

$$\sum F(\omega_{b,j},\omega_c) = S_{Fi} - S_{Fe} + F_{Hi} - F_{He} = F_{CBC}(j), \tag{4}$$

with:

$$S_{FI} = F_{A,\,idx} + F_{L,\,idx} - F_{R,\,idx} \quad (idx = 1, 2), \tag{5}$$

$$S_{Fe} = F_{A,\,idx} + F_{L,\,idx} - F_{r,\,idx} \quad (idx = 3, 4). \tag{6}$$

2.6. Contact forces between cage and rings. Short bearing effect

The frequent case of the cage guided on the inner raceway(s) is presented in Fig. 3 [15]. For this situation the global effect of $F_{CBC}(j)$ forces is a contact force denoted F_{cCR}:

$$F_{cCR} = \sqrt{s_{Fax}^2 + s_{Frz}^2}, \tag{7}$$

where:

$$\begin{aligned} s_{Fax} &= \sum_{j=0}^{Z-1}\begin{bmatrix} F_{CBC}(j)\cos(\psi(j)+\Delta\psi+u)+ \\ +\mu_{C\_BI}F_{CBC}(j)\sin(\psi(j)+\Delta\psi+u) \end{bmatrix}, \\ s_{Frz} &= \sum_{j=0}^{Z-1}\begin{bmatrix} -F_{CBC}(j)\sin(\psi(j)+\Delta\psi+u) \\ +\mu_{C\_BI}F_{CBC}(j)\cos(\psi(j)+\Delta\psi+u) \end{bmatrix}, \end{aligned} \tag{8}$$

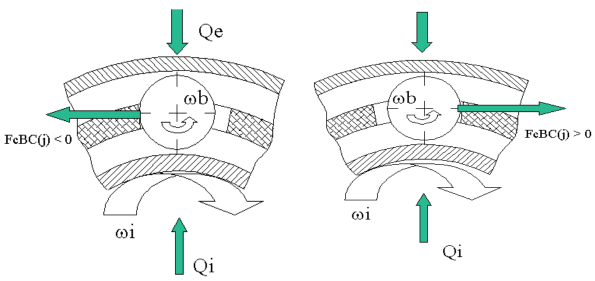

Fig. 2. The sense for ball-cage contact force.

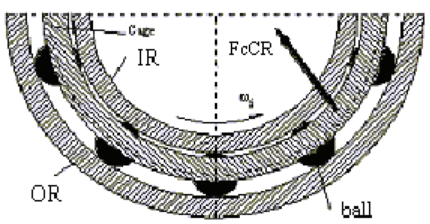

Fig. 3. Cage-inner ring sliding contact.

with:

$$u \approx \arctan\left(\frac{d_w}{d_m}\right), \quad \psi(j) = \frac{2\pi j}{Z}, \quad \Delta\psi \subset \left[0....\frac{2.\pi}{Z}\right]. \tag{9}$$

and $\psi + \Delta\psi$ [rad], being the angular position of the ball.

The contact between cage and one of bearing's rings is assumed as a short journal bearing, with main geometrical parameters [11, 15] presented in Fig. 4.

In this assumption the Sommerfeld number, denoted S_o, governs all the computation [7]:

$$S_o = \frac{W_{LF}(T,0)n}{p_m\Psi^2}, \tag{10}$$

with:

$$\Psi = \frac{D - D_r}{D_r}, \quad n = \left|\omega_{i,e} - \omega_c\right| \text{ [rot/s]}, \qquad p_m = \frac{F_{cC}R}{L_c D_r} \text{ [Pa]}, \tag{11}$$

where: D is the diameter of the surface cage in contact with the ring ($D_{\text{min cage}}$ or $D_{\text{max cage}}$, for inner or outer ring contact, respectively); D_r is the diameter of the ring surface in contact with the cage (D_{eiav} or D_{eeav}).

The necessary link between the Sommerfield number so and the relative eccentricity ε is adopted as that presented by Frene [7] and Olaru [10]:

$$S_o = \frac{D}{L_c}\frac{\left(1-\varepsilon^2\right)^2}{\pi\varepsilon\sqrt{16\varepsilon^2 + \pi^2\left(1-\varepsilon^2\right)}}. \tag{12}$$

The values of the friction coefficient result as:

$$\mu = \frac{\Psi\pi^2 S_o(2+\varepsilon)}{(1+\varepsilon)\sqrt{(1-\varepsilon)}}. \tag{13}$$

The functions $F_{cCR}\mu$ represent an active force for the inner guiding case and a resistant force for the outer guiding case.

The torque equilibrium equation applied to cage provides the values of angular rotational speeds ω_c and $\omega_{b,j}$.

$$Mc(\omega_{b,j},\omega_c) \approx \left[\sum_j F_{CBC}(j)\frac{d_m}{2}\right] \pm F_{cCR}\mu\frac{D_r}{2} = 0. \tag{14}$$

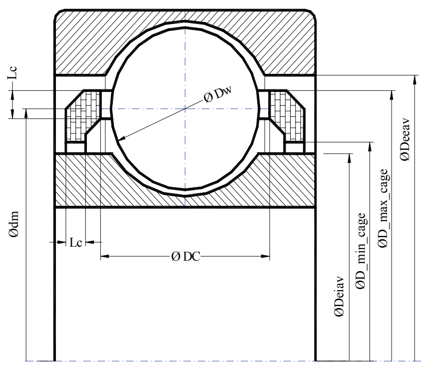

Fig. 4. Cross-section bearing geometry.

3. APPLICATIONS

The presented mathematical model is applied to the 4PCBB-1234 and 4PCBB-13 bearing types, both of them having the cages guided on the inner rings. The main geometrical parameters are presented in Table 1. The lubricant is OIL Mobil Jet II, and the lubricant debit is $Q_h = 3$ l/h.

In the followings, three analyses, (A1, A2) are presented, in all cases the inlet contact temperature is $T = 120$°C.

3.1. Effect of inner ring rotational speed on some of quasi-static and quasi-dynamic parameters of a 4PCBB-1234 structure

The case of 4PCBB-1234 bearing subjected to a pure axial external load, $F_{ax} = 10\,000$ N and the ball $j = 0$ is considered (Fig. 5–10).

3.2. Sliding speeds and film thickness as function of axial force

Both the internal geometry of the considered 4PCBB-13 structure, (deep groove ball bearing), and the operating conditions, ($F_r = 0$, $\omega_i = 60\,000$ rot/min) are similar to those used in [8]. The results provided by the presented model are presented in Figs. 11–13, being comparable with the results given in [8].

In addition, the model includes an analysis of the contact ellipse truncation when occurring. A complex kinematics and distribution of power losses has been found when the bearing operates with 3 or 4 contact points

Table 1

The bearings geometry

Parameter	4PCBB-13	4PCBB-1234
Number of balls, Z	16.000	20.000
Ball diameter, d_w [mm]	7.938	19.050
Pitch diameter, d_m, [mm]	50.000	149.000
Radial clearance, [μm]	79.000	150.000
Inner rings curvature factor, f_i	0.525	0.525
Outer rings curvature factor, f_e	0.510	0.510
Inner rings shim angle, [deg]	–	20.000
Outer rings shim angle, [deg]	–	30.000
Ball roughness, [μm]	0.150	0.150
Raceways roughness, [μm]	0.050	0.050

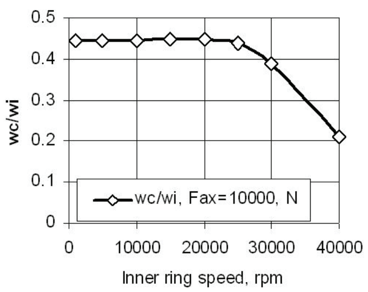

Fig. 5. The dependence of cage to inner ring angular speeds ratio on inner ring angular speed.

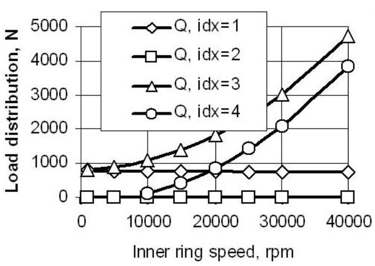

Fig. 6. The load distribution on ball j as function of the inner ring rotational speed.

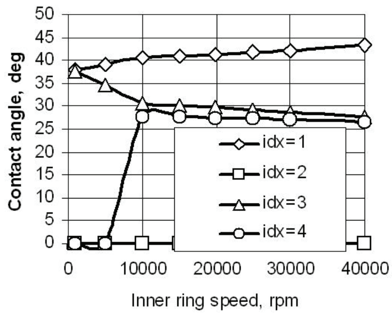

Fig. 7. The dependences of contact angles values on inner ring rotational speed.

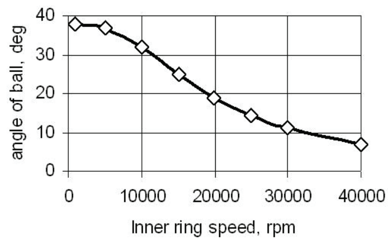

Fig. 8. The dependence of the angle that defines the direction of ball's angular speed vector on inner ring rotational speed.

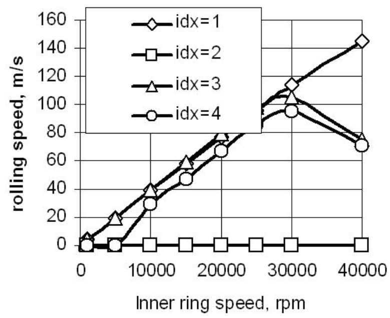

Fig. 9. The dependences of rolling speeds on inner ring rotational speed.

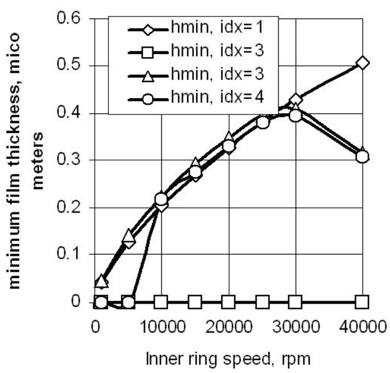

Fig. 10. The dependence of the minimum film thicknesses on inner ring rotational speed.

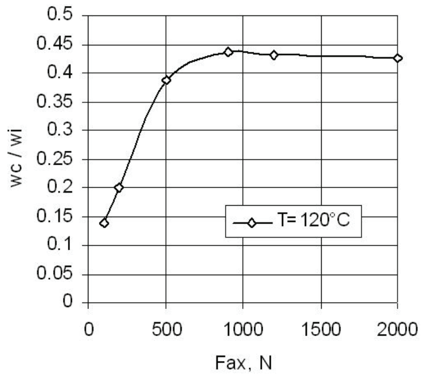

Fig. 11. The dependence of angular speeds ratio, cage to inner ring on axial load.

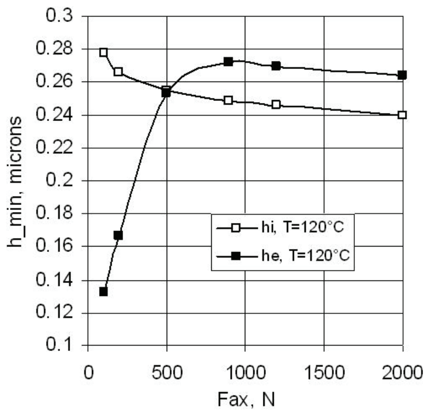

Fig. 12. The dependence of minimum film thickness on axial load.

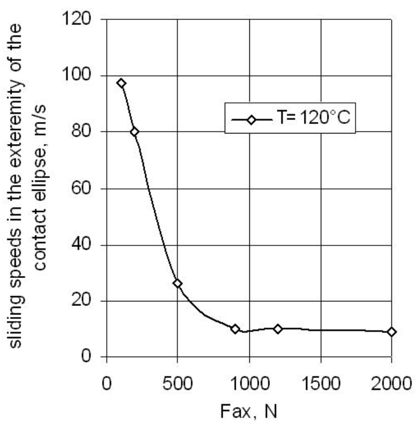

Fig. 13. The dependence of sliding speeds on axial force.

points, themselves depending on the composite surface roughness and lubricant rheological properties.

4. CONCLUSIONS

A unitary mathematical model, in five degrees of freedom, to describe the quasi-static and quasi-dynamic behavior of lubricated ball bearings, either with 2, 3 or 4 contact points, is presented.

The contribution of various interactions within bearing elements, in terms of normal and tangential loading and the principle of power minimization has been used to predict the ball and cage kinematics.

REFERENCES

[1] Bercea, I, Cretu, Sp., Nelias, D. (2003). *Analysis of Double-Row Tapered Roller Bearings*, *Part I – Model*, Tribology Transactions, **46**, *1*, pp. 491–498.

[2] Cretu, Sp., (2002). *Contact Mechanics*, “Gh. Asachi” Technical University, Iaşi, Romania.

[3] Hamrock, B. J., Dowson, D. (1977). *Isothermal Elastohydrodynamic Lubrication of Point Contacts*, Part IV, ASME Jour. of Lub. Technol., **99**, *1*, pp. 15–23.

[4] Hamrock, B. J., Anderson, W. J. (1973). *Analysis of an Arched Outer Race Ball Bearing Considering Centrifugal Forces*, ASME Jour. of Lub. Technol., **95**, *3*, pp. 265–276.

[5] Harris, T. A. (2001). *Rolling Bearing Analysis*, 4th Ed., John Wiley & Sons, New-York.

[6] Houpert, L. (1985). *Fast Numerical Calculation of EHD Sliding Traction Forces, Application to Rolling Bearing*, ASME, Jour. of Trib., **107**, pp. 234–240.

[7] Frene, J., Nicolas, N. (1990). *Lubrification hydrodynamique. Palier et Butéées*, Edit. Eyrolles, Paris.

[8] Nélias, D. (1989). *Etude du glissemenet dans les roulements a billes grande vitesses de turbomachine. – Influence de la pollution du lubrifiant*, Ph.D. Thesis, INSA de Lyon.

[9] Nélias, D., (1999), *Contribution à l'étude des roulements. Modélisation globale des roulements et avaries superficielles dans le contacts EHD pour des surfaces réelles ou indentées*, HdR, INSA de Lyon.

[10] Olaru, D. (2002). *Lubrication Fundamentals*, “Gh. Asachi” Technical University, Iaşi, Romania.

[11] Rezmires, D, Nelias, D. (2002). *Logiciel BB20, Convention de stage INSA de Lyon*, SNECMA Moteur.

[12] Yasutomi, S., Bair, S., Winer, W. O. (1984). *An Application of a Free Volume Model to Lubricant Rheology I – Dependence of Viscosity on Temperature and Pressure*, ASME, Jour. of Trib. **106**, *2*, pp. 291–303.

[13] Zhou, D., Cheng, S. (1988). *Effect of Surface Roughness on the Point Contact EHL*, ASME, Jour. of Trib., **110**, pp. 32–37.

[14] Rezmires, D., Racocea, C., Racocea, C. (2005). *Solution to desctibe the internal kinematics in ball bearings with 2, 3 or 4 contact points*, The annals of “Dunarea de Jos” Unniversity of Galați, pp. 107–111, Fascicle VII.

[15] Rezmires, D. (2003). *Theoretical and Experimental Research Regarding the Dynamics of Spherical Roller Bearings,* Ph.D. Thesis, http://daniel-rezmires.tripod.com

[16] Rezmires, D., Nelias., D, Racocea, C. (2004). *Hertzian and non-Hertzian Contact Analysis in Ball Bearings*, The Annals of University “Dunărea de Jos “ of Galați, pp. 105–108, Fascicle VIII, http://daniel-rezmires.tripod.com

Authors:

Cezar RACOCEA, Prof. Dr. Eng., “Gh. Asachi” Technical University of Iaşi, Dept. of Machine Design and Mechatronics, E-mail: cracocea@netscape.net

Daniel REZMIRES, Dr. Eng., S.C. SARCA. S.A., Piatra Neamț, E-mail: drezmir@yahoo.com