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COMPARATIVE ASPECTS BETWEEN EHD LUBRICATION OF GEARS, BEARINGS AND RADIAL JOURNAL BEARINGS UNDER HUGE LOADING

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Abstract: This paper presents comparative aspects between EHD lubrication of gears, bearings and radial journal bearings with HD lubrication in the case of huge challenging working. It is estimated the possibility of the EHD lubrication barely appearance in the case of radial journal bearings, similar situation can be met with cylindric gears and bearings with rolls.

Key words: *bearing, elastohydrodynamic lubrication, elastic deforming, pin and bush.*

1. INTRODUCTION

The working of the manufactural engines with liniar or punctiform contact (gears, bearings and other couples with hertziene's rubbing) under huge charging takes place in good lubrication and anti-usage conditions because of a thin lubrifiant film in the contact zone. Researches have demonstrated that the contact zone is suffering an elastic distorsion owing to the huge charging which brings about a bigger interstice with a thin lubrifiant film, similar situation with the sliding bearings [6]. The elastic deforming of the contact zone surfaces is associated with a rapid increase to 15 times of the viscousity with pressure [3].

Figure 1 shows the deformed zone of a cylindric gear's flanks, as well as the hertziene's pressure distribution with top pressure in the exit point, which brings about interstice distorsion forming a threshold where the lubrifiant film thickness is h_0=0,8 h; as the value of the speed is increasing ($V_3 > V_2 > V_1$), the top pressure is moving towards the entrance point [6].

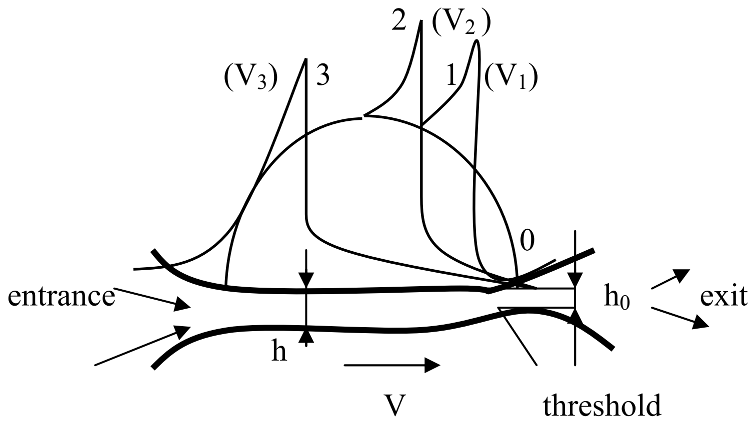

Fig. 1 The EHD interstice of a gear and the distribution of the proper pressure [6]

The first result regarding the thickness of the lubrifiant film in EHD conditions were obtained by Grubin (1949) and Dowson (1966) and includes 4 adimensional paramethers: H_{min} for the lubrifiant film, W_0 for the charging, V for the speed and G_0 for materials [7].

Thus, the Dowson-Higginson relation for rolling with sliding that can be applied to gears is [6]:

$$H_{min} = 0{,}96\ G_0^{0,6}\ U^{0,7}\ W_0^{-0,13}\ , \qquad (1)$$

where H_{min}= h_{min}/R, and R is the curving-ray of the contact in m, („-“ is the sign for interior contact): $1/R = 1/R_1 \pm 1/R_2$.

In the case of pure rolling, for example the case of bearings, Dowson-Higginson relation is [7]:

$$H_{min} = 1{,}6\ G_0^{0,6}\ U^{0,7}\ W_0^{-0,13} \qquad (2)$$

Dowson şi Higginson [5], consider that, as for huge charging, the hydrodynamic pressure distribution is getting near to Hertz's values, differences are to be noticed only in the entrance and the exit points, elastic displaces in the contact zone being comparable to those owing to hertziene's pressure.

The h_{min} lubrifiant conditions is imposed by rugosity, which interferes with specific weight; for rugosities with R_a >1 μm of both flanks, the lubrifiant conditions become EHD partial [8].

The viscousity increase because of pressure and the elastic distorsion of the surfaces that compete with each other in the process of keeping lubrifiant in the contact zone, leads to films building thicker than those obtained through HD clasic treatment.

In figure 2 [6], it is showed the subordination of the lubrifiant film thickness depending on the speed in case of bearings with rolls.

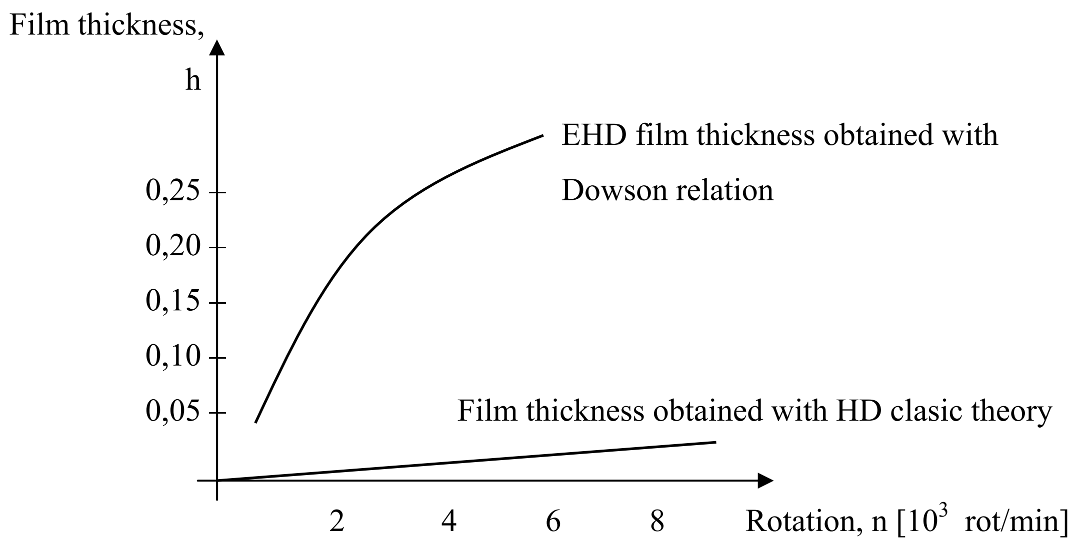

Fig. 2 The subordination of the lubrifiant film thickness depending on the speed in EHD conditions and clasic HD theory [6]

In the case of gears, the lubricating working conditions vary from the entrance point to the exit point, as a result of the changes suffered by the load, of the speed rolling and sliding, of the curvature-ray. Depending on the cinematic and tribological parameters, the working condition can be HD, "critic" HD, and EHD [9].

Taking into account the lubrifiant adimensional parameters, X_u, the elasticity, X_e, the speed, X_v, and the load X_p, with the following expressions [6]:

$$X_u = \frac{\alpha \cdot W}{R_{w_1} \cdot y_\rho}\sqrt{\frac{W}{\eta_0 \omega_1 R_{w_1} y_v}}\ ;\ X_e = \frac{W}{R_{w_1} \cdot \sqrt{E \cdot \eta_0 \cdot \omega_1 \cdot y_\rho \cdot y_v}}\ ;$$

$$X_v = \alpha \cdot \left(E^3 \frac{\eta_0 \omega_1 y_v}{y_\rho} \right)^{1/4} \quad X_p = \alpha \cdot \left(\frac{W \cdot E}{2\pi \cdot R_{w_1} y_\rho} \right)^{1/2}, \tag{3}$$

where α is the variation parameter of the viscousity with the pressure, W is the normal load on the tooth length corresponding to the curent point, y_ρ and y_v are the curvature-ray functions and in the same time the total speed value, the following lubricated working conditions are to be met [6]:

- for $X_p < 1$ şi $X_v < 1,8$ regim HD;
- for $X_u > 1,5$ şi $X_e > 0,8$ regim HD „critic";
- for $X_u > 1,5$; $X_e > 0,6$; $1,8 < X_v < 100$; $1 < X_p < 100$ regim EHD.

With mineral oils lubricated bearings the working conditions is partial or total EHD. It is provided the catalogue dynamic capacity, according to EHD theory, if the film parameter X_h has values from 1,5 to 4.

The most favourable film paramether is showed in figure 4 [6] and it was experimentaly measured by researchers from I. P. Iaşi.

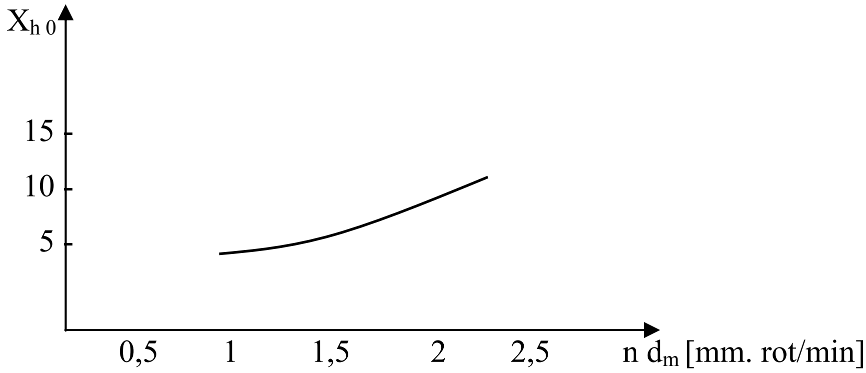

Fig. 3 The most favourable EHD film paramether variation with the diameter-revolution product of a bearing [6]

Taking into account the case of sliding contact met at radial bearings under huge loading, similar with the the case of cylindric gears contact where there are 2 possibilities: one, HD lubrication in the case of flanks with less rugosity, huge peripheric speed and moderate loading conditions and two, EHD lubrication frequently met with mechanically remade cylindric gears, the lubricating working conditions can be estimated depending on the film paramether $X_h = h_{min} / R_a$, where $R_a = (R_{a1}^2 + R_{a2}^2)^{1/2}$ is the couple's surfaces rugosity after polishing (Fig. 4) [3].

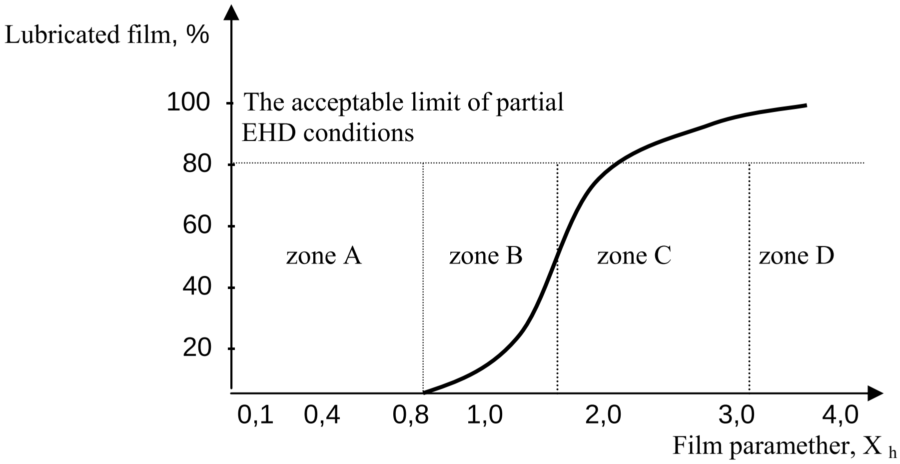

Fig. 4 SKF diagrame [3]

Thus, depending on the determined minimum thickness, h_{min} , taking into account the bush and pin's surfaces rugosity, the working condition can be measured, with the help of X_h paramether. According to figure 3, for $X_h < 0,8$ the lubricating working condition is limited or dried – zone A; for $X_h = 0,8\ldots1,5$ the lubricating working condition is mixed or limited – zone B; for $X_h = 1,5\ldots3$, the lubricating working condition is EHD partial – zone C, and for $X_h > 3$, the lubricating working condition is EHD total – zone D.

În the case of sliding the medium height of the roughness can be reckoned for the spindle $R_{1\,max} \cong 5$ μm; as for the bronze bushing $R_{2\,max} \cong 5$ μm is to be considered [4]. So, in the case of fluid rubbing functioning conditions, the minimum lubrifiant thickness has to be bigger than an allowable value, $h_{min,a} \geq 10$ μm. At macroscopic level, huge loadings make elastic distorsions. The film's thickness dependency of the contact maximum pressure is presented in figure 5 [3]:

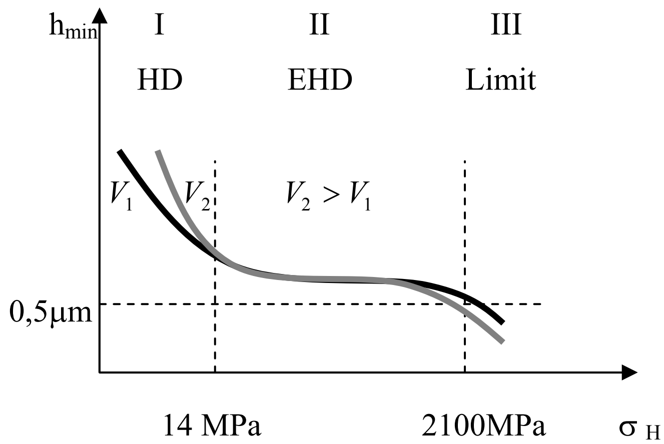

Fig.5 Film thickness dependence, taking into consideration the contact maximum pressure [3]

The lubricating changes are to be noticed in figure 2: hidrodynamic, until maximum pressure of about 14 MPa (zone I), elastohidrodynamic, until pressure of about 2100 MPa and h_{min}< 0,5μm (zone II), and onctuos-limited (zone III).

It is admitted that, in the case of huge loading forcess, at limited level, the pressure developing in the lubrifiant film may reach the second domain - EHD - figure 5, so as it is taken into account that in the case of sliding contact as well, which is to be met with radial journal bearings under huge loading, may appear elastic distorsions of the two bodies in contact with the lubrifiant [1].

Figure 6 [5] shows an example of a journal bearing with elastic contact:

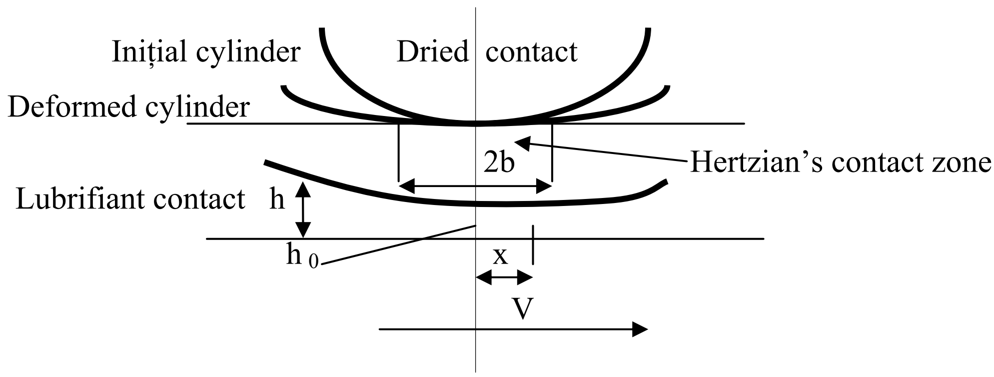

Fig. 6 Example of a journal bearing with elastic contact [5]

The lubrifiant film's minimum thickness is determined by the following relation [2]:

$$h(R,\theta) = h_0 + v(R,\theta) + \delta(\theta). \quad (4)$$

where h_{min} is the lubrifiant film's thickness in HD conditions.

2. CONCLUSIONS

Taking into account the case of radial journal bearings under huge loading with liniar contact between the pin and the bush, we may notice the HD-EHD zone conveying, in accordance with figure 5, with the possibility of the bush and the pin's elastic distorsion for pressures bigger than 14 MPa; a similar situation occures with bearings with rolls.

In the case of radial journal bearings, for loadings bigger than 30000 N, the hertziene's pressure exceeds the double limit that can be reached when passing from HD conditions to EHD conditions (at about 30 MPa), and the elastic distorsion reaches values comparable with the lubrifiant film's acceptable thickness between the pin and the bush.

In accordance with SKF diagram in figure 4, in the case of radial journal bearings, the mixed condition limit can be obtained at values of 5,5 μm for the lubrifiant film's thickness, and the partial EHD condition limit at values of 10,5 μm for the lubrifiant film's thickness, comparable with 1,2 μm for mixed conditions and 2,2 μm for partial EHD conditions, in the case of gears with polished flanks.

Concluding, it can be asserted that in the case of radial journal bearings under huge loading also, the pressure's increase may produce elastic distorsions of the pin and bush, estimating their limited functions in the field of elastohydrodynamic conditions, a similar situation can be met with cylindric gears and bearings with rolls.

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