SIMULATION OF THE OIL REPLENISHMENT IN A BALL - RACE CONTACT

C.M.Oprisan[1], A.Tufescu[1], D.N.Olaru[1]
[1] "Gheorghe Asachi" Technical University, Iasi, ROMANIA, mariuca.oprisan@yahoo.com

Abstract: *A simulation program to determine the oil replenishment in the rolling path for a radial ball bearing has been realized by the authors. Based on the previous oil replenishment models for a ball race contact, the authors simulated the replenishment phenomena for three oil viscosities applied to a 6206 ball-race contact. Was evidenced the variation of the oil replenishment profiles both as function of the time and as function of the oil viscosity.*
Keywords: *oil replenishment model, ball-race contact, oil viscosity, Fourier transforms.*

1. INTRODUCTION

In oil lubricated ball bearings, the time between two successive passes of the balls on a point from the inner or outer race is depending on the ball bearing speed and on the bearing geometry. When a ball passes on a point from the race, the oil is ejected in the lateral sides and the replenishment of the track is governed, generally by the surface tension, the oil viscosity and the time. At normal rotational speed, in the time between two successive balls passes on the race, the oil replenishment of the track can assure the necessary oil layer to realize an EHD lubricant film in a ball-race contact, according to the existing formulas [1,2]. By increasing the speed, the time between two successive balls becomes smaller and the oil quantity obtained by replenishment is not sufficient to realize normal EHD film thickness and starvation phenomena appears[1, 2, 3].
Chiu [3] elaborated a sophisticated model for the oil replenishment of a track between a ball and a plane surface. Experimental results by optical interferometer confirmed his theoretical model.
Olaru [1,2] extended the theoretical oil replenishment model for a track realized between a ball and a ball bearing race. For small quantities of the lubricant in the lateral sides, the theoretical model developed by Olaru can be approximated by the Chiu's replenishment model.
In his PhD thesis, Olaru [2] applied this model for a 7206C angular contact ball bearing operating at high speed. Later, Olaru and Gafitanu [1] published the theoretical model and experimental results applied to 7206 C angular contact ball bearing. Both Chiu and Olaru used an approximation formula to obtain the variation of the oil layer in the center of the ball-race contact as function of the parameter $\frac{T}{2\eta}\cdot t$, where t is the time between two successive passing of the balls, T is the oil surface tension and η is the oil viscosity.
In the present paper, the authors realized a simulating program to follow the flow mechanism in the oil replenishment process. Was used the Chiu's model and obtained various replenishment profiles as function of the oil viscosity, lateral oil quality and time for the 6206 ball-outer race contact.

2. THEORETICAL MODEL

In Figure 1 is presented the geometrical parameters of the oil track realized by a ball in the outer race at initial time $t=0$.
In Figure 1 following geometrical parameters are used: R is the race radius, R_b is the ball radius, h_∞ is the oil layer on lateral of the ball-race contact, h_0 is the EHD central film thickness and $2a$ is the width of the race. By using a coordinate system (xOz) as is indicated in Figure 1, following equation can be written for the race curve:

$$f(x)=\begin{cases} h_\infty, \ if -a > x > a \\ \sqrt{R^2 - x^2} - \sqrt{R - a^2} + h_\infty, \ if -a \le x \le a \end{cases} \quad (1)$$

The oil shape generated by the ball at the time $t = 0$ is given by the function $g(x,0)$ defined by following equation:

$$g(x,0)=\begin{cases} 0, \ if -a^* > x > a^* \\ \sqrt{R_b^{\,2} - x^2} - \sqrt{R - a^{*2}} + R - R_b + h_\infty - h_0, \ if -a^* \le x \le a^* \end{cases} \quad (2)$$

The width of the oil track $2\,a^*$ is a function of the lateral oil layer defined by equation:

$$a^* = \sqrt{R_b^{\,2} - (\sqrt{R^2 - a^2} - h_\infty - R + R_b + h_0)^2} \quad (3)$$

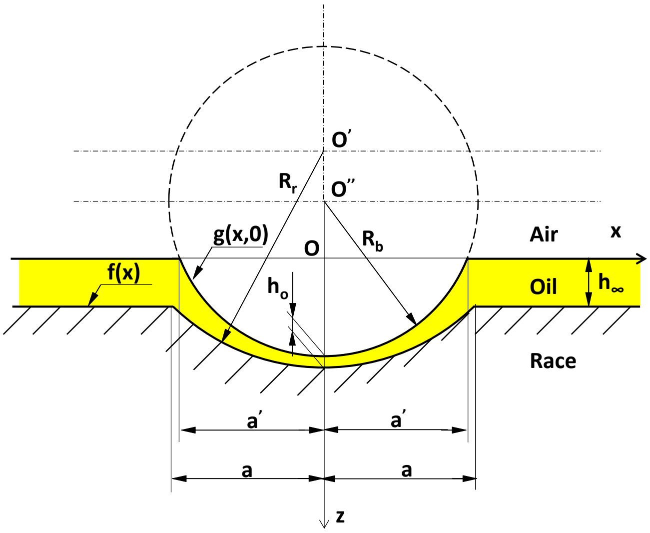

Figure 1: The geometrical elements and the oil layer in the ball-race contact

The flow of the oil in the two directions x and z is governed by following differential equations [1]:

$$\frac{\partial p}{\partial x} = \eta \cdot \left(\frac{\partial^2 v_x}{\partial x^2} + \frac{\partial^2 v_x}{\partial z^2} \right) \quad (4)$$

$$\frac{\partial p}{\partial z} = \eta \cdot \left(\frac{\partial^2 v_z}{\partial x^2} + \frac{\partial^2 v_z}{\partial z^2} \right) \quad (5)$$

and the continuity equation for incompressible fluid:

$$\frac{\partial v_x}{\partial x} + \frac{\partial v_z}{\partial z} = 0 \quad (6)$$

The Eqs. (4), (5) and (6) were solved in [1, 2] and the analytical relations for the two components of the oil speeds v_x, v_z and the variation of the oil pressure p have been obtained with the following limiting and initial conditions:

- On the race surface $f(x)$, the two components of the oil speed v_x and v_z are null, $v_x = v_z = 0$;
- On the free oil surface $g(x, t)$ the tangential stresses τ_{xz} are null and the pressure is given by Laplace equation:

$$p(x,t) = T \cdot \frac{\partial^2 g(x,t)}{\partial x^2} \quad (7)$$

Where T is the surface tension of the oil.

- For $t = 0$ $\quad p(x,0) = T \cdot \dfrac{\partial^2 g(x,0)}{\partial x^2}$

The variation of the oil shape $g(x,t)$ as function of the x and time (t) is obtained according to equation [1,2]:

$$g(x,t) = g(x,0) + \int_0^t v_z(x,t)dt \tag{8}$$

Eq.(8) was solved by Olaru[1,2] and for small lateral oil layers leads to the Chiu's equation [3]:

$$g(x,t) = a \cdot \int_0^\infty G(\xi) \cdot exp\left(\frac{-T}{2a^*\eta} \cdot k(\xi) \cdot t\right) \cdot cos\left(\frac{\xi \cdot x}{a}\right)d\xi \tag{9}$$

where

$$k(\xi) = \frac{1 - 4 \cdot \frac{\xi \cdot h_\infty}{a^*} \cdot exp\left(-2\xi \cdot \frac{h_\infty}{a^*}\right) - exp\left(-4\xi \cdot \frac{h_\infty}{a^*}\right)}{1 + exp\left(-2\xi \cdot \frac{h_\infty}{a^*}\right)} \tag{10}$$

$G(\xi)$ is the Fourier transform of the function *g(x,0)* given by the equation:

$$G(\xi) = \frac{2}{\pi \cdot a^2} \cdot \int_0^\infty g(x,0) \cdot cos\left(\frac{\xi \cdot x}{a}\right)dx \tag{11}$$

Chiu established from Eq. (9) a linear dependence for the variation of the oil rebounding thickness in the center of the contact *g(0,t)* and the time:

$$g(0,t) = C \cdot \frac{T}{2\eta} \cdot t \tag{12}$$

Where *C* is a dimensionless parameter depending only of the lateral oil layers, h_∞ [3].

The same linear dependence has been applied by Olaru [1,2] to evaluate the lubrication rebounding layer in 7206C high speed ball bearings considering the values of h_∞ as the oil lateral meniscuses in a ball-race contact.

3. SIMULATION OF THE OIL FLOW IN A BALL-RACE CONTACT

The analytical function of the oil layer at the time $t=0$ given by Eq.(2) has been used to simulated the replenishment process given by Eq.(9).

Eq.(9) has been numerical solved by a computer program for various oil viscosity as function of *x* and time *t* applied to the outer ball-race contact for 6206 radial ball bearing. The following geometrical parameters are used: ball diameter $D_b = 9.525\ mm$, race radius $R = 0.54D_b$ and race width $2a = 7\ mm$. The simulation was realized for a lateral oil layer $h_\infty = 1mm$ and the surface tension for oil was $T = 0.03\ Nm$.

In Figure 2 are presented the two functions *f(x)* and *g(x,0)* to be evidenced the interstices between the ball and the race. The elastic contact between the ball and race was neglected and it was considered that in the center of the race is a given film thickness h_0 between ball and race, having the following value: $h_0 = 0.001mm$.

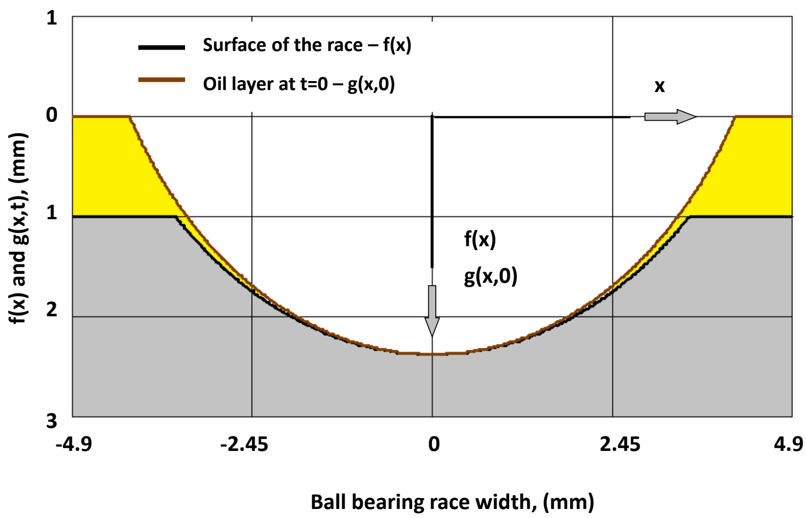

Figure 2: The layer of the oil on the race surface at initial time, t = 0

To integrate the Eq. (9) was necessary to establish the acceptable limit instead of infinite. Were tested various limits until to the values obtained from Eq. (9) for the initial time $t = 0$ corresponds to the initial layer of the oil $g(x,0)$ given by Eq. (2). In Figure 3 can be observed a very good agreement between the two functions $g(x,0)$ determined both by Eq. (2) and Eq.(9).

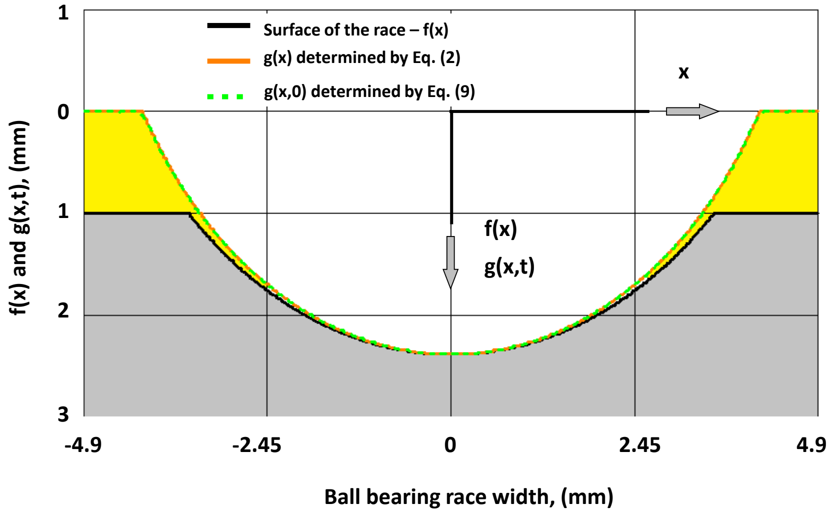

Figure 3: The correspondence between the values of the functions g(x,0) determined by the two Eqs.: (2) and (9)

3.1 Influence of the time in the oil flow model

To evidence the variation of the oil replenishment layers as function of the time, some simulation were realized with an oil having the dynamic viscosity $\eta = 0.02$ *Pas* for three values of the time: 0.001 seconds, 0.002 seconds and 0.01 seconds. The profiles of the replenishment oil layers are presented in Figure 4.

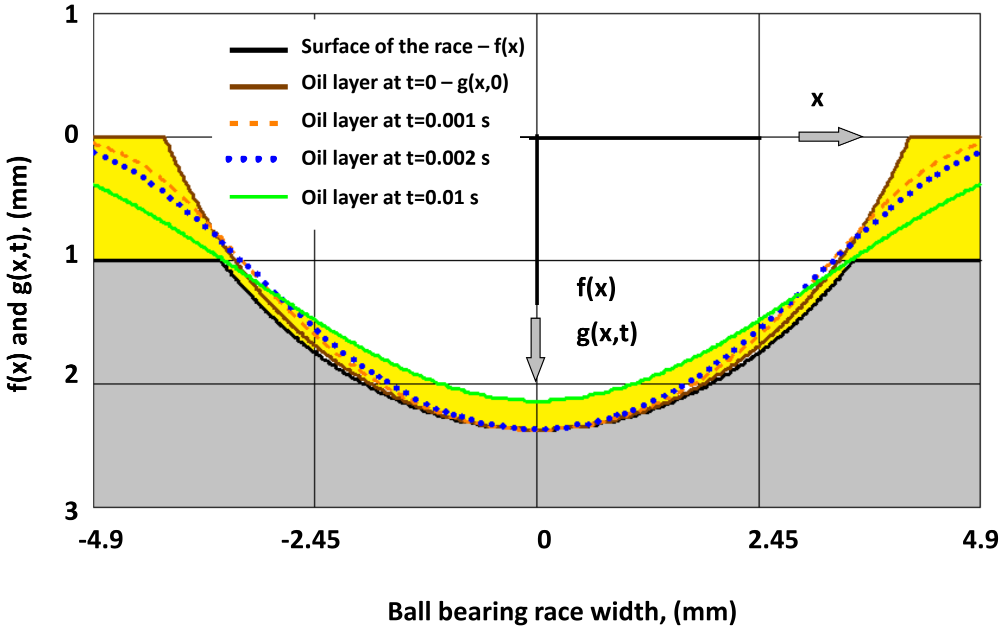

Figure 4: Variation of the oil replenishment layers for t = 0.001 seconds, t = 0.002 seconds and t = 0.01 seconds for an oil with dynamic viscosity η = 0.02 Pas

Comments

(i) It can be observed that by increasing of the time, the replenishment layer increases in thickness that suggests a good simulation of the real phenomena.

(ii) For the 6206 ball bearing, the time between two passings of a ball on the outer race is given by the following Eq.[4]:

$$t = \frac{120}{n \cdot \left(1 - \frac{D_b}{d_m}\right) \cdot Z} \quad (13)$$

where n is the rotational speed of the inner race (rot/min), d_m is the average diameter of the bearing and z is the number of the balls. For a standard 6206 ball bearing $d_m = 46\ mm$ and $z = 8$. By considering the limit speed of this ball bearing lubricated by oil ($n_{limit} = 13{,}000\ rot/min$ [5]), the time between two successive balls at this limit speed is about $t = 0.0015$ *seconds*. According to the Figure 4, at this very short time the replenishment oil layer is too thin to assure a good lubrication and the starvation phenomena can occurs. According to the SKF lubrication recommendation [5] for this rotational speed, the recommended viscosity is of 0.008 Pas. Also, for a rotational speed of the inner race of 1500 rot/min, the time between two successive balls is 0.01 and thick oil layer was developed at this time (See Figure 4).

3.2 Influence of the oil viscosity in the oil flow model

To evidence the influence of the oil viscosity on the replenishment layers three oil viscosities were used to simulate the oil flow: $\eta = 0.02$ *Pas*, $\eta = 0.05$ *Pas*, $\eta = 0.1$ *Pas*.
The profiles of the replenishment oil layers for a given time $t = 0.01$ seconds and for the three above mentioned viscosities are presented in Figure 5.

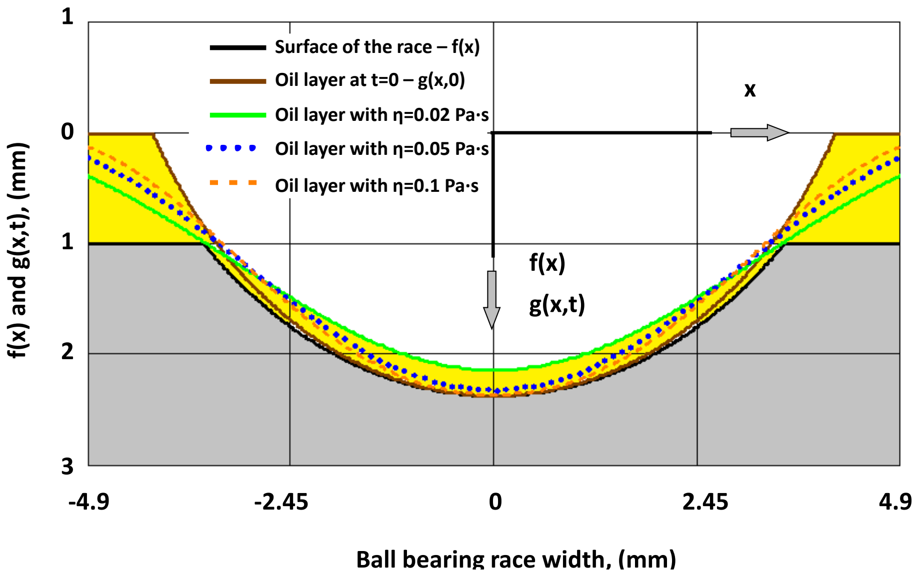

Figure 5: Variation of the oil replenishment layers for oil dynamic viscosities of η = 0.02 Pas, η = 0.05 Pas, η = 0.1 Pas and for the time t = 0.01 seconds

It can be observed that by increasing of the oil viscosity the replenishment layer decreases in thickness for the same simulated time. This behavior of the oil flow is in good correlation with the real phenomena.
The oil film thickness Δh in the center of the ball – race contact can be determined by the following equation:

$$\Delta h = g(0,t) - f(0) \quad (14)$$

Variations of the oil film thickness Δh as function of the time for the three oil viscosities are presented in the Figure 6.
In Figure 6-a can be observed that by decreasing of the viscosity the oil film thickness increases. At a very short time for all three viscosities it can be observed that the negative values of Δh appears as result of cavitations phenomenon as is presented in Figure 6-b.
The cavitations phenomena are developed for a very short time depending of the oil viscosity. The cavitations have been evidenced also by Chiu [3] in a ball- plane contact for a very short duration following the contact.

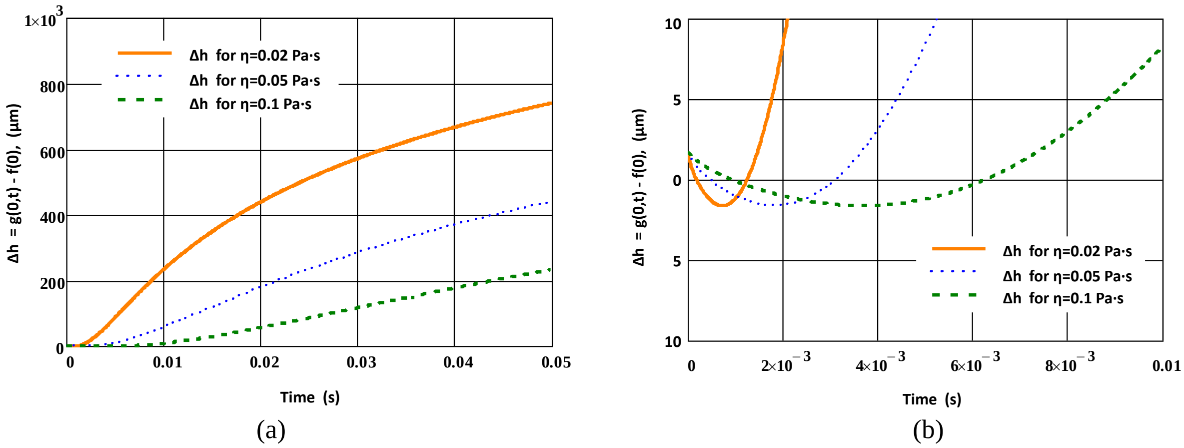

Figure 6: Variation of the oil film thickness Δh in the center of the ball – race contact for three oil viscosities: 0.02 Pas, 0.05 Pas, 0.1 Pas

4. CONCLUSIONS

A simulation program to evidence the oil replenishment process in a radial ball bearing race has been developed by the authors.

The simulation was realized by using the simplified Chiu's model applied to 6206 ball-outer race contact.

Various replenishment profiles as function of the time and oil viscosity for a given lateral oil layer have been obtained.

The simulated results evidenced a real behavior of the oil flow from the lateral sides to the center of the race for following oil viscosities: 0.02 Pas, 0.05 Pas and 0.1 Pas.

REFERENCES

[1] D.N. Olaru, M.D. Gafitanu, Starvation in Ball Bearings, Wear, Volume 170, Issue 2, 1993, 219–234.

[2] D.N.Olaru, Research regarding the increasing of the rotational speed of radial and angular contact ball bearings, PhD Thesis, Iasi, 1992 (In Romanian).

[3] Y.P.Chiu, An Analysis and Prediction of Lubricant Film Starvation in Rolling contact Systems, ASLE Transaction, Volume 17, 1, 22-35, 1974.

[4] T. Harris, Rolling Bearing Analysis, John Wiley & Soons, Ed.1991.

[5] www.skf.com