Determination of Lubricant Layer Thickness with an Allowance for Oil Absorption Capacity of Contacting Surfaces

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Abstract. The paper describes an analytical model for calculating an extreme pressure lubricant layer thickness with an allowance for oil absorption capacity of contacting surfaces. The model is practically assessed in laboratory conditions. The experimenting work shows that the speed of the relative displacement of contacting surfaces and the oil absorption capacity of their rough layers contributes to the thickening of the lubricant. The analysis of the results shows that it is the oil absorption capacity of rough layers that has the strongest influence on the change in the thickness of the extreme pressure lubricant layer.

1 Introduction

The main reason for a failure of units and mechanisms is not a breakdown of parts under the influence of increased or peak loads, but the wear of contacting surfaces. Therefore, an increase in the lifetime of heavy-duty friction assemblies is one of the priority aspects in design and exploitation of machine units and parts.

Numerous studies show that friction surfaces with oil-retaining geometry have significantly better tribotechnical properties [1-4], and the insufficient oil absorption capacity of contacting surfaces can result in adhesion and more intensive wear of friction assemblies in the course of breaking-in and wear of microprotrusions.

In most cases, oil absorption capacity is understood as an ability of a surface to accumulate a certain amount of lubricant, which goes to a friction contact in the event of oil starvation. As for the work of a friction couple (in the event of a relative displacement of contacting surfaces), oil absorption capacity is the most important and essential property if we speak about real processes of tribology application in metallurgical and other types of equipment.

To increase oil absorption capacity of rough surfaces, various ways of forming oil retaining micro relieves are used. The micro relieves look like a system of grooves and cavities with a closed or open profile to accumulate a lubricant [4,5].

Thus, oil absorption capacity of rough surfaces plays a positive role in the interaction of friction surfaces of heavy-duty friction assemblies. However, the reviewed researches don't

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provide with complete information on the quantitative data of the influence of oil absorption capacity of contacting surfaces on the lubricant film formation and the life extension of friction assemblies.

That is why theoretical and experimental researches focused on the evaluation of the dependence of an oil absorption surface and its influence on the formation of a lubricant layer between contacting surfaces become especially important.

The aim of the paper is to find the dependence of an oil film thickness on operational modes of heavy-duty friction assemblies regarding oil absorption capacity.

The carried out analytical model for calculating a lubricant layer thickness in heavy-duty friction assemblies is based upon the contact hydrodynamic lubrication theory, theory of elasticity, and the task of the mechanics of contact interaction of elastic bodies.

We use the solution of Flamant problem of displacement of an edge of a smooth semi-plane with an equivalent elasticity modulus at a coordinate x , being under pressure with continuous distribution. In this case, the displacement of a point of an edge of a semi-plane can be described with an equation:

$$\delta(x) = -\frac{4}{\pi E'} \int_{S_1}^{S_2} P(S) \ln(x-S)^2 dS + C \quad (1)$$

where S is an additional coordinate along x axis, C is a constant.

The form of a gap between contacting bodies is defined by displacement of points of a semi-plane edge regarding an arbitrary point of contact. That is why, for a specific value of a contact region, constant C in the formula can be set to 0.

According to A.N. Grubin's theory, we suppose that the deformation of a cylinder in case of a dry contact and in case of feeding lubricant to a contact zone are similar (fig.1).

According to Hertz's theory, the value of displacement of points of an elastic body outside a contact zone will be

$$\delta_2 = h - h_0 = \frac{4W}{\pi E_2^1} \left[\sqrt{x(x^2 - 1)} - \ln \left(x - \sqrt{x^2 - 1} \right) \right] \quad (2)$$

where W is a load per unit contact surface, forcing the corresponding displacements.

Fig. 1. The geometry of a surface contact a) without a lubricant b) with a lubricant.

1 and 2 are non-deformed and deformed cylinders correspondingly.

Under high pressure values, which are specific to heavy-duty friction surfaces, there appears a dependence of lubricant viscosity on the pressure in a contact zone. This dependence has an exponential character for the majority of lubricants.

Upon integrating, Reynolds' equation looks as

$$\frac{dP}{dx} = 12\eta v \frac{h-h_0}{h^3} \quad (3)$$

Inserting the expression of the dependence of viscosity of a lubricant on the pressure $\eta = \eta_0 e^{\alpha P}$ into the equation (3) and dividing the variables, we get:

$$\frac{dP}{dx} e^{-\alpha P} = 12\eta_0 v \frac{h-h_0}{h^3} = \frac{dq}{dx} \quad (4)$$

The right part of the equation (4) is a pressure gradient in a film at constant lubricant viscosity. The value q is called reduced pressure. Integrating equation 4 regarding the boundary conditions $P=0$ at $q = 0$, we get a dependence between the values P and q in the form

$$q = \frac{(1-e^{-\alpha P})}{\alpha} \quad (5)$$

The received equation shows that provided $q \rightarrow \frac{1}{\alpha}$, insignificant changes of a film form can result in great changes of pressure. This property demonstrates the validity of the equality of dry and lubricated contacts assumed by A.M. Ertel and A.I. Gubin.

Regarding the assumed correlation $q \approx \frac{1}{\alpha}$, the equation (4) is worked out to

$$\frac{dq}{dx} = \frac{48v}{E_s} \sqrt{\frac{W}{2\pi E_s R}} \frac{h-h_0}{h^3} \quad (6)$$

where R is an equivalent of the radius of curvature, defined by the correlation $\frac{1}{R} = \frac{1}{R_1} + \frac{1}{R_2}$

The assumed pressure at a boundary of a joint:

$$q_{x=-1} = \frac{48v}{E_s} \sqrt{\frac{W}{2\pi E_s R}} \int_{\infty}^{-1} \frac{h-h_0}{h^3} dx \quad (7)$$

The numerical integration of equation 7 with the use of aquadrature and Gauss multipliers results in:

$$q_{x=-1} = 4,36 \frac{v}{E_s} \sqrt{\frac{W}{2\pi E_s R}} \left(\frac{\pi}{2W} \right)^2 \left(\frac{\pi h_0}{2W} \right)^{-1,375} \quad (8).$$

The pressure at a boundary of theregion of a static contact ($x = -1$) should be compared with Hertz maximum pressure, i.e. the assumed pressure reaches the value $\frac{1}{\alpha}$. Replacing

the left part of equation 8 by $\frac{1}{\alpha}$ and then solving it regarding the parameter of a lubricant layer thickness at the contact of bodies with rough surfaces, we get

$$h_0 = 0,888(\alpha \eta v)^{0,727} R^{0,364} \left(\frac{E_s}{W} \right)^{0,111} \quad (9)$$

Regarding the oil absorption capacity of a rough layer (2.12) the final thickness of a lubricant film is

$$h_0 = 0,888(1 + m' + m'')(\alpha \eta v)^{0,727} R^{0,364} \left(\frac{E_s}{W} \right)^{0,111} \quad (10)$$

Taking into account oil absorption of rough layers, this dependence makes it possible to calculate a lubricant layer thickness, when the surfaces are in contact with a relative displacement.

To determine the validity of the mathematical model for calculating a lubricant layer thickness, experimental studies are carried out on laboratory equipment.

The experimental researches of a lubricant layer thickness between contacting surfaces and the evaluation of validity of the developed analytical model

2 Research materials and methods

In the course of further experimental studies, a laboratory setup (SMC-2 friction machine) is used [5-7] in order to physically simulate the formation of a lubricating film in a heavily loaded friction pair. To study the change in the thickness of an oil film formed in the "roller-roller" friction pair we use a weight-of-coating test. In particular, it is an analytical balance with a measurement accuracy of 0.0001 g. The general view of the laboratory setup and the installed rollers are shown in Figure 2.

Fig. 2. a) General view of the laboratory setup; b) the "roller-roller" friction pair

The friction machine (Fig. 2a) consists of actual mechanical and measuring units (item 1), a unit for processing and transforming information (item 2) and a switchboard cubicle (item 3) [7].

To carry out the experiments, three pairs of rollers with different microtopography of surfaces are made. The variants of the microtopography of surfaces are obtained by turning, grinding and polishing.

In the course of the experiments, the clamping force and the rotary velocity are changed, thereby providing a specific load per unit contact length of 15, 60, and 100 kN /m and a rolling speed of 0.5, 1.3 and 2.0 m/s at the contact. At the same time, a lubricant with a viscosity class of 220 and a mass of 0.03 g is fed to the upper roller. For each pair of rollers, five experiments are carried out at the corresponding loads and rolling speeds. In total, 135 tests and measurements are carried out.

The square of a lubricating film emerging at the contact is measured with the help of samples taken from the surfaces of the rollers by placing a grid with a cell size of 1 mm in Photoshop CS6 program. A lubricating film thickness h_0 is defined as a ratio of mass of a supplied lubricant m_{cm} to the product of its density ρ and square of a lubricating film S_{cn} , i.e. $h_0 = m_{cm} / (\rho S_{cn})$. Taking into account the precision of measurements, the calculation accuracy is no less than 85%. Table 1 shows average values h_0 obtained by experiments.

The values of the relative oil absorption capacity are set according to the results obtained in [8-10].

3 Research results and discussion

The values of lubricating film thicknesses, calculated on the basis of the mathematical relationships [11, 12], turn out to be significantly lower than those obtained in the course of the experimental studies. This discrepancy is due to the fact that known equations do not take into account a rough layer characterized by certain oil absorption capacity, though the latter makes an additional contribution to the formation of a lubricating film.

Some values of lubricating film thicknesses, obtained by generalizing theoretical data on dependence 10 and as a result of our experimental studies, are shown in Fig. 3.

Fig. 3. The results of theoretical and experimental studies of thicknesses of a lubricating film

The analysis of the presented diagrams shows that with an increase of a load force acting on contact surfaces in friction assemblies, there appears a decrease in a lubricating film thickness. The intensity of the changes is more noticeable in the region of low values of loads than in the region of the increased ones. In the range of changes from 15 to 100 kN/m, i.e. 6.7 times, the lubricating film thickness decreases 1.23 times.

Table 1. Thicknesses of a lubricating film, microns, in a contact zone of a friction pair "roller-roller" under various conditions of: oil absorption capacity of surfaces, specific loads and rolling speeds

Grinding(m = 0,6)			Turning(m = 0,5)			Polishing (m = 0,4)		
Rolling speed, m/s								
0,5	1,2	2,0	0,5	1,2	2,0	0,5	1,2	2,0
Specific load 15kN /m								
2,24 (2,54)	5,52 (4,80)	6,32 (6,95)	2,19 (2,35)	5,33 (4,44)	6,75 (6,43)	2,46 (2,14)	3,56 (4,04)	4,98 (5,86)
Specific load60kN /m								
2,44 (2,18)	3,65 (4,11)	6,45 (5,96)	2,17 (2,01)	3,42 (3,80)	4,82 (5,51)	1,62 (1,84)	2,95 (3,47)	5,63 (5,03)
Specific load100kN /m								
1,83 (2,06)	3,42 (3,89)	6,76 (5,63)	1,67 (1,90)	3,95 (3,59)	4,90 (5,21)	1,40 (1,73)	3,54 (3,28)	4,23 (4,75)

Note Values in parentheses are calculated on the dependence (10).

An increase in a lubricating film thickness is facilitated by a rate of relative displacement of contacting surfaces and oil absorption capacity of their rough layers. An increase in speed from 0.5 to 2.0 m/s, i.e. 4 times, leads to an increase in a lubricating film thickness by an average of 2.8 times. A change in the relative oil absorption capacity from 0.4 to 0.6 increases a lubricating film thickness by about 1.6 times.

Thus, the parameter of relative oil absorption capacity of rough contacting surfaces has the greatest influence on the change in a lubricating film thickness in friction assemblies.

According to GOST 18855-94 (ISO 281-89), a corrected rating life of special bearing properties and special operating conditions are calculated by the formula:

$$L_{na} = a_1 a_2 a_3 L_{10} \tag{11}$$

- where L_{10} , standard rating life;
- a_1 , coefficient, correcting life according to reliability;
- a_2 , coefficient correcting life depending on special properties of a bearing;
- a_3 , coefficient that corrects life depending on operating conditions of a bearing.

The value of a_1 coefficient is established and regulated by GOST, a_2 coefficient is determined by a manufacturer of bearings, and, in general, does not exceed 1.0. According to GOST, a_3 coefficient can be more than 1 under favorable lubrication conditions.

The lubrication conditions and the mode of friction between contacting surfaces is determined by the parameter λ , which depends on the thickness of a lubricating layer. In this regard, coefficient a_3 is proposed to be determined as follows:

- at $3 \leq \lambda \leq 7, a_3 = 1$;
- at $3 < \lambda, a_3 = 0,8$;
- at $\lambda > 7, a_3 = 1,1$.

This approach to determine the life is also valid for gears used in metallurgical units.

4 Results

1. Using main theses of the contact-hydrodynamic theory of lubrication of heavy-duty tribo-couplings, an equation is obtained, which is supplemented by a complex oil absorption parameter. The equation takes into account geometric (radius of curvature of contacting bodies) and physical (speed of moving surfaces) characteristics of contacting surfaces, as well as a processing method. The dependence makes it possible to calculate a thickness of a lubricating layer when surfaces are in contact, regarding their relative displacement and oil absorption capacity of their rough layers.

2. The experimental studies carried out in laboratory conditions with an aim to determine a thickness of a lubricating layer between contacting surfaces shows that oil absorption capacity of rough layers has the greatest influence on the change in the thickness of a lubricating film in friction assemblies.

It is determined that with an increase in a load force acting on the contact of surfaces in friction assemblies, the thickness of a lubricating film decreases. In the region of low values of a load force, the intensity of changes is more noticeable than in the region of the increased ones. In the range of changes from 15 to 100 kN/m, i.e. 6.7 times, the thickness of a lubricating film decreases 1.23 times. An increase in speed from 0.5 to 2.0 m/s, i.e. 4 times, leads to an increase in the thickness of a lubricating film by an average of 2.8 times. The change in the relative oil absorption capacity from 0.4 to 0.6 increases the thickness of a lubricating film by about 1.6 times. The convergence of the experimental values and those calculated by the proposed method was 85%.

3. A coefficient is proposed which corrects the basic resource of rolling bearings and gear drives. It can take the values from 0.8 to 1.1, depending on the oil absorption capacity of contacting rough layers and the operating parameters of machine and assembly units.

5 Discussion

The comparison of the obtained data on the thickness of a lubricating layer (regarding the oil absorption capacity which was experimentally obtained and analytically calculated) shows that the thickness of a lubricating layer is higher than it was previously thought. Analyzing the data obtained, it is found that the parameter of relative oil absorption capacity of rough contacting surfaces has the greatest influence on the change in the thickness of a lubricating film in friction assemblies.

The verification of the validity of the developed analytical model for calculating the thickness of a lubricating layer in a heavy-duty friction pair is carried out on modern laboratory equipment according to an original technique, taking into account various parameters of interaction of contacting surfaces in the presence of a lubricant. It is noted that an increase in the thickness of the lubricating film is facilitated by the rate of relative displacement of contacting surfaces and the oil absorption capacity of their rough layers. With an increase in the load acting on the contact surfaces in friction assemblies, the thickness of a lubricating film decreases.

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