Transitional lubricating process in plain bearings in machines

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Abstract. Basing on the summary of existing models of the interaction between sliding surfaces, a model of a transitional lubrication process in lubricated plain bearings has been developed; parameter Pg of longevity of the lubricating film was used to assess it. Relation of this parameter has been reviewed to load and speed, and design and technological state of kinematic pairs in machines.

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Introduction.

Operating conditions of plain bearings in machines that contain lubricating material have been examined in literature well enough [1-7]. Results of analysis of these conditions are in the basis of the existing quality management system for plain bearings. However, plain bearings in machines often have insufficiently high reliability. This determined the need to extend existing ideas about working conditions of these friction units.

Plain bearings in real mechanical systems experience friction due to alternate appearance and disappearance of bearing lubricating film between friction surfaces. Here we have the so-called "transition lubrication process" [2] that lies in the fact that the dynamic load alternately passes through the oil film wedge or directly through the contact surfaces. Each of the stated kinds of interaction has its own mechanism of surfaces friction and wear; therefore intensity of wear for different parts is different. In case of fluid friction, it is 7-13 orders less than in case of contact interaction.

Authors have studied regularities in the transitional lubricating process in plain bearings depending on design and technological status, as well as changes in operation. Influence of load and speed of plain bearings operation on the transitional lubrication process has been studied.

Model of forming parameters of plain bearings in operation. The idea of the working conditions of plain bearings during operation is provided by the structural model of its parameters during operation shown in Figure 1[8]. It has some significant differences from traditional models.

Conditions of plain bearings' operation are determined by the state of their parts [9]: initial state before S_H operation is described by vectors S_K and S_T that are formed at the stages of design and manufacturing, respectively. The state of plain bearings in course of CE machines operation is a variable and depends on both S_H and changes in its own value [delta] S_E in course of operation.

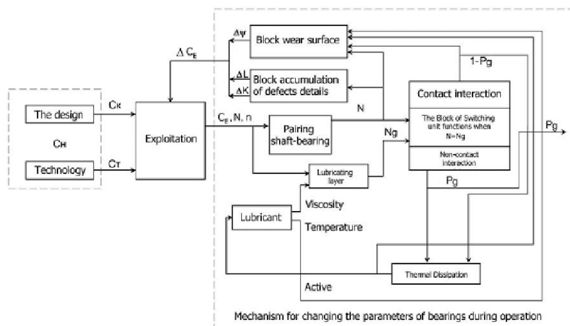

Figure 1 - Structural model of forming parameters of plain bearings in operation.

Operation of bearings in the transitional lubrication process is described by parameter P_g of longevity of the lubricating film [2]. It shows how long the lubricating film existed in the plain bearing. Remaining time (1- P_g) sliding surfaces were in contact interaction.

Parameter P_g is sensitive to the state of plain bearings and current load vector, and determines intensity of wear of sliding surfaces. It can be applied to main modes of interaction in these bearings: non-contact (fluid friction) with P_g =1, contact (dry and boundary friction) with P_g =0 and transition lubrication process where 0< P_g <1.

Specific nature of surfaces interaction in a plain bearing is defined by [10] the relationship between ultimate load capacity of lubricating film N_g, and external load N transmitted from one part to another including through damping lubrication film. The lubricating film between friction surfaces is kept with N_g >N. In boundary state that corresponds to the moment of changing the nature of surfaces interaction, dynamic balance N_g =N is established. With N_g <N, the lubricating film is destroyed and contact interaction occurs that mainly leads to wear of friction surfaces.

Components of the dynamic balance depend on many factors, including those caused by operational changes of volume parameters of the "shaft-lubricant-bearing" tribotechnical system. We will focus the analysis of dynamic balance on identifying the structure of its components - forces N and N_g.

One component of the external load is N working load N_p dependent on the work made by the mechanism used for intended purpose that is defined during machine design. Another structural component of the external load is additional load Na that depends on volumetric parameters of the shaft and bearing.

From the set of parameters that characterize volumetric physical, mechanical and geometrical properties of bearing assembly parts for analysis of additional load, stiffness K and runout L of shaft axis were taken. Shaft axis runout is a consequence of plain bearings long-term operation in machines.

The mechanism of the forming additional load has been examined on the example of a crankshaft main bearer of a car engine, with the assumption that the insert and the shaft have regular geometric shape; insert support stiffness many times exceeds stiffness of the shaft; shaft axis runout is not zero, bearing axis runout is zero.

Value N_a varies according to crank angle [alpha] and is determined by stiffness of crankshaft K, axis runout L and diametrical clearance [psi]:

$$N_a = K(L - \psi / 2) sin\alpha \ . \quad (1)$$

The analysis made makes it possible to conclude that the total external load is

$$N = N_p + N_a \ . \quad (2)$$

The above components of value Na are not exhaustive. Components of other origin can be added to them, which also change the dynamic picture in plain bearings in cars.

Another component of the dynamic balance, the ultimate bearing capacity of lubricating film N_g, depends in this case on the state of plain bearing and the lubricant. With their constant states, value N_g almost linearly depends on shaft rpm n in the insert [3, 4, 6]. Ultimate operating bearing capacity N_g differs from design $N_{g.d}$. In-service defects of parts not only lead to increasing loan N_a, but to increasing lubricant leakage from the loaded area, reducing pressure in the oil wedge. This causes change in the ultimate bearing capacity of lubricant film N_g by value [delta] N_g:

$$N_g = N_{g.d} + \Delta N_g \ . \quad (3)$$

The dynamic balance in a plain bearing in boundary conditions of transition from one type of surfaces interaction to another with regard to relations (2) and (3) can be written as:

$$N_p + N_a = N_{g.d} + \Delta N_g \ .$$

The ultimate bearing capacity of the lubricating film $N_{g.d}$ is usually calculated for N_p. To ensure continuity of lubricating film in operation of the bearing, a reserve should be ensured for its bearing capacity for additional load N_a, which is not considered in practice.

Parameter P_g depends not only on the volumetric properties of the part, but also on geometric parameters of the interacting surfaces and their relative position. One of such characteristics is diametrical clearance [psi] in bearing connection. From the analysis of dependencies (1) and (2) it follows that load N is inversely related to diametrical clearance [psi]. With increasing clearance [psi], e.g., due to wear during operation, with constant volumetric properties of the shaft, value of load N is reduced due to component N_a. At first glance, it may be regarded as a favorable process. However, with increasing diametrical clearance [psi], reduction occurs in lubricating film ultimate load capacity N_g.

From the analysis of dependence N_g=f ([psi]) [2, 3, 6] it follows that value N_g is inversely proportional to the diametrical clearance squared [psi]. Therefore, load N_a decreases slower than value N_g. This increases shaft rpm n_1 where volumetric properties of the lubricant start manifesting themselves, and shaft rpm n_2 increases, which characterizes formation of stable liquid friction. With a significant increase of the diametrical clearance [psi], value n_2 may exceed the ultimate rpm of machine shaft $n_{\lim}$, and in this case it is impossible to reach the beginning of stable liquid friction.

Thus, the factor determining the existence and transition of surfaces interaction from non-contact to contact, along with other, is deviation of parts' bearing couplings parameters and lubricant properties from their design values. This leads to deterioration in hydrodynamic characteristics of a plain bearing while increasing its load, which provides reduction of parameter P_g, increasing wear rate of the sliding surfaces, and decreasing reliability.

Method for calculating duration of lubricating film parameter.

In order to experimentally determine parameter P_g, appropriate methods and technical means [11, 12] were developed, in particular, analyzer of friction modes ART-1, which combines an electric pulse generator, a transducer, and a cymometer that acts as computing and analytical device.

Generator of input commands generates bipolar electrical pulses with voltage amplitude U_g=500 mV and frequency K_g=800 KHz, which are sent to the plain bearing. Upon surfaces contact, pulses are transmitted to the machine frame and are not registered with hardware. In the presence of lubricating film between sliding surfaces, electrical signals pass through the plain bearing without changing their frequency and amplitude surge, and are recorded by the computing and analytical device after conversion. Figure 2 shows a waveform that explains transmission of electrical pulses in the bearing. Surge of oscilloscope beam corresponds to the moment of non-contact interaction of surfaces.

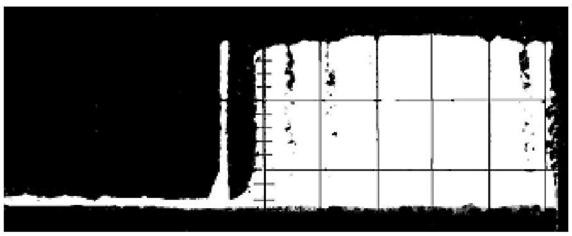

Figure 2 - Waveform of an electrical signal in a bearing

As a result of alternate appearance and disappearance of the lubricant film in a plain bearing, electrical signal is a sequence of electrical pulses. The computing and analytical device determines the number of electrical pulses K_f per second. The relation $K_f/K_g=P_g$ determines duration of lubricating film existence.

Parameters of the lubricating film in plain bearings were tested using friction machine SMC-2. The friction machine has been modified compared to the standard design in order to ensure smooth change of shaft speed from 0 to 30 rpm by installing an electric DC motor with thyristor control. The bearing was electrically insulated from the machine frame, and the shaft was connected to the housing by means of the current collector. This created the conditions for studying conditions for an electrical signal to pass through lubricating film in the bearing.

The part that serves as shaft sample and installed on the friction machine is a metal disc made of Grade C steel, 15 mm thick with diameter about 50 mm. Three shaft and insert samples were taken so that the diametrical clearances for bearings were: # 1 - 80 µm, # 2 - 363 µm, # 3 - 210 µm. C 20W-20 motor oil was used, supplied to the bearing by gravity.

Shaft speed varied from 2 to 20 rpm, radial load - from 0 to 1000 N. Bearings were tested as follows. Radial load N=0 was set on bearing, shaft speed changed from 2 to 20 rpm with 2 rpm increments and for each value, n was determined by parameter P_g. Thereafter, the load was increased with increments [delta] N=250 N to 1000 N, and test stages were checked similar to the above.

Each lain bearing was tested for wear resistance. Volumetric oil temperature was stable within 30 ± 2 C. Test conditions were as follows: shaft speed n=14 rpm; radial load N=500 N; test duration was 40 hours.

Properties of plain bearings during operation in transient lubricating process.

Methods of assessing the state of plain bearings is based on examination of dynamics parameter P_g depending on shaft speed n and radial load N [11, 12]. Changing parameter P_g with increasing n makes it possible to observe the basic processes in the lubricating film: emergence of the lubricating film, increasing its limiting carrying capacity N_g and its subsequent decrease due to intensification of heat radiation in the lubricating film.

Figure 3 shows dependence of parameter P_g on shaft speed n with different radial loads N and diametrical clearance [psi] for bearings # 1 and #2. Influence of load N on curves $P_g=f$(n) is ambiguous. Figure 3b shows dependence of parameter P_g on radial load N at different shaft speeds n for the same bearings.

From the dependences shown it follows that bearing # 2 in the absence of radial load (N=0) reaches steady state of hydrodynamic lubrication at shaft speed n=1 rpm, which exists in the whole selected range of rotation speed (not shown in the figure). With increasing radial load N to 250 N, this bearing develops carrying capacity of the lubricant film N_g from smaller values of rotation speed (for n=2 rpm, parameter P_g is equal to 0.62) and reaches steady mode of lubrication at n=6 rpm. However, settled lubricating film does not already exist in the entire determination range n, but starting with n=16 rpm, parameter P_g decreases, which indicates intensification of heat radiation processes in the bearing interface.

With further increase of load to 500 N, 750 N (not shown), the lubricant starts manifesting its volumetric properties at high rotation speeds n: with N=500 N, rotation speed n=4 rpm, with N=750 N, n=6 rpm.

Along with that, value of parameter Pg decreases (0.2 and 0.02, respectively). Characteristic of parameter P_g dependence on increasing n at large values of N is the fact that in bearing # 2, formation of lubricating film is not reached, but the predetermined highest values are reached where parameter P_g decreases with increasing n, and reaches zero.

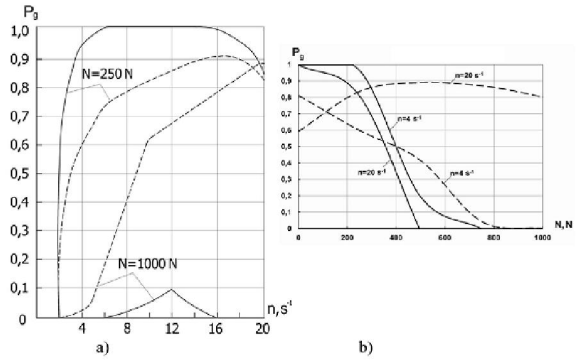

Figure 3 - Dependence of parameter P_g on shaft speed n under various loads N (a) and on radial load *N* at different shaft speeds *n* (b): solid and dotted lines - [psi] is equal to 363 and 80 µm, respectively.

Different character of dependencies is observed for plain bearing # 1. At radial load N=0, parameter P_g increases with increasing n, the state inherent for a steady lubricating film is not reached. After a certain value of n, parameter P_g starts decreasing. Similar dependence $P_g=f(n)$ is observed at radial load N=250 N. However, with further increase of N, character of function $P_g=f(n)$ in bearing # 1 changes. This change consists in the fact that an increase of parameter P_g is observed with increasing shaft rotation speed n throughout its range. Inverse process is absent. Along with that, parameter P_g does not reach unity.

Analysis of these dependences $P_g=f(n)$ for bearings # 1 and # 2 shows an ambiguous effect of radial load N on them. Dependence P_g=F(N) for bearing # 2 has monotonically decreasing nature. However, intensity of its decreasing varies at various intervals of N. It is also possible to characterize this dependence for bearing # 1, but only for low rpm n (n=2, 4, 8 rpm). For other values of rpm n, this uniqueness disappears and dependence P_g=F(N) becomes more complex. Two trends can be observed in its development. The first is that as load N increases from zero to 500 N, parameter P_g increases, the second is that with N> 500 N, parameter P_g decreases.

Experimentally obtained dependences $P_g=f(n)$ and P_g=F(N) are similar to those mentioned in literature [2]. Along with that, there are a number of significant clarifications about behavior of the lubricating film that go beyond traditional concepts.

The theory of hydrodynamic friction in plain bearings [3, 4, 6] relates behavior of the lubricant film to the generalizing coefficient of Sommerfeld So, which considers the technical state of kinematic couplings (in particular, diametrical clearance [psi]) unchanged. The results obtained provide evidence of the need to consider technical state, which may become an important addition to development of this theory that clarifies its scope and shows ways of improving design, technological and other parameters of movable couplings and their parts.

Before durability test, bearing # 1 had the highest value of parameter P_g=0.93, and wear rate was $9\cdot10^{-11}$; bearing # 2, before the test, with the lowest value of parameter P_g =0.50 had wear rate $1.05\cdot10^{-9}$. At the same time it was found that wear rate per unit of distance of samples boundary friction was not different and was $1.17\cdot10^{-9}$.

The tests performed made it possible make the conclusion about quasilinear dependence of wear on the distance traversed by the plain bearing surfaces of parts during contact interaction, and about the fact that wear of parts usually happens during this kind of friction. In addition, it can be said that parameter P_g largely determines durability of plain bearings: the greater is the value, the less is the wear rate. Thus, ceteris paribus, the wear rate in bearings # 1 and # 2 differ 11.6 times. Therefore, reliability of plain bearings can also be controlled using parameter P_g by various factors. Parameter P_g should be controlled bearing in mind the diametrical clearance [psi] in the bearing. Thus, from the relationship shown in Figure 4, we can conclude that the dynamics of the lubricating film depends on diametrical clearance.

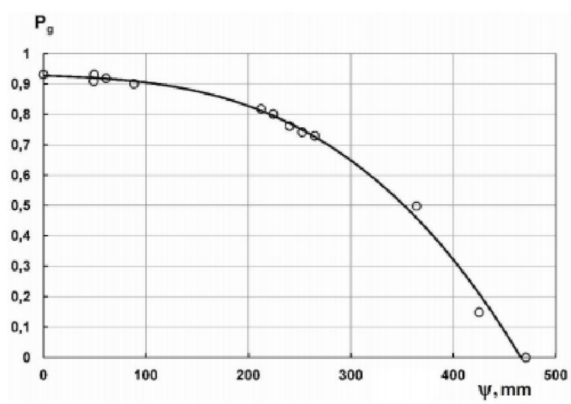

Figure 4 - Dependence of parameter P_g on diametrical clearance [psi]

Experimental data have been approximated by a square trinomial

$$P_g = 0{,}98 - 2{,}7\cdot10^{-5}\psi - 3{,}74\cdot10^{-6}\psi^2 \cdot$$

Parameter P_g is determined by the diametrical clearance [psi] in quadratic dependence, same as the carrying capacity of the lubricant film N_g. Similarity of dependencies of parameters P_g and N_g on the diametrical clearance [psi] indicates correctness of the approach of considering parameter

P_g as a function of load ratio N_g that acts on the lubricant film from the outside and its carrying capacity N_g.

Optimization of plain bearings load and speed operation mode.

Determining load and speed operation mode of plain bearings in order to ensure the lowest wear rate during operation. Favorable operating mode of plain bearings should be the one where parameter P_g will have the greatest value. This is evidenced by the data presented in the coordinate system "radial load *N* - shaft rpm *n*" (Figure 5) in the form of dependencies with equal values of P_g for bearings # 1 and # 2.

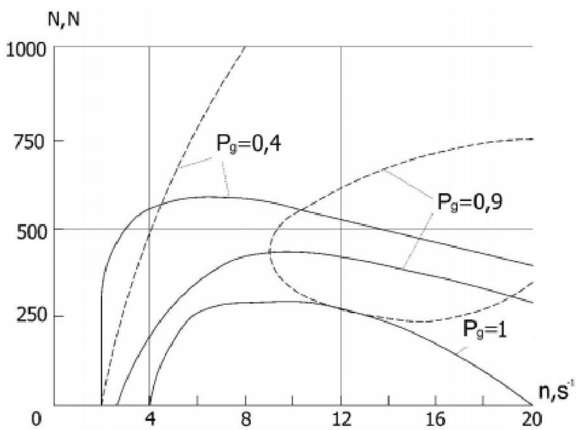

Figure 5 - Dependence of radial load *N* on shaft rotation speed *n* for various values of P_g (solid and dotted lines - [psi] is equal to 363 and 80 µm, respectively)

Figure 5 shows that the highest value of parameter P_g (0.9 or greater) for bearing # 1 can be achieved under the conditions:

$$250 \leq N \leq 750 \text{ N}; 10 \leq n \leq 20 \text{ rpm}$$

In case of a change of the load and speed operation mode beyond the indicated area, parameter P_g takes smaller values equal to or greater than 0.4 with $0 \leq N \leq 1000$ N, $2 \leq n \leq 20$ rpm.

This area includes areas where $P_g \leq 0.9$ and $0.4 \leq P_g \leq 0.9$. With that, in order to minimize wear rate of plain bearings, the load and speed operation mode with $P_g \geq 0.9$ is of the greatest interest.

Similar dependencies of parameter P_g for bearing # 2 are located in areas of lower *N* and *n* values. So, values $P_g \geq 0.9$ are obtained at $0 \leq N \leq 250$ N, $4 \leq n \leq 20$ rpm; $P_g \geq 0.4$ - at $0 \leq N \leq 600$ N, $0 \leq n \leq 20$ rpm.

In selecting the load and speed operation mode for plain bearings conditions should be created where $P_g \geq 0.9$. However, one should keep in mind that loading area is a function of technical state of plain bearings, in particular of their diametrical clearance. This area is displaced under the influence of high rotation speed and load with a small diametrical clearance to low values with increased diametrical clearance.

The obtained data are related to the intensity of a plain bearing operation according to its functional purpose as follows. Bearing # 1 in the area of $P_g \geq 0.9$ can perform more work than bearing #2. At the same time, creating light load and speed operation mode for bearing # 1will lead to reduction of this work, and at the same time to an increase in its wear rate. In order to ensure the condition of $P_g \geq 0$ 9, bearing # 2 should work in less loaded conditions than bearing # 1. Exaggeration of conditions of its operation will shift parameter P_g to the area of lower values and will increase wear rate.

Thus, location of desired areas of load and speed operation mode for plain bearings is changed depending on their technical state. In order to obtain high wear resistance due to parameter P_g, the bearing that have small diametrical clearance should be operated at high shaft rotation speeds n and high radial load *N*. On the opposite, bearings that have large diametrical clearance have to be operated at lower *n* and *N*.

Conclusions

Obtained results make significant adjustments to the traditional idea of plain bearings' operation in machines. These include changing the balance of forces in such couplings by increasing the load transmitted through the plain bearing, on the one hand, and reducing the ultimate load capacity of the lubricating film in it, on the other. Their combined manifestation is expressed by a significant exaggeration of operation conditions, which adversely affects reliability.

Obtained results form the basis of a scientific approach in defining a load and speed operation mode for operation of plain bearings, more complicated mechanical systems that contain such bearings, and for process operations for their manufacturing and repair, e.g., during running-in and testing of overhauled engines.

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