Problems of Tribology, V. 28, No 2/108-2023, 15-19

Problems of Tribology
Website: *http://tribology.khnu.km.ua/index.php/ProbTrib*
E-mail: *tribosenator@gmail.com*

DOI: https://doi.org/10.31891/2079-1372-2023-108-2-15-19

Influence of lubricant material in the point contact zone of rolling friction on fatigue life for friction bearing units

A. Milanenko*, A. Savchuk, Y. Turitsa

National Transport University, Kyiv, Ukraine
E-mail: milanmasla@gmail.com

Received: 5 April 2023: Revised: 14 April 2023: Accept:5 May 2023

Abstract

A comprehensive methodology has been developed to assess the rheological and tribotechnical properties to establish the impact of the lubricant in dynamic lubrication conditions. The results of the research indirectly affect the fatigue life of bearing units under conditions of rolling friction.

Key words: Point contact zone, fatigue life, bearing units, elastohydrodynamic (EHD) lubrication, lubricant material (lubricant), rheological and tribotechnical properties, rolling friction.

Introduction

Although lubrication is necessary for the satisfactory operation of rolling bearings, the effect of lubricant on the fatigue life of the bearing has not been sufficiently studied. In recent times, the theory of elastohydrodynamic (EHD) lubrication [1] has been used to explain the different effects of lubricants. According to this theory, the thickness of the lubricating layer separating the moving elements of the bearing is determined by the viscosity-pressure dependence of the lubricant. The contact of surface micron irregularities does not occur if it is possible to maintain a sufficient thickness of the lubricating layer - in this case, the long-term durability of the bearing is ensured. If the film thickness is reduced to a level where surface irregularities are encountered, the fatigue life rapidly decreases with increasing contact frequency. In any case, a comprehensive calculation methodology is needed that would allow to take into account the influence of lubricant on the fatigue life of bearing units.

The purpose of the work

To develop a comprehensive methodology for assessing the influence of lubricant on rheological and tribotechnical properties under rolling friction conditions for bearing units.

Technique for researching the properties of coatings for the influence of lubricant material.

For modern multigrade transmission, aviation and motor-transmission oils, it is necessary to evaluate the dynamic viscosity on a rotational viscometer in order to approximate the real operating conditions of various bearing units - namely, at high shear rates to ensure mechanical stability and film strength under high contact pressure, maximum contact and sub-surface tangential stresses and deformations, which is the cause of fatigue wear of the ball and sections of the friction paths of the bearing cage.

Of great importance for the technology of lubrication of heavily loaded parts of tribomechanical systems are the conditions of formation of an elastohydrodynamic (EGD) lubricating layer in a local (point) friction contact, under which pressure significantly affects the viscosity of the lubricant, as well as the deformation of the contacting bodies (film shape). The viscosity of compressible fluids increases with increasing pressure. The relatively low compressibility of mineral and synthetic oils leads to a significant increase in viscosity at high pressures.

Consequently, at points subjected to high load, the effective viscosity becomes higher than the nominal viscosity of oils at equal temperatures. At the same time, an increase in temperature under such conditions leads to a decrease in the effect of viscosity increase with increasing pressure. Also very important is the influence of the rheological behavior of the oil, which is observed, for example, when the dynamic viscosity increases with respect to atmospheric pressure and temperature at the entrance to the contact, which leads to an increased effect on the change in viscosity with increasing pressure and temperature.

This special (non-Newtonian) rheological behavior of the oil with increasing pressure and temperature leads to a strong dilution of the structural viscosity of the oil due to the addition of polymeric viscosity modifiers to the base base to extend the viscosity-temperature range of operation, which, when the temperature changes, shrink/expand, bringing the characteristics of the base bases to the required values. At high shear rates, polymers line up in the direction of flow and shrink, resulting in oil thinning. In addition, some polymers at high shear rate, on the contrary, thicken (expand their volume), and the fluidity characteristics of such fluids somewhat lose linearity depending on temperature.

Taking care of this problem, it was necessary to develop a methodology for calculating rheological and tribotechnical characteristics, simulating real operating conditions for various bearing units.

As oils, were investigated: two multigrade gear oils with different viscosity classes SAE 75W-90 and SAE 80W-90 (manufactured by KSM PROTEC), respectively, which are used for most friction units of the transmission of passenger cars, trucks and agricultural machinery; aviation oil MS-8p (MS-8pn, manufactured by ZTM "ARIAN") for lubrication of friction bearing units of gas pumping units based on aviation GTD and universal motor-transmission oil MT-8p (EMT-8, manufactured by ZTM "ARIAN") for lubrication of friction units of auxiliary internal combustion engine (up to 35 hp) and transmissions of military vehicles. s.) and transmissions of military and agricultural tracked vehicles.

Dynamic viscosity (η) is a measure of the resistance to fluid flow or deformation. The dynamic viscosity of an oil (η) at temperature (t) is calculated from the kinematic viscosity using the following formula:

$$\eta = \upsilon\,\rho, \qquad (1)$$

where υ is the kinematic viscosity, mm^2/s.

ρ is the oil density at the same temperature at which the kinematic viscosity υ was determined, g/sm^3;

From the point of view of the rheological characteristics of the oils under study, a more universal definition of dynamic viscosity should be used, which would be equally suitable for Newtonian and non-Newtonian oils. For this purpose, the dynamic viscosity is defined as the ratio of shear stress (τ) and shear rate (D):

$$\eta_0 = \frac{\tau}{D}, \qquad (2)$$

Measurement of dynamic viscosity (η_0) at atmospheric pressure and temperature at the contact inlet is carried out by means of a rotational viscometer (Fig. 1) using a coaxial-cylindrical measuring device (Fig. 2).

When determining the dynamic viscosity, rotational viscometers have the advantage that they allow to measure viscosity as a function of time at high shear rate up to $10^6\ s^{-1}$ and to determine the presence of hysteresis of elastic-viscous properties.

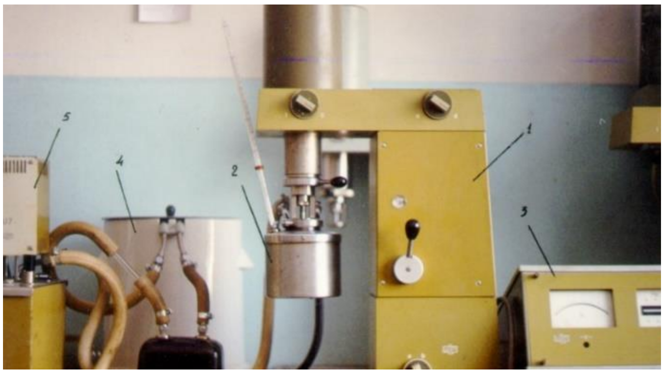

Fig. 1. General view of the rotational viscometer REOTEST 2.1.

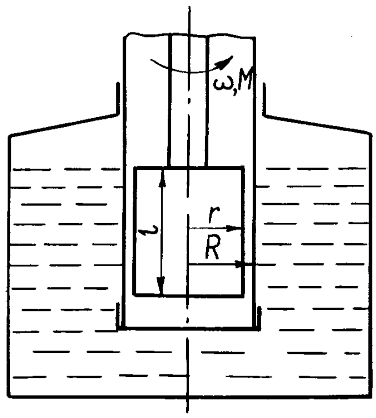

Fig. 2. Scheme of the coaxial-cylindrical system REOTEST 2.1.

Shear stress (τ) and shear rate (D) are determined according to the following relations:

- Shear stress (τ):

$$\tau = \frac{M}{2\cdot\pi\cdot l\cdot r^2}; \qquad (3)$$

- Shear rate (D):

$$D = \frac{2 \cdot \omega \cdot R^2}{R^2 - r^2}; \tag{4}$$

To determine the dynamic viscosity as a function of pressure, the exponential law [1] of the viscosity-pressure relationship described by the Barus dependence is used:

$$\eta = \eta_0 \cdot e^{\alpha \cdot P}, \tag{5}$$

where η is the viscosity at pressure P; η_0 is the viscosity at the contact inlet at P = 0; α is the coefficient of dependence of viscosity on pressure (piezo-viscosity coefficient).

The piezo-viscosity coefficient (α) [2] characterizes the effective viscosity with changing pressure and temperature and is determined by the following equation:

$$\alpha \cdot P = A \cdot \left[\left(\frac{T - 138}{T_0 - 138} \right)^{-S_0} \cdot B^Z - 1 \right], \tag{6}$$

where $A = ln(\eta_0) + 9{,}67$ – rheological constant for the oil under study;
$B = 1 + 5{,}1 \cdot 10^{-9} \cdot P_{max}$, *Pa;*
η_0 – dynamic viscosity at $P_{max} = 0$, *Pa·s*;
P_{max} – maximum Hertzian pressure, *Pa;*
T_0 – atmospheric temperature, *293 K (20⁰C);*
$Z = 0{,}67$ – a constant that depends on the pressure P_{max};
S_0 - a constant that depends on temperature (T) and can be found by the following relation:

$$S_0 = \frac{\beta \cdot (T_0 - 138)}{A}, \tag{7}$$

where β – thermal viscosity coefficient.

In the conditions of EGD lubrication mode, the time of passage of the lubricant through the contact zone is very short, and the process is accompanied by large changes in pressure, temperature, stress and shear rate gradients. Under this friction regime, even lubricants of mineral origin, which are traditionally considered Newtonian, can show deviations from this model.

Non-Newtonian fluid is characterized by the fact that the shear stress between the layers is not directly proportional to the shear rate. It can also be characterized by a shear stress and depend, in addition to the velocity gradient in the oil flow, on temperature, pressure, etc.

The rheological behavior of oils depends to a large extent on the frequency of loading, because at short-term but high-frequency loads, corresponding, for example, to rolling bearings, the oil measured in the pressure zone does not have time to relax and acquire the input properties. Therefore, at excessive pressure and shear rate, the influence of viscoelasticity (rheology) on the thickness of the lubricating layer is noticeable. Non-Newtonian properties, which change the value of normal and tangential stresses, significantly affect the friction force and the thickness of the lubricating layer.

Thus, there are cases when some oils, which under normal conditions act according to the laws of Newtonian fluid, in some cases, in particular under high-speed, high-load and high-temperature conditions, can behave as non-Newtonian fluids, distorting the results obtained theoretically. At the same time, the minimum and central thickness of the lubricating layers between the friction surfaces of the bearing units change significantly.

The main characteristic of the point contact - the central thickness of the lubricating layer (H_0) - can be generalized as a function of the change of three dimensionless parameters of speed (U), load (W) and material properties (G), which can be written in the following form for the point (circular) friction contact [2]:

$$H_o = 3{,}49 \cdot U^{0{,}75} \cdot W^{-0{,}206} \cdot G^{0{,}426}. \tag{8}$$

The dimensionless parameters in the above formula are defined as follows:

1. Dimensionless film thickness in the central contact area:

$$H_0 = \frac{h_0}{R}. \tag{9}$$

2. Dimensionless velocity parameter:

$$U = \frac{\eta_o \cdot V}{E' \cdot R}. \tag{10}$$

3. Dimensionless load parameter:

$$W = \frac{F}{E' \cdot R^2}. \tag{11}$$

4. Dimensionless parameter of materials:

$$G = \alpha \cdot E'. \tag{12}$$

Results of calculations for the influence of lubricant materials

Table 1 shows the input data and formulas for calculations:

Table 1

The input data and formulas for calculations.

Modulus of elasticity of steel $E_1 = 2{,}07 \cdot 10^{11}$ *Pa*	The reduced radius of curvature $R = 6{,}35 \cdot 10^{-3}$ *m*
Modulus of elasticity of glass $E_2 = 0{,}757 \cdot 10^{11}$ *Pa*	Rolling velosity *V = 1,2 m/s*
Poisson's ratio of steel $\upsilon_1 = 0{,}3$	Temperature at the contact inlet *T = 293K*
Poisson's ratio of glass $\upsilon_2 = 0{,}25$	Operating temperature in contact *T = 343K*
Reduced modulus of elasticity $E' = \frac{2}{\left(\frac{1-\nu_1^2}{E_1} + \frac{1-\nu_2^2}{E_2}\right)} = 1{,}19186 \cdot 10^{11}, Pa$	Thermal viscosity coefficient, β: *β (MS-8p) = 0,027;* *β (SAE 75W-90) = 0,030;* *β (MT-8p) = 0,030;* *β (SAE 80W-90) = 0,045.*
Dynamic viscosity in atmospheric conditions at the entrance to the contact, η_0, *Pa·s:* η_0 *(MS-8p) = 0,185;* η_0 *(SAE 75W-90) = 0,244;* η_0 *(MT-8p) = 0,264;* η_0 *(SAE 80W-90) = 0,478.*	Dynamic viscosity η depending on pressure and temperature is calculated by formula (5). The piezo-viscosity coefficient α is calculated by formula (6).

Calculations of the change in dynamic viscosity η/η_0 with pressure and temperature (rheological properties) and the change in the central thickness h_0 of the lubricating layer with pressure and temperature (tribotechnical properties) were made for two multigrade gear oils with different viscosity classes SAE 75W-90 and SAE 80W-90, respectively, aviation oil MS-8p and universal motor-transmission oil MT-8p, and the corresponding results are presented graphically (Fig. 3 - 4).

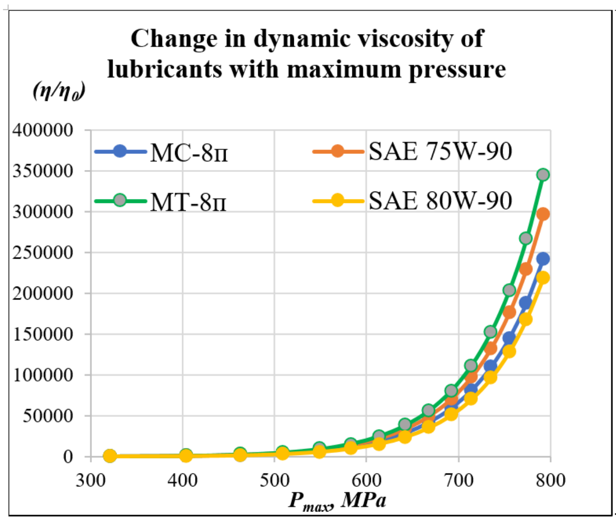

Fig. 3. Influence of the rheological feature (change in dynamic viscosity) of lubricants of different physical and chemical composition with increasing maximum pressure, taking into account the temperature in the contact zone T = 343 K for friction bearing units

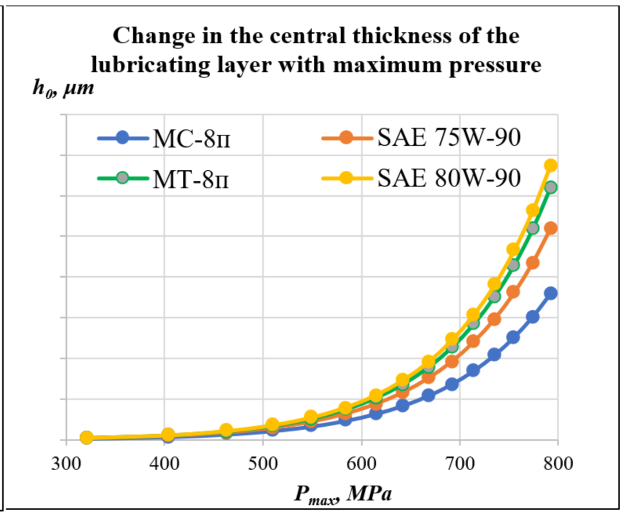

Fig. 4. Change in the central thickness of the lubricating layer with increasing maximum pressure, taking into account the temperature in the contact zone T = 343 K for friction bearing units lubricated with lubricants of different physical and chemical composition

Conclusions

A complex method of calculation of the influence of rheological and tribotechnical properties of the lubricant with increasing maximum pressure, taking into account the temperature in the point contact zone, which indirectly expresses the fatigue life of bearing units under EGD lubrication conditions, is presented.

References

1 Dmitrichenko N.F. The Elastohydrodynamics: Theory and Practice / N.F. Dmitrichenko. - Lviv: Publishing house of National University "Lviv Polytechnic", 2000. – 224p.

2. Dmytrychenko M.F. Lubricating action of oils in conditions of elastohydrodynamic lubrication / M.F.Dmytrychenko, O.A.Milanenko - K.: Informavtodor, 2009. 184p.

Міланенко О., Савчук А., Туриця Ю. Вплив мастильного матеріалу в зоні точкового контакту на втомну довговічність в умовах тертя кочення для підшипникових вузлів тертя.

Представлена комплексна методика розрахунку впливу реологічних й триботехнічних властивостей мастильного матеріалу при збільшенні максимального тиску з урахуванням температури в зоні точкового контакту, що опосередковано виражає втомну довговічність підшипникових вузлів в умовах ЕГД мащення. Встановлено, що для найбільш високов'язкої трансмісійної оливи SAE 80W-90 характерна найменша залежність динамічної в'язкості від максимального тиску, з урахування того, що дана трансмісійна олива створює більші товщини мастильного шару, що підтверджує наші припущення про те, що для високонавантажених трансмісій потрібно використовувати трансмісійні оливи більшої в'язкості.

Нехарактерні кращі реологічні властивості (найменша залежність динамічної в'язкості від максимального тиску при певній температурі в зоні контакту) найбільш високов'язкої трансмісійної оливи SAE 80W-90 підтверджують уявлення про те, що, крім обмеженого впливу в'язкості оливи, втомну довговічність визначають і й хімічний склад оливи, що обумовлює селективні хімічні реакції між окремими компонентами (поверхнево-активними речовинами) оливи та металічними поверхнями.

Ключові слова: Точковий контакт тертя, втомна довговічність, підшипники кочення, еластогідродинамічне (ЕГД) мащення, мастильний матеріал, реологічні і триботехнічні властивості, тертя кочення.