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Influence of Load and Design Conditions on Lubricant Film Thickness in the Deep Groove Ball Bearing

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Keywords： Ball bearing, Load distribution, EHL, Lubricant film thickness.

ABSTRACT

The study investigates the influence of the load distribution on lubricant film thickness between the rolling elements and raceways of the deep groove ball bearing using the conventional theory of elastohydrodynamic lubrication (EHL) with an applied original mathematical model of load distribution between the rolling elements developed by the authors. A presented mathematical model takes into account the influence of bearing operational load and internal geometry (number of balls and radial internal clearance) on lubricant film thickness between balls and raceways. The analyses show that the lubricant film thickness between the balls and raceways decreases with the decrease of the number of balls, a decrease of bearing load and increase of radial internal clearance, due to the increase of inequality of load distribution. The obtained analytical results can be useful for bearing design development as well as in analyses of lubrication conditions of rolling elements bearings in general.

INTRODUCTION

Rolling bearings are vital machine elements. In many applications, a machine's service life depends on the operational ability and reliability of the rolling bearings that are installed in it. To analyze, predict and prevent bearing failures, caused by different forms of damage, knowledge of actual lubricant film thickness in the bearing is necessary. According to Harnoy (2003), Elastohydrodynamic lubrication (EHL) is a dominant mechanism of lubrication in rolling bearings. In the mid-20th century, Grubin provided an approximate analytical solution to the EHL problem, which takes into account the contact elastic deflection effect and the viscosity-pressure effect in line contact, Harris e*t al*. (2006). Dowson and Higginson developed the mathematical model for determining lubricant film thickness in the line contact. That model, as well as the Hamrock-Dowson equation for lubricant film thickness in the point EHL contact, was derived in the latter half of the 20th century (1976 and 1977), and are still in use, Harnoy (2003), Harris *et al*. (2006), Harris (1984), Dowson (1998) and Szeri (1998). The lubrication mechanism of rolling elements in contact with the raceways was not mathematically described until the late 1940s and experimentally proved before the early 1960s, Harris *et al.* (2006). It has been relatively recently found that in rolling bearings a lubricant film can separate the rolling surfaces that are exposed to extremely high pressures in the contact zone. This phenomenon comes from the fact that viscosity is an exponential function of the pressure. Following his own analytical and experimental results, Tallian (1976) had shown that service life of rolling bearings operating under full EHL lubrication conditions is substantially extended, as well as wear resistance. Under conditions of high operating speeds and appropriate lubricant viscosity, a film thickness of several micrometres can be generated in rolling bearings. However, under real conditions, minimum film thickness does not exceed 1 μm, as it is shown by Harnoy (2003), Harris *et al*. (2006), Harris (1984), Dowson (1998). Furthermore, a lubricant film in contact areas between the balls and raceways may become thinner or be broken from various reasons. This can cause severe damages due to fatigue and intensive wear of contact surfaces, Oh *et al*. (2015). If lubricant is contaminated by hard, sharp particles larger than a film thickness the abrasive wear of the contact surfaces occurs, Halme *et al*. (2010). The film thickness change is also affected by the raceways macro geometry, i.e., waviness caused by the

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technological manufacturing procedure. Both experimental and theoretical research results presented in the paper of Ren *et al*. (2014) show that raceway waviness significantly affects bearing operating ability in terms of the lubricant film. Venner *et al*. (2012) have modelled a thin layer flow and film decay for grease-lubricated rolling bearings and have developed their model using the traditional method of determining load distribution in bearing.

This paper analyses the influence of load conditions on the EHL of deep groove ball bearing considering the change of lubricant film thickness. A model is developed using the conventional Hamrock-Dowson expression for lubricant film thickness extended by the original model of load distribution in bearing. The developed model enables the analysis of load distribution dominant factors (bearing internal geometry and load) influence on lubricant film thickness between the contact surfaces of the rolling elements.

LOAD DISTRIBUTION IN DEEP GROOVE BALL BEARING

The bearing load is transmitted from one ring to the other one through the rolling elements – balls (Figure 1). The distribution of radial load in a deep groove ball bearing is unequal, i.e the balls are not equally engaged in load transmission from one ring to the other one.

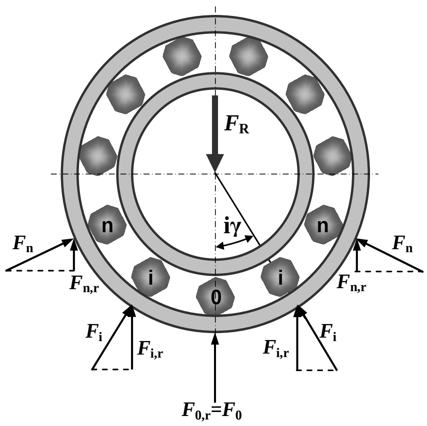


Fig. 1. Distribution of radial load in a deep groove ball bearing.

Inequality of bearing radial load distribution between the balls is affected by the intensity of load, bearing parts stiffness, raceways geometry, number of balls, radial internal clearance. To know the load distribution in the ball bearing for given operating conditions means to know the load for each ball. Due to the presence of more than one influence factor, as well as the complexity of their influence on load distribution in bearing, a necessity emerged to introduce an adequate quantity. By this quantity, the character and intensity of load distribution transmitted by each ball would be determined. Consequently, the load distribution factor is defined by Lazović (2001, 2003). It indicates the contribution of each ball in transmitting load. This factor can be used to analyse different influences on load distribution in bearing. So, if F_R is the bearing radial load (Fig. 1) and if i-ball transmits load F_i, then the F_i/F_R ratio indicates the engagement of i-ball in transmitting load. It was analyzed in detail in studies of Lazović *et al.* (2001, 2003, 2008, 2013, 2009, 2010). This force ratio represents the dimensionless load distribution parameter in bearing and is referred to as a load distribution factor:

$$k_{\mathrm{i}} = \frac{F_{\mathrm{i}}}{F_{\mathrm{R}}},\ \mathrm{i} = 0,1,...,\mathrm{n} \qquad (1)$$

where n = (z_l-1)/2 (Fig. 1).

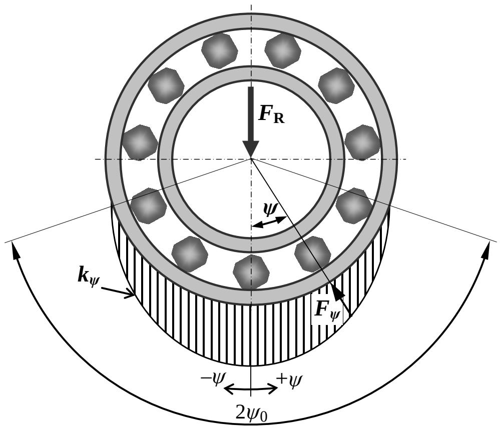


Fig. 2. Load distribution function in deep groove ball bearing.

Transforming discrete values of the load distribution factor determined by Equation (1) to a continuous function of the load distribution factor in the bearing loaded zone, the load distribution function can be obtained. The angular position of each ball in the bearing is determined by the product of its number and angular distance between the balls iγ (Fig. 1). If this product is substituted by the ball's current angular position ψ in the loaded zone (Figure 2) and after appropriate mathematical transformations, the function of load distribution, developed by Lazović *et al.* (2008, 2013, 2009, 2010) can be written in the form:

$$k_{\psi} = \frac{\left(1 - \left(1 + \frac{e}{2\delta_0}\right)(1 - \cos\psi)\right)^{3/2}}{\frac{z}{2\pi}\int_{-\psi_0}^{\psi_0}\left(1 - \left(1 + \frac{e}{2\delta_0}\right)(1 - \cos\psi)\right)^{3/2}\cos\psi \mathrm{d}\psi} \qquad (2)$$

where $e/2\delta_0$ is relative radial clearance, introduced by Lazović *et al.* (2008, 2013, 2009, 2010); $\delta_0 = \delta_{01} + \delta_{02}$ is total contact deflection between the most loaded 0-ball and raceways (Figures 1 and 3).

As Equation (2) indicates, with an increase of a total number of balls in bearing, the load distribution factor is decreased, because the load is shared between several balls in the loaded zone. The maximum possible width of the loaded zone of a radially loaded bearing is 180º. With the increase of the relative clearance, the loaded zone is getting narrow. The loaded zone size depends on the radial internal clearance and load intensity.

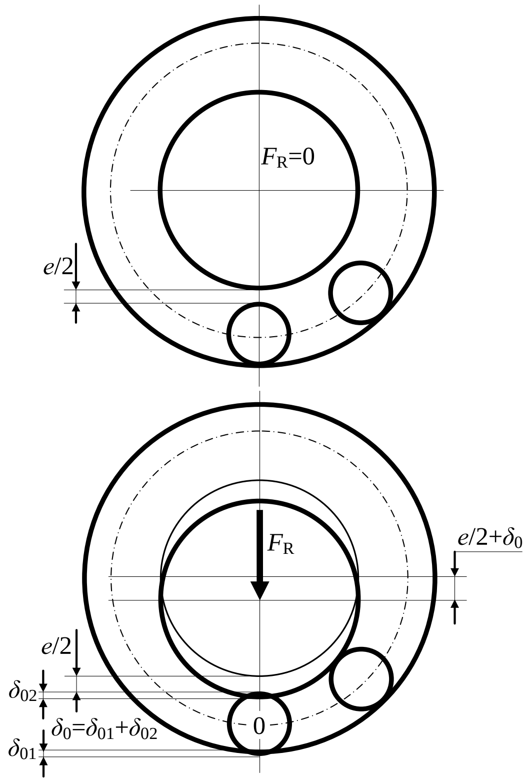


Fig. 3. Radial internal clearance and contact deflections of bearing parts.

The load distribution between the balls of the bearing loaded by the radial load is unequal. Inequality of load distribution is the lowest for the case of a bearing with zero radial internal clearance when the loaded zone is the widest $2\psi_0$=180°. The maximum load is then carried by the 0-ball and the minimum load by the balls on the ends of the loaded zone. With the increase of the radial internal clearance, the loaded zone is getting narrow ($2\psi_0$<180°), whereas unequal load distribution is increased. The most loaded 0-ball and adjacent balls are additionally loaded, and other balls are unloaded, partially or fully. Ball load at each point of the loaded zone, determined by angular coordinate ψ, for known bearing load F_R can be defined by the load distribution function:

$$F_\psi = k_\psi F_R \tag{3}$$

Bearing load F_R in Equation (3) can be represented as a function of bearing dynamic load rating, which is a constant characteristic of any bearing defined in ISO standard (1990):

$$F_R = k_C C \tag{4}$$

Respecting Equation (4), Eq. (3) for load in contact between the balls and raceways obtains the form:

$$F_\psi = k_\psi k_C C \tag{5}$$

or, according to Eq. (2):

$$F_\psi = \frac{\left(1-\left(1+\dfrac{e}{2\delta_0}\right)(1-\cos\psi)\right)^{3/2} k_C C}{\dfrac{z}{2\pi}\displaystyle\int_{-\psi_0}^{\psi_0}\left(1-\left(1+\dfrac{e}{2\delta_0}\right)(1-\cos\psi)\right)^{3/2}\cos\psi\,\mathrm{d}\psi} \tag{6}$$

Relative radial clearance $e/2\delta_0$ from Equation (6) depends on radial internal clearance and load. Based on numerical data from, relative radial clearance can be determined for four series of deep groove ball bearings with a bore diameter d = 30 mm (6006, 6206, 6306 и 6406). The results of the calculations are given in Table 1. The bearing load is the function of dynamic load rating through the relative load factor k_C according to Eq. (4). Radial internal clearance is within the limits of standard values for deep groove ball bearings with a bore diameter d = 30 mm, unassembled and under the zero measuring load. The assembled bearing clearance is smaller than the unassembled bearing clearance. This is caused by the reduction of internal clearance due to the shaft and housing fitting and thermal expansion of bearing rings in operation. The data from Table 1 shows that relative radial clearance increases as radial internal clearance increases in bearings of all sizes. Also, relative radial clearance decreases as bearing load increases, which is caused by smaller contact deflections of the most loaded 0-ball. Relative radial clearance is maximum ($e/2\delta_0$ = 1.646) for the case of the light bearing series (6006) with internal clearance larger than normal (e = 40 μm) and loaded with a very light load (5% of dynamic load rating).

The relative radial clearance dependence on bearing size, radial internal clearance and radial load is shown in Figure 4. The diagrams indicate negligibly small nonlinearity of relative radial clearance dependence on radial internal clearance. The slope of these approximately linear functions decreases as the bearing size (series) increases, and this is caused by the internal bearing geometry (balls and raceways radii).

Table 1. Relative radial clearance $e/2\delta_0$.

	k_C	e (μm) 0	10	20	30	40
6006	0.05	0	0.476	0.896	1.280	1.646
	0.20	0	0.200	0.385	0.558	0.726
	0.50	0	0.111	0.216	0.318	0.415
	1.00	0	0.070	0.139	0.205	0.269
6206	0.05	0	0.378	0.715	1.022	1.301
	0.20	0	0.156	0.304	0.445	0.581
	0.50	0	0.086	0.169	0.249	0.328
	1.00	0	0.054	0.107	0.160	0.211
6306	0.05	0	0.303	0.574	0.821	1.048
	0.20	0	0.124	0.243	0.357	0.466
	0.50	0	0.087	0.135	0.200	0.263
	1.00	0	0.054	0.086	0.127	0.169
6406	0.05	0	0.228	0.435	0.625	0.801
	0.20	0	0.093	0.183	0.269	0.352
	0.50	0	0.051	0.101	0.150	0.198
	1.00	0	0.032	0.064	0.096	0.127

Table 2. Radial internal clearance in deep groove ball bearings, according to ISO and DIN standards.

Bearing clearance classes		**Clearance (bore diameter d = 30 mm)**
C2	min	1
μm	max	11
Normal	min	5
μm	max	20
C3	min	13
μm	max	28
C4	min	23
μm	max	41

The influence of radial internal clearance can be considered and quantified through the relative radial clearance. The lubricant film thickness in the bearing will be considered in this way in the present paper. Equation (6) is a mathematical description of the influence of radial clearance, a total number of balls and the radial load on the load transmitted through balls from one ring to the other one. Based on known load distribution F_ψ, the film thickness distribution between the balls and raceways along the loaded zone can be determined.

LUBRICANT FILM THICKNESS

Under conditions of the contact between the hard surfaces, pure rolling and sufficient lubricating, according to the Hamrock-Dowson expression, from papers of Harnoy (2003), Harris *et al*. (2006), Harris (1984), Dowson (1998) and Szeri (1998), the film thickness at the centre of EHL contact (Figure 5) can be determined by the expression:

$$h = 2.69U^{0.67}G^{0.53}W^{-0.067}\left(1 - 0.61\mathrm{e}^{-0.73\kappa}\right)R_{\mathrm{eq}} \tag{7}$$

The contact between the ball and outer ring raceway is a contact between the convex surface of radius R_w and the concave surface of radius R_1 (Figure 6).

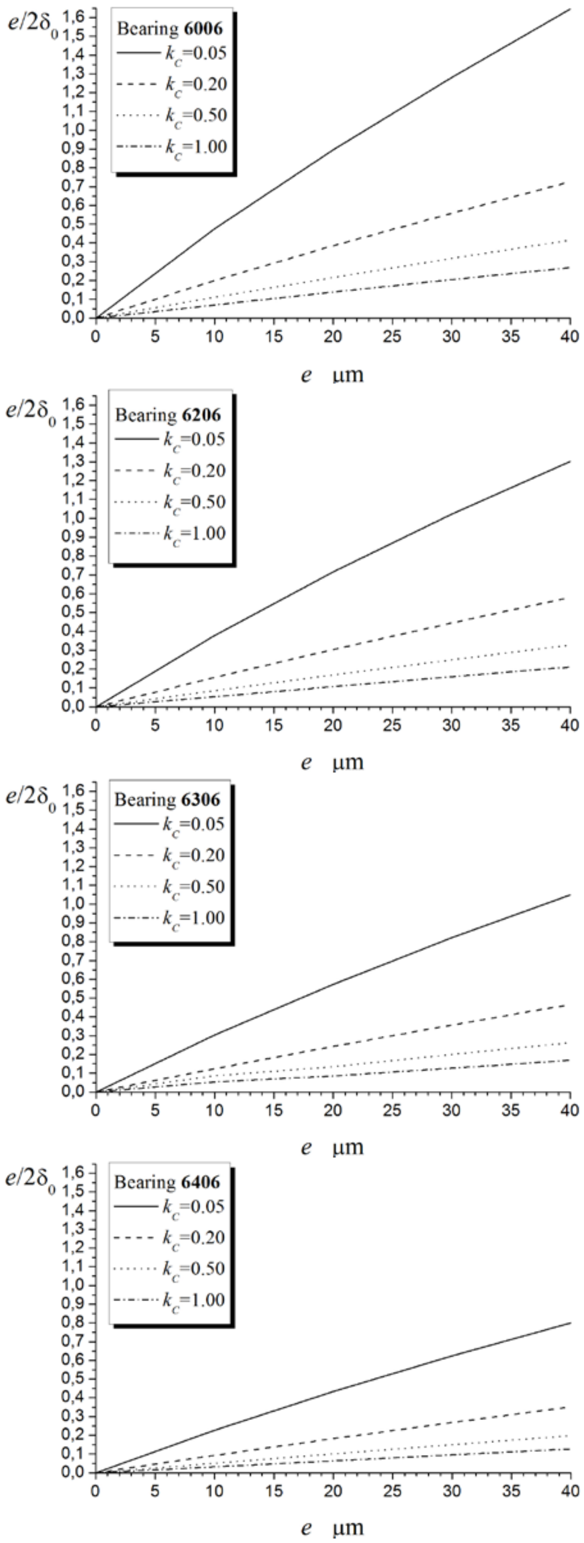


Fig. 4. Relative radial clearance.

The contact between the ball and the inner ring raceway is a contact between two convex surfaces of radii R_w and R_2. Equivalent curvature radii of the contact surfaces between the balls and raceways are, according to Harnoy (2003), Harris *et al*. (2006), Harris (1984) and Lazović (2013):

$$R_{\mathrm{eq,q}} = \frac{R_\mathrm{q} R_\mathrm{w}}{R_\mathrm{q} + (-1)^\mathrm{q} R_\mathrm{w}} \tag{8}$$

where q = 1 for the inner ring and q = 2 for the outer ring (Fig. 6).

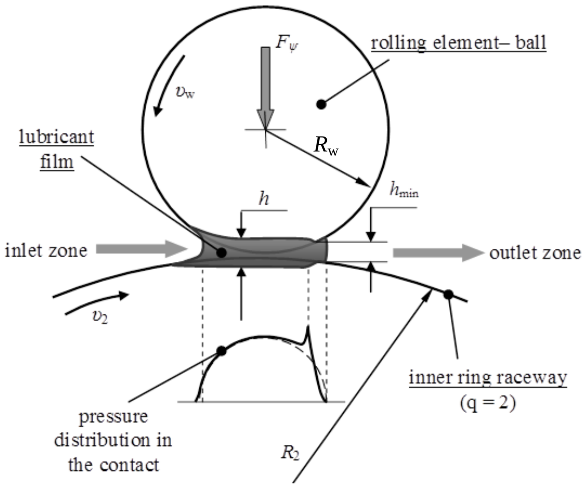


Fig. 5 Lubricant film in EHL contact between the ball and inner raceway

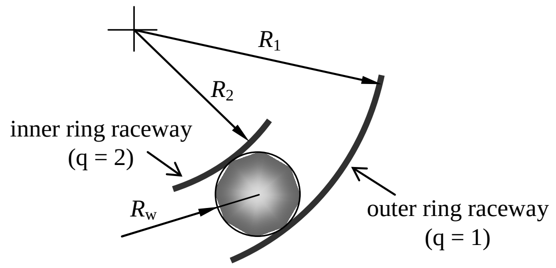


Fig. 6 Contact between the ball and raceways

Dimensionless material parameter G in Equation (7) includes the effect of lubricant material and bearing parts material on the film thickness:

$$G = \alpha \cdot E_\mathrm{eq} \tag{9}$$

where α is the pressure-viscosity coefficient, FAG (2002); E_eq is the equivalent elasticity modulus of the bearing parts.

The equivalent elasticity modulus in contact between the rings and balls is:

$$E_{\mathrm{eq,q}} = \frac{2 E_\mathrm{q} E_\mathrm{w}}{E_\mathrm{q}\left(1-\nu_\mathrm{w}^2\right) + E_\mathrm{w}\left(1-\nu_\mathrm{q}^2\right)} \tag{10}$$

If the balls and rings are made of the same material – steel, so that:

$$E_1 = E_2 = E_\mathrm{w} = E; \quad \nu_1 = \nu_2 = \nu_\mathrm{w} = \nu \tag{11}$$

For identical materials of bearing parts, Equation (10) obtains a simple form:

$$E_\mathrm{eq} = \frac{E}{1-\nu^2} \tag{12}$$

For steel material properties $E = 2.1 \cdot 10^{11}$ N/m^2 and $\nu = 0.3$, in accordance with Equation (11) equivalent elasticity modulus in EHL contact between balls and raceways is $E_\mathrm{eq} = 2.3 \cdot 10^{11}$ N/m^2.

The kinematic parameter of the lubricant film in the rolling bearing is a dimensionless speed parameter U in the expression for the lubricant film thickness given by Eq. (7). It takes into account the influence of relative velocities of the contact surfaces on the film thickness between them. This parameter is determined based on the expression:

$$U = \frac{\eta_0 \upsilon}{E_\mathrm{eq} R_\mathrm{eq}} \tag{13}$$

The dynamic viscosity at the atmospheric pressure is determined according to the expression:

$$\eta_0 = \nu_\mathrm{k} \rho \tag{14}$$

Velocity υ in Equation (13) is determined by the velocity of contact surfaces of the bearing parts in relative motion – balls and inner/outer ring raceways (Figure 7). For pure rolling, circumferential velocities at the contact point must be equal. For the case of a rotating inner ring ($\omega_2 > 0$) and stationary outer ring ($\omega_1 = 0$):

$$\begin{aligned} \upsilon_\mathrm{w} &= \upsilon_1 \Rightarrow \omega_\mathrm{w} R_\mathrm{w} = R_1 \omega_\mathrm{C} \\ \upsilon_\mathrm{w} &= \upsilon_2 \Rightarrow \omega_\mathrm{w} R_\mathrm{w} = R_2\left(\omega_2 - \omega_\mathrm{C}\right) \end{aligned} \tag{15}$$

Based on Equation (15) angular velocities are, Lazović (2013):

$$\omega_\mathrm{C} = \frac{1}{1+\dfrac{R_1}{R_2}} \omega_2 \tag{16}$$

$$\omega_\mathrm{w} = \frac{R_1}{R_\mathrm{w}} \cdot \frac{1}{1+\dfrac{R_1}{R_2}} \omega_2 \tag{17}$$

Based on Eqs. (15), (16) and (17), the rolling velocity of contact surfaces can be determined necessary for defining the dimensionless load parameter in Eq. (13):

$$\upsilon = \frac{R_1 R_2}{R_1 + R_2} \omega_2 \tag{18}$$

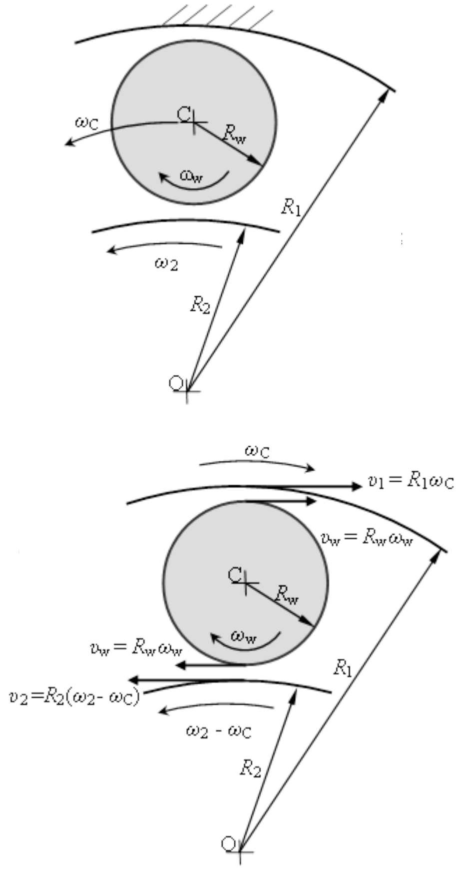


Fig. 7 Angular and tangential velocities of the ball and raceways

The dimensionless load parameter W in Eq. (7) is a lubricant film dynamic parameter:

$$W = \frac{F_\psi}{E_{eq} R_{eq}^2} \qquad (19)$$

Based on Equation (19), it can be inferred that load parameter W is proportional to the normal load in contact between the balls and raceways and that its value depends on load distribution in bearing.

INFLUENCE OF LOAD DISTRIBUTION ON LUBRICANT FILM THICKNESS

Load parameter W in Eq. (19) is proportional to the normal load in contact between balls and raceways so that its value depends on load distribution in ball bearing $F_\psi = F(\psi)$ determined by Eq. (6). Substituting Eqs. (9), (13) and (19), Eq. (7) obtains the form:

$$h_\psi = \frac{2.69\eta_0^{0.67}\upsilon^{0.67}\alpha^{0.53}E_{eq}^{-0.073}R_{eq}^{0.464}\left(1-0.61e^{-0.73\kappa}\right)}{F_\psi^{0.067}} \quad (20)$$

Based on Equation (20), it can be concluded that film thickness in EHL contact between the balls and raceways depends on: lubricant viscosity η_0; the velocity of balls and raceways' relative motion υ; the pressure-viscosity coefficient α; elastic properties of balls and bearing rings material E_{eq}; the geometry of balls and raceways R_{eq}; contact ellipse parameter κ (the ratio of the semi-major and semi-minor axes of contact ellipse).

Material parameters of lubricant and bearing parts, as well as bearing internal geometry and dynamic load rating have constant values for each observed rolling bearing, lubricated by a certain type of lubricant. A bearing load during bearing operation can be variable or constant, but its distribution between bearing balls is always unequal, more or less. If Equation (5) for the load for F_ψ in contact between balls and raceway at the loaded zone point with coordinate ψ is substituted in Eq. (20), the expression for lubricant film thickness can be written as:

$$h_\psi = \frac{2.69(\eta_0\upsilon)^{0.67}\alpha^{0.53}E_r^{-0.073}R_{eq}^{0.464}\left(1-0.61e^{-0.73\kappa}\right)}{\left(k_\psi k_C\right)^{0.067}C^{0.067}} \quad (21)$$

In Equation (21), the quantities invariable for the observed bearing can be grouped and substituted by a single constant:

$$h_\psi = C_h\left(k_\psi k_C\right)^{-0.067} \qquad (22)$$

where C_h is a film thickness constant.

Equation (22) can be also written as:

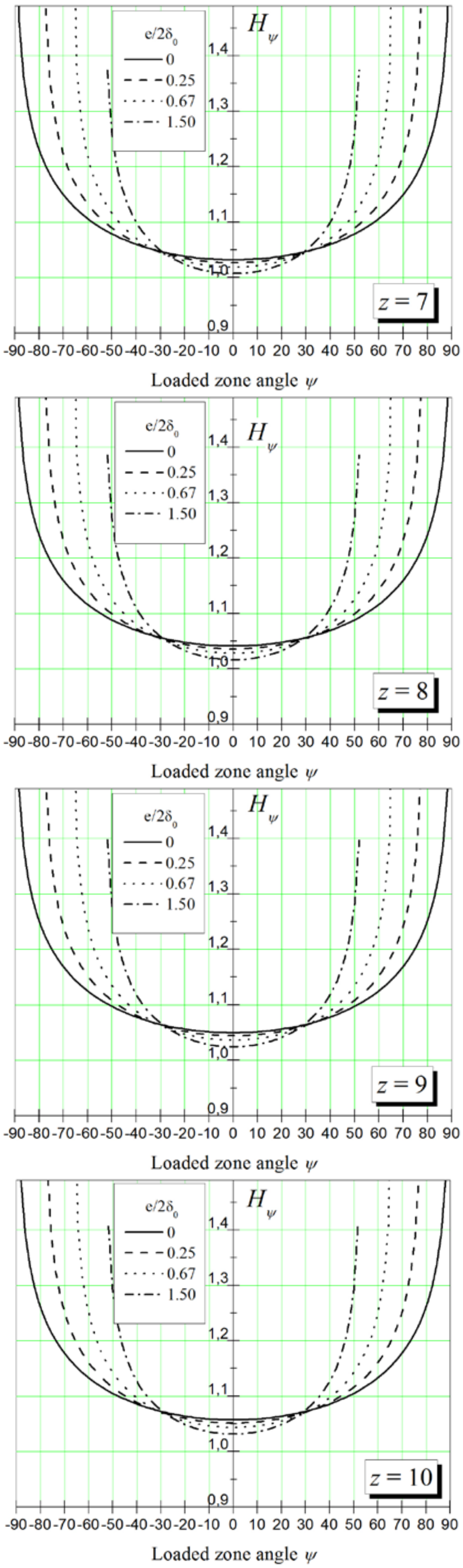
$$H_\psi = \frac{h_\psi}{C_h} = \left(k_\psi k_C\right)^{-0.067} \qquad (23)$$

where H_ψ is a dimensionless film thickness parameter as a function of load distribution in bearing.

Based on Eq. (22), it can be concluded that the film thickness in bearing depends on the bearing radial load and load distribution between the balls, under constant operating conditions (velocity temperature, lubricant viscosity). The lubricant film thickness parameter H_ψ depending on angle ψ of the loaded zone (ball position in the loaded zone) and relative radial clearance $e/2\delta_0$ of bearings with a different number of balls is shown by diagrams in Figure 8. Relative radial clearance is within the limits $e/2\delta_0$ = 0...1.5, according to Table 2 and Fig. 4. Bearing load equals dynamic load rating ($F_R = C$), i.e., the relative load factor is $k_C = 1$.

Based on diagrams in Fig. 8, the following can be concluded:

- as the number of balls in bearing increases, film thickness increases, due to contact load reduction resulting from reduced inequality of load distribution;



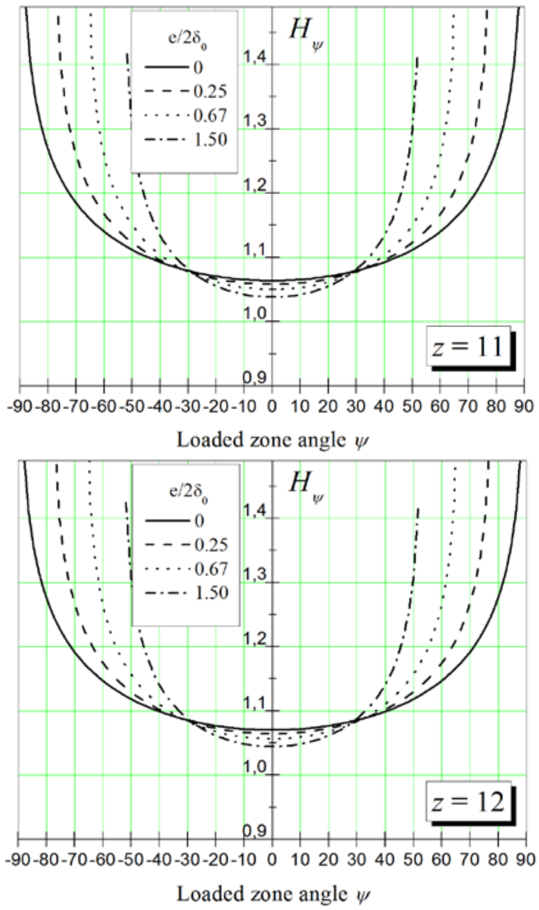
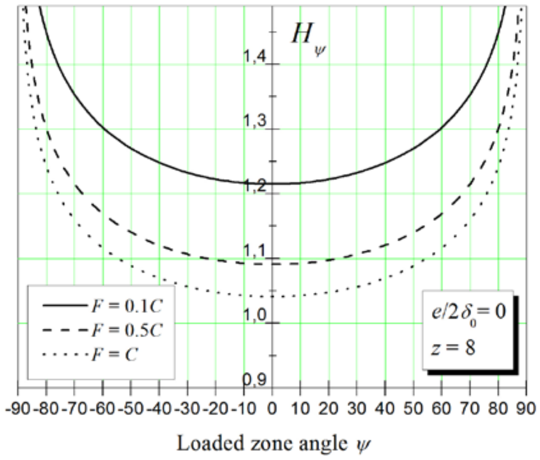


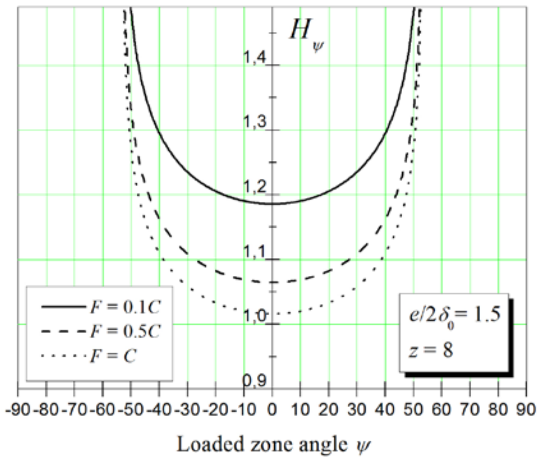
Fig. 8 Film thickness parameter depending on internal radial clearance and number of balls

- the minimum film thickness is at the spot of the most loaded 0-ball ($\psi = 0$) and is increased as the loaded zone angle increases;
- as radial internal clearance increases, inequality of load distribution increases, the film thickness increases or decreases, depending on ball position in the loaded zone;
- balls on the ends of the loaded zone ($\psi = \psi_0$) become unloaded with increasing radial internal clearance, which causes film thickness increase;
- as radial internal clearance increases, balls with angular position $\psi \approx \pm(25...35)°$ are additionally loaded, and therefore film thickness decreases;
- in the loaded zone part $\psi \approx \pm(25...35)°$ the gradient of change in film thickness is relatively small;
- as the loaded zone angle exceeds $\psi \approx \pm(25...35)°$, the gradient of change in film thickness is increased.

Diagrams of the influence of bearing load on the film thickness parameter for the example of ball bearing with z = 8 balls are shown in Figure 9. Considerations involve two extreme cases of the values for relative radial clearance: $e/2\delta_0$=0 (Fig. 9a) and $e/2\delta_0$=1.5 (Fig. 9b). As the bearing load increases, the film thickness decreases. The gradient of change in film thickness is also higher at lower loads for the case of a ball bearing with positive radial clearance. So, the bearing with radial internal clearance is more sensitive to the load influence on film thickness than the bearing that has a zero clearance. The analysis indicates that the film thickness between the balls and raceways changes along the loaded zone and that it depends on load distribution. Besides, the film thickness in balls' contact with outer raceway differs from film thickness in balls' contact with the inner raceway. This results from different values of the contact ellipses parameters κ_1 and κ_2 and equivalent curvature radii of the contact surfaces between the balls and raceways $R_{eq,1}$ and $R_{eq,2}$, determined by the Equation (8).



a)



b)

Fig. 9 Influence of bearing load on film thickness parameter in the bearing with z = 8 balls at a) $e/2\delta_0 = 0$ and b) $e/2\delta_0 = 1.5$

The film thickness in the ball's contact with the q-ring raceway can be as follows:

$$h_{\psi q} = C_{hq}\left(k_\psi k_C\right)^{-0.067} \tag{24}$$

A NUMERICAL EXAMPLE

According to Eqs. (21) and (24), a constant of film thickness can be determined by the expression:

$$C_{hq} = \frac{2.69\left(\eta_0 \upsilon\right)^{0.67} \alpha^{0.53} E_{\mathrm{r}}^{-0.073} R_{\mathrm{eq}}^{0.464}\left(1-0.61\mathrm{e}^{-0.73\kappa}\right)}{C^{0.067}} \tag{25}$$

Data on bearing parts material and geometrical properties, lubricant (mineral oil) properties, the rotation frequency of bearing as well as dynamic load rating for deep groove ball bearings 6006, 6206, 6306 and 6406, are given in Table 3. The film thickness constants can be calculated from Equation (25).

Table 3. Data for determining film thickness constants in deep groove ball bearings, Lazović (2013).

	Quant.	Unit	Deep groove ball bearing 6006	6206	6306	6406
Lubricant	ν_k	m²/s	$85\cdot10^{-6}$			
	ρ	kg/m³	880			
	η_0	kg/ms	0,075			
	α	m²/N	$2.5\cdot10^{-8}$			
Geometry	R_1	mm	24.822	27.762	32.152	38.334
	R_2	mm	17.678	18.238	19.848	21.666
	R_w	mm	3.572	4.762	6.152	8.334
	R_{eq1}	mm	4.172	5.748	7.608	10.649
	R_{eq2}	mm	2.972	3.776	4.696	6.019
Material	E	N/m²	$2.1\cdot10^{11}$			
	ν	-	0.3			
	E_{eq}	N/m²	$2.3\cdot10^{11}$			
Speed	n	min⁻¹	4252			
	v	m/s	4.6	4.9	5.5	6.2
Contact ellipse	κ_1	-	4.9	5.6	5.6	7.7
	κ_2	-	14.1	13.8	14.0	10.7
Dynamic load rating	C	N	13300	19500	28100	43600

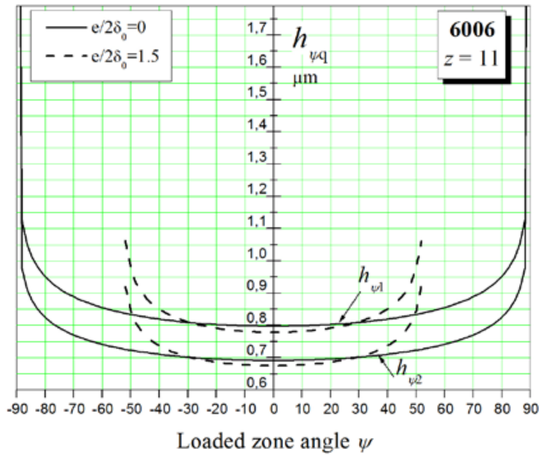
The constants of film thickness in balls' contact with inner and outer ring raceways for bearings 6006, 6206, 6306 and 6406, lubricated with mineral oil and the inner ring rotation speed of 4252 min^{-1} (rotation speed of the device used for corresponding tests), is given in Table 4.

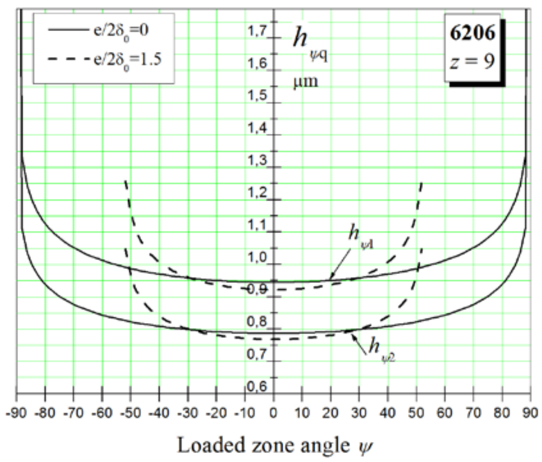
Table 4. Film thickness constants.

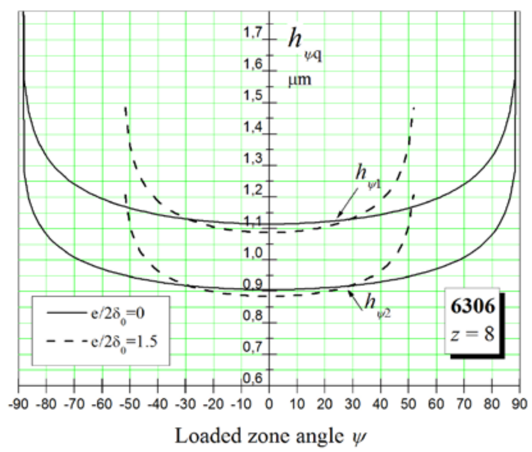
Ring	C_{hq}	Unit	Deep groove ball bearing 6006	6206	6306	6406
Outer (q=1)	C_{h1}	μm	0.75	0.90	1.07	1.33
Inner (q=2)	C_{h2}	μm	0.65	0.75	0.87	1.03

Based on data from Table 4, it can be concluded that bearings, belonging to different series, have different values of film thickness constants. The 'heavier' the bearing series is, the higher the values of film thickness constant are. Consequently, in considered bearings, the film thickness will not be the same under the same operating conditions. The film thickness between the balls and raceways, depending on ball position in the loaded zone and radial internal clearance of bearings 6006, 6206, 6306 and 6406 is shown in Figure 10. These diagrams were formed using the Eqs. (2) and (24) and data from Table 4.

It is assumed that bearing load is equal to bearing dynamic load rating ($F_R = C$), i.e., relative load in Equation (24) is $k_C = 1$. The analysis was carried out for zero clearance ($e/2\delta_0 = 0$) and the maximum clearance ($e/2\delta_0 = 1.5$).







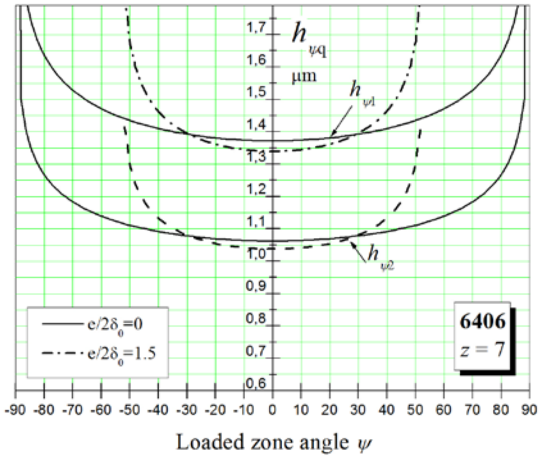


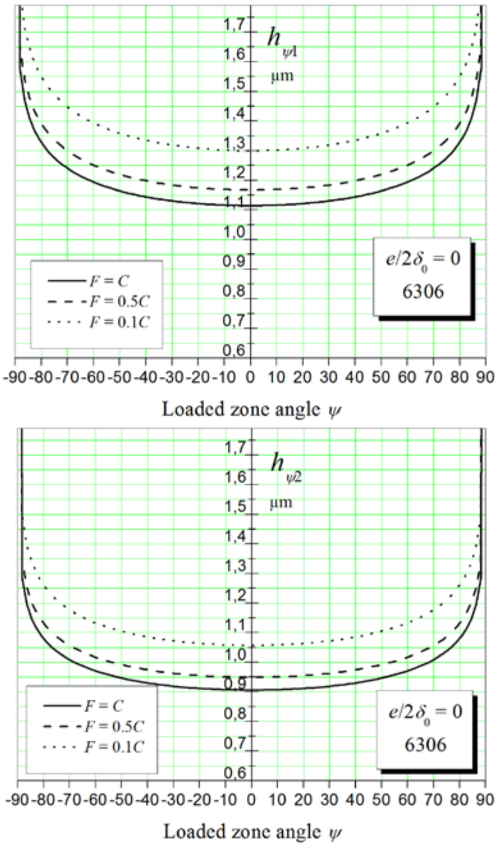
Fig. 10 Film thickness between the balls and raceways in bearings 6006, 6206, 6306 and 6406 for $F_R = C$ ($h_{\psi 1}$ – outer raceway, $h_{\psi 2}$ – inner raceway)

Based on diagrams of film thickness in balls' contact with raceways, for described operating conditions, it can be concluded (Fig. 10):

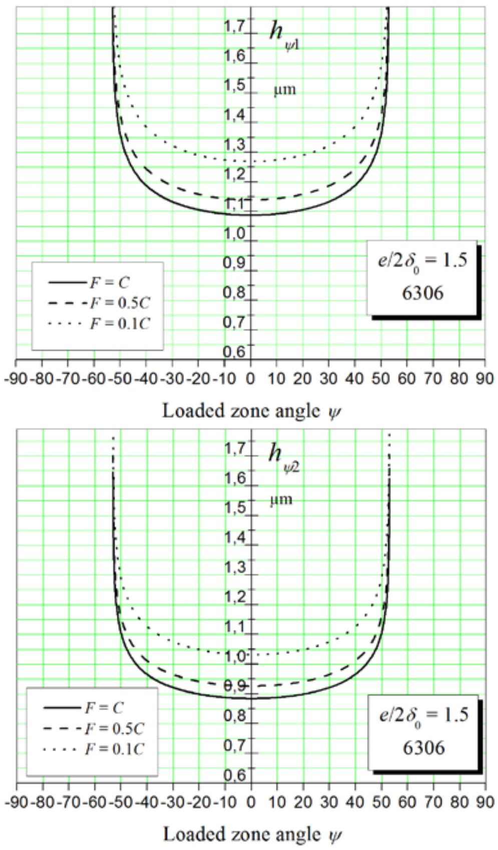
- the film thickness changes along the loaded zone;
- the gradient of change in film thickness along the loaded zone is relatively low;
- the film thickness is the smallest of the most loaded 0-ball ($\psi = 0$);
- at the ends of the loaded zone, a rapid rise of film thickness occurs, caused by zero-load;
- as radial internal clearance increases, film thickness slightly decreases, the influence of radial clearance on film thickness is small;
- the film thickness in balls' contact with outer raceway ($h_{\psi 1}$) is larger than film thickness in balls' contact with inner raceway ($h_{\psi 2}$) due to the different geometry of contact;
- the difference between film thickness in balls' contact with outer and inner raceways is the smallest in bearing 6006 ('light' series) and the largest in bearing 6406 ('heavy' series), which is the result of different relationships between film thickness geometry parameters;
- the film thickness is the smallest in bearing 6006 and the largest in bearing 6406, resulting from different values of film thickness constants C_{hq}.

Diagrams of film thickness in Fig. 10 correspond to the bearing load equal to dynamic load rating ($F_R = C$) when the relative load in Eq. (24) has the value $k_C = 1$. The influence of the bearing load on the film thickness is shown in Figure 11.

The analysis was carried out for bearing load: 10% ($k_C = 0.1$) and 50% ($k_C = 0.5$) of dynamic load rating. Considerations included two extreme cases of relative radial clearance values: $e/2\delta_0$=0 (Fig. 11a) and $e/2\delta_0$=1.5 (Fig. 11b).



a) $e/2\delta_0 = 0$



b) $e/2\delta_0 = 1.5$

Fig. 11 Influence of bearing load on film thickness between the balls and raceways (6306)

With increasing bearing load, film thickness decreases on both raceways. The gradient of film thickness decrease is higher in the balls' contact with the outer raceway ($h_{\psi 1}$). So, lubricant film between the balls and outer raceway is more sensitive to the change in bearing load. This is caused by the geometry (dimensions, shape) of the contact surfaces.

DISCUSSION

This paper analyses the influence of load conditions on film thickness in the centre and a major part of the contact surface between the balls and raceways. Lazović (2013) indicates that the ratio of the minimum film thickness and the film thickness in the rest part of the EHL contact is $h_{\psi\min}/h_{\psi} \approx 0.8$, irrespective of the bearing series and operating conditions. Therefore, to get the distribution of minimum film thickness in deep groove ball bearing, the values in the diagrams from Figs. 10 and 11 should be reduced by 20%. The mathematical model of film thickness represented by the Eq. (21) has been developed for the case of oil lubrication. In the case of grease lubrication, the required parameters of lubricant rheological properties are determined for the base oil of applied grease. Depending on operating conditions (rotation speed, type of lubricant, temperature), the film thickness in grease-lubricated bearing, determined by the Eq. (21) can be corrected. The recommended correction factors are the result of experimental research. According to research results obtained by Farcas *et al.* (1999), the ratio of grease film thickness and the film thickness of corresponding base oil is $h_{\psi,\ \mathrm{grease}}/h_{\psi,\ \mathrm{oil}} \approx 0.7$. So, in bearing lubricated with grease, the film thickness in the diagrams from Figs. 10 and 11 should be additionally reduced by 30%. Film thicknesses in these diagrams correspond to a relatively high bearing rotation speed (over 4000 rpm). As rotation speed decreases, film thickness decreases too, as it is shown in papers of Harnoy (2003) and Cann *et al.* (2004).

The minimum film thickness, analytically obtained, may be of the order of one or a few tenths up to 1 micrometre, under identical operating conditions (contact surfaces geometry and material properties, lubricant type, contact surfaces relative velocity, bearing load). This corresponds to the results of experimental research and data reported in the publications of Hamrock *et al*. (1977), Harnoy (2003), Dowson (1998), as well as in recently published results of investigations conducted by Zhou *et al*. (2019).

Statistical data in the paper given by Dwyer-Joyce (1998) indicate that the loss of mass due to wear is larger on the rotating inner ring raceway (21% of bearing the total loss of mass) than on the stationary outer ring raceway (1%). The reason is that wear has a longer way to take, but also a smaller film thickness between the balls and inner raceway, found by calculations and analyses carried out in the present study (Figs. 11 and 12).

A mathematical model for the film thickness analysis in ball bearing, from the viewpoint of load conditions, has been developed for the need of more accurate evaluation of the lubricating conditions in bearing, which determine other mechanisms in bearing (wear of contact surfaces, heat generation, and transfer, vibrations, noise, fatigue). Knowledge about lubricant film thickness is important for improving lubricant properties according to operating conditions, proper choice of lubricant, as well as for determining optimum lubricant amount in bearing. To have a developed mathematical model of film thickness, which takes into account load distribution conditions in bearing, is important because of the possibility to determine the friction forces and power loss in rolling contacts, optimization of bearing parts' internal geometry, dynamic analyses to reduce the levels of vibrations and noise generated in bearing. Besides, the surface finish quality of contact surfaces in bearing (roughness) must be correlated with film thickness. If the film thickness is larger than irregularities on the contact surfaces, it then completely separates the rolling surfaces, thus eliminating wear caused by direct dry contact. Therefore, for the optimal conditions, minimum film thickness must be larger than surface roughness. Also, if the bearing operates in the contaminated environment and if it is known that the size of abrasive particles that may enter the bearing is larger than the film thickness, the probability of abrasive wear occurrence and bearing failure is large.

CONCLUSION

The lubricant film thickness in deep groove ball bearing depends on lubricant viscosity, bearing internal geometry (balls and raceways' shape and dimensions, number of balls, radial internal clearance) and operating conditions (load, speed, temperature, the contamination of the operating environment). In this study, a mathematical model has been developed and used to consider the influence of load conditions and bearing internal geometry on the lubricating film thickness. Based on obtained results it can be concluded:

- as bearing load increases, the film thickness decreases by approx. 17% on both inner and outer raceway, independently of radial internal clearance;
- light-loaded bearings with a larger radial internal clearance are more sensitive to the change in load distribution, and therefore the change in film thickness is more expressed in them;
- in the same ball bearing, under the same operating conditions, the film thickness is smaller in contact between the balls and inner ring raceway than between the balls and outer ring;
- depending on the bearing series size, the difference in film thickness between the inner and outer raceway can be 15%...30% and the probability of oil starvation, damage and wear of inner ring raceway contact surfaces is larger;
- radial internal clearance has the small influence on the film thickness (regardless of a bearing series, with clearance increase, the film thickness can be reduced by 2.5% on both inner and outer raceway, in bearings of all series).

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NOMENCLATURE

e	Radial internal clearance in bearing
E_q, E_w	Elasticity moduli of the rings and balls
G	Material parameter
k_C	Relative load factor
R_1, R_2	Radii of the inner and outer raceway
R_{eq}	Equivalent curvature radius of the contact surfaces
R_w	Radius of the ball
U	Speed parameter
W	Load parameter
z	Number of balls in bearing
z_l	Number of balls in the loaded zone, i.e., the number of balls engaged in load transmitting
δ_{01}, δ_{02}	Local deflections in contact with the 0-ball with the outer and inner raceway
η_0	Dynamic viscosity of the lubricant at the bearing operating temperature and atmospheric pressure
κ	Ellipticity ratio (the ratio of the semi-major and semi-minor axes of contact ellipse)
ν_q, ν_w	Poisson’s ratios of the rings and balls
ν_κ	kinematic viscosity
ρ	Lubricant density
υ	Mean velocity of the rolling contact surfaces
υ_1,υ_2	Tangential velocities of the inner and outer raceway
υ_w	Tangential velocity of the ball’s rotation about its own axis
ψ	Angular position of ball
ψ_0	Half of the loaded zone width
ω_2	Angular velocity of the outer ring
ω_C	Angular velocity of the ball’s rotation about the bearing axis
ω_w	Angular velocity of the ball’s rotation