

*Теоретично показано, що для маятника у змащених опорах ковзання процес коливань описується диференціальним рівнянням другого порядку з в'язким опором, пропорційним швидкості коливань. На основі моделі коливань маятника отримано, що коефіцієнт поглинання енергії дорівнює подвійному логарифмічному декременту затухань. Визначена процедура розрахунку динамічної в'язкості мастила по декременту затухань коливань*

*Ключові слова: опора ковзання, метод маятника, затухаючі коливання, коефіцієнт поглинання, в'язкість масла*

*Теоретически показано, что для маятника в смазанных опорах скольжения процесс колебаний описывается дифференциальным уравнением второго порядка с вязким сопротивлением, пропорциональным скорости колебаний. На основе модели колебаний маятника получено, что коэффициент поглощения энергии равен удвоенному логарифмическому декременту затухания. Определена процедура расчета динамической вязкости смазки по декременту затухания колебаний*

*Ключевые слова: опора скольжения, метод маятника, затухающие колебания, коэффициент поглощения, вязкость масла*

# DETERMINING THE CHARACTERISTICS OF VISCOUS FRICTION IN THE SLIDING SUPPORTS USING THE METHOD OF PENDULUM

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## 1. Introduction

The material of machine parts possesses a property to absorb energy at deformation. This physical property of material, typically called internal friction, is closely connected with its operational performance properties: strength and wear resistance. Internal friction in the volume of material was studied well enough and its patterns are widely used in materials science when creating the materials.

The laws that govern the processes of absorbing the external energy by the surface of machine parts are explored considerably less. Processes in the surface layer are more complex than in the volume. Two forms of friction are combined here: internal and external. In this case, the processes proceed in the very small volumes of material and their study requires the application of fairly complicated methods of contact mechanics.

The methods for determining the characteristics of internal friction are divided into direct (energy, thermal, hysteresis loop) and indirect (damping oscillations, resonance curve, phase).

The testing procedures and the devices that implement the methods indicated are based on different techniques:

- the longitudinal or torsional vibrations of tubular samples;
  - torsional vibrations of solid samples;
  - the high-frequency longitudinal vibrations of a solid rod.
- The test procedures, based on the lateral oscillations of the samples, have been used most frequently:
- at the cantilever fastening of a sample;
  - at the freely hung samples;
  - in pure bending conditions.

All the known procedures are implemented on rather sophisticated installations, equipped with devices for the automatic measurement of oscillation amplitudes.

Among many methods of determining the absorption coefficient, a damped oscillation method is the easiest one. As far as the study of surface energy properties is concerned, the damped oscillation method is most effectively realized with the help of swinging pendulums.

The pendulum damped oscillation method was first employed by Newton for investigating the resisting forces to motion of a solid body in fluids and gases. The obtained results laid the foundation for hydrodynamics of laminar flow and contemporary viscosimetry.

A wide application of a pendulum method for examining the processes of friction is explained by the fact that the damping is fully predetermined by external friction under different operating conditions: materials, surface condition, temperature, composition of lubricants, layer thickness. It is important that it is possible by a change in the shape of attenuation curves to draw conclusions about the influence of these conditions on the processes of friction.

A lack of reliable procedures for determining the characteristics of viscous friction considerably complicates the task of developing the designs of friction units of contemporary machines and the optimization of energy losses during friction in order to decrease them.

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## 2. Literature review and problem statement

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A number of papers address theoretical and experimental studies into the methods for determining the energy, absorbed by a surface, including the processes of friction at damped oscillations.

By using the pendulum method, described in monograph [1], it was established that under conditions of viscous friction, oscillation attenuation curve on the elastic layer of molecules has a linear law. However, the oscillation attenuation curve on an unordered layer of lubricant is characterized by the logarithmic law of damping. These data were obtained experimentally and the study lacks a theoretical description of the mechanisms of viscous friction, which explain the results received.

Article [2] examined experimentally and theoretically free fluctuations of a physical pendulum, which rests with two balls on a flat surface. The theory of damping fluctuations proposed in the study for evaluating the energy losses is of a phenomenological nature and is not based on a strict mathematical description of the mechanisms of fluctuation process, which makes it difficult to give the quantitative assessment of losses caused by friction.

In paper [3], a differential equation for the swinging pendulum is obtained, taking into account the deformation of a surface and a deformation component of the friction force. However, the authors examined only a particular case of small deformations, commensurate with the dimensions of contact area of the ball support of a pendulum. This solution is not applicable for describing the macro-displacements in the support of a pendulum, characteristic for the actual friction units of sliding in machines.

Authors of article [4] conducted a nonlinear regression analysis based on the results of experimental studies into the damped oscillations of a physical pendulum, which made it possible to formulate the criteria for choosing a theoretical model of the dissipation of viscous friction. However, the experiments were performed for the conditions of balls rolling on the support, that is, for the small areas of contact in comparison with dimensions of the balls, which also does not make it possible to adequately utilize the obtained results for evaluating the characteristics of viscous friction in cylindrical sliding supports (bearings).

Paper [5] describes an experimental study of losses to friction in the friction pairs made of metal-polymer materials. In this case, dry friction without the use of a lubricant is examined, which is not characteristic for the cylindrical sliding supports that work under boundary or even liquid friction mode.

Article [6] explores approaches in the development of procedures for determining the characteristics of friction convenient for the practical application under operating conditions of machines. Simple formulas for calculating the coefficient of friction at reciprocating rocking movements on a pendulum device are proposed. The formulas received are not based on the differential equations, which describe the process of viscous friction, and yield only rough estimates of the indices of friction.

Paper [7] deals with the estimated-experimental procedures for determining the characteristics of friction based on a variation approach. This approach implies the use of approximating dependences of experimental data, which renders the solution approximate, depending on the subjective factors.

The experimental method for measuring the characteristics of friction was applied in article [8] for examining special features of high-speed friction under conditions of thermal loading and at limited lubrication. The criterion of damping the friction fluctuations proposed in this case is not applicable to the case of viscous friction, where such fluctuations are practically absent as a result of the damping action of a layer of lubricant.

Paper [9] employed a pendulum method in order to solve a problem on determining the contact rigidity of steel spherical models at dynamic loading using the system of differential equations. However, the tangential component of contact stresses from the frictional forces in this case was not considered.

Articles [10, 11] proposed and implemented design concepts of devices for the accelerated tests on friction and adhesive properties of the actual friction units of the “wheel – rail” type. The design of a pendulum device, applied in the work, for studying the adhesive properties according to the scheme “cylinder – plane” does not allow using it for evaluating the viscous friction of cylindrical sliding supports.

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## 3. The aim and tasks of research

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The aim of present work is a theoretical and experimental examination of the characteristics of viscous friction using the method of damped oscillations of a pendulum in the sliding supports with a lubricant.

To accomplish the set aim, the following tasks were to be solved:

- to construct a theoretical model of pendulum in the sliding supports with a lubricant using a second order differential equation with viscous resistance;
- based on the pendulum oscillation model, to develop a procedure for determining the characteristics of friction: coefficient of energy absorption at friction and coefficient of fluid friction – viscosity;
- to examine the influence of different types of lubricating and design materials on the characteristics of viscous friction.

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## 4. Theoretical model of pendulum in the sliding supports with a lubricant for examining the viscous friction

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Viscosity or internal friction is the most important physical property of oils, which ensures the separation of hard surfaces and their normal operation at minimal wear.

The tangential force  $T_{Fr}$  that occurs at relative sliding of the adjacent layers of a fluid (by force of viscous friction), is determined by the Newton law:

$$T_{Fr} = \mu \frac{dV_x}{dy} F,$$

where  $\mu$  is the dynamic viscosity;  $F$  is the displacement area;  $V_x$  is the speed in direction of the  $x$  axis;  $dy$  is the direction perpendicular to the  $x$  axis.

By dividing the force of viscous friction  $T_{Fr}$  to the displacement area  $F$ , we shall obtain the stress of viscous shear:

$$\tau = \frac{T_{Fr}}{F} = \mu \frac{dV_x}{dy}.$$

In the case of small thickness  $h$  of the layer, it is possible to accept:

$$\frac{dV_x}{dy} = \text{const} = \frac{V}{h}.$$

Thus we obtain:  $\tau = \mu \frac{V}{h}.$

Let us examine the schematic of pendulum oscillations (Fig. 1) with a support in the form of sliding bearing 1. When the gap in a sliding bearing is fully filled with a lubricant, the working area of shaft friction along the lubricant is equal to:

$$F_{Fr} = \pi R B,$$

where  $B$  is the size of bearing on the generatrix;  $R$  is the radius of bearing.

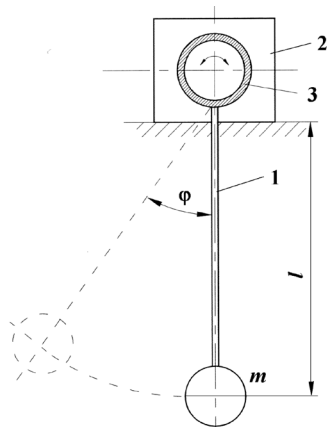


Fig. 1. Schematic of pendulum with a support in the form of a sliding bearing with a lubricant: 1 – pendulum; 2 – sliding support; 3 – oil

Shearing force over the entire circle in the bearing (Fig. 1):

$$T_{Fr} = \tau F_{Fr} = \mu \frac{V}{h} 2\pi R B.$$

If one considers that the linear sliding speed

$$V = R\dot{\phi} = R \frac{d\phi}{dt},$$

we receive

$$T_{Fr} = \frac{\mu}{h} 2\pi R^2 B \dot{\phi}, \tag{1}$$

where  $\phi$  is the angle of rotation of the shaft.

The moment of resistance to the rotation of shaft from the viscous lubricant:

$$M_{\mu} = \mu^* R^2 \phi, \tag{2}$$

where

$$\mu^* = \frac{\mu}{h} 2\pi R B. \tag{3}$$

In addition to the viscous resistance to the rotation, the shaft is exposed to two more moments:

the inertial force moment:

$$M_j = J\ddot{\phi} = ml^2\ddot{\phi} \tag{4}$$

and the pendulum weight moment:

$$M_Q = Ql \sin \phi = mgl \sin \phi. \tag{5}$$

The equation of equilibrium includes the sum of all moments:

$$M_{\mu} + M_j + M_Q = 0 \tag{6}$$

or

$$ml^2\ddot{\phi} + \mu^* R^2 \phi + mgl \sin \phi = 0. \tag{7}$$

At small oscillations  $\phi \approx \sin \phi$ , we have a classical oscillation equation with viscous resistance:

$$\ddot{\phi} + 2n\dot{\phi} + p^2\phi = 0, \tag{8}$$

where

$$2n = \frac{\mu^* R^2}{ml^2}.$$

Considering (3):

$$n = \frac{\pi\mu R^3 B}{hml^2},$$

$$p^2 = \frac{g}{l}. \tag{9}$$

Characteristic equation of the differential oscillation equation (8):

$$s^2 + 2ns + p^2 = 0$$

has two imaginary roots:

$$s = -n \pm i\sqrt{p^2 - n^2} = -n \pm ip_1;$$

$$p_1 = (p^2 - n^2)^{\frac{1}{2}}.$$

In the presence of linear roots, equations (8) take the form:

$$\varphi = e^{-nt} (c_1 \cos p_1 t + c_2 \sin p_1 t) \quad (10)$$

or, by denoting:

$$c_1 = A \cos \varphi;$$

$$c_2 = A \sin \varphi,$$

we obtain another form of the solution:

$$\varphi = Ae^{-nt} \cos(p_1 t + \varphi), \quad (11)$$

where

$$A = (c_1 + c_2)^{\frac{1}{2}}; \quad \text{tg} \varphi = -\frac{c_2}{c_1}. \quad (12)$$

Constants  $c_1$  and  $c_2$  are determined from the initial conditions (in the initial period of time the pendulum is deflected by angle  $\varphi_0$ ):

$$\varphi(t=0) = \varphi_0; \quad \dot{\varphi}(t=0) = 0. \quad (13)$$

Considering the solution (12), we receive:

$$c_1 = \varphi_0,$$

$$\dot{\varphi}_0 = c_1 n + c_2 p_1 = 0,$$

$$c_2 = -\varphi_0 \frac{n}{p_1}. \quad (14)$$

By substituting constants  $c_1$  and  $c_2$  in (12), we obtain:

$$A = \varphi_0 \left( 1 + \frac{n^2}{p_1^2} \right)^{\frac{1}{2}}. \quad (15)$$

It follows from the solution that in the presence of viscous friction the motion of load is described by the periodic damping oscillations with a period

$$T = \frac{2\pi}{p_1} = \frac{2\pi}{\sqrt{p^2 - n^2}}, \quad (16)$$

where  $p_1$  is the angular frequency of damping oscillations.

From the relation of two sequential maximum deflections of pendulum  $\varphi_k$  and  $\varphi_{k+1}$  from the position of equilibrium, we obtain:

$$\varphi_k = \varphi_{k+1} e^{nT}, \quad (17)$$

which corresponds to a geometric progression.

More frequently used in practice is the logarithm from the magnitude of ratio of the adjacent amplitudes, which are called the logarithmic damping decrement:

$$\delta = \ln \frac{\varphi_k}{\varphi_{k+1}} = nT. \quad (18)$$

If the oscillations are damped slowly and ratio  $\frac{\varphi_k}{\varphi_{k+1}} \approx 1$  is close to unity, then:

$$\delta = \ln \frac{\varphi_k}{\varphi_{k+1}} = \ln \frac{\varphi + \Delta \frac{\varphi}{2}}{\varphi - \Delta \frac{\varphi}{2}} \approx \frac{\Delta \varphi}{\varphi}, \quad (19)$$

where

$$\Delta \varphi = \varphi_k - \varphi_{k+1};$$

$$\varphi = \frac{(\varphi_k + \varphi_{k+1})}{2}.$$

Thus, at slow damping, the logarithmic decrement is approximately equal to the ratio of amplitude change during period  $T$  to the mean amplitude  $\varphi$ . We should note that viscous friction does not practically affect the oscillation frequency.

### 5. Results of determining the characteristics of viscous friction by the method of pendulum in the sliding supports

The procedures for determining the characteristics of friction are proposed based on the given theory of pendulum oscillation in the sliding support with a lubricant: coefficient of energy absorption at friction and coefficient of fluid friction – viscosity.

Pendulum is an ideal instrument for determining the energy absorbed by a surface at contact interaction. The decrease in amplitude over a cycle is connected to a decrease in the potential energy of the load that causes oscillations. The less the losses the longer the pendulum oscillates and the longer the oscillations last the more accurate the measurement of energy losses is.

The basic pattern in the oscillations of a pendulum with viscous friction takes the form of the type (17).

By definition, absorption coefficient  $\psi$  is the magnitude, equal to the ratio of energy  $\Delta P$ , lost over a cycle, to the total energy  $P$  over the cycle. Considering (19), we obtain :

$$\psi = \frac{\Delta P}{P} = \frac{2\Delta \varphi}{\varphi}. \quad (20)$$

Therefore

$$\psi = 2 \frac{\varphi_k - \varphi_{k+1}}{\varphi_k} = \frac{2\Delta \varphi}{\varphi} = 2 \ln \frac{\varphi_k}{\varphi_{k+1}} \quad (21)$$

or

$$\psi = 2 \left( 1 - \frac{\varphi_{k+1}}{\varphi_k} \right) = 2(1 - e^{-nT}). \quad (22)$$

When calculating the absorption coefficient based on the results of tests at viscous friction, it is possible to examine several limits and to approximately determine the averaged value of coefficient  $\psi$  by formula:

$$\psi = \frac{2}{N} \frac{\varphi_k - \varphi_{k+N}}{\varphi_k}. \quad (23)$$

The procedure for determining the absorption coefficient by the damped oscillations of a pendulum:

- 1) the envelope curve  $\phi(t)$  is constructed;
- 2) absorption coefficient for two adjacent amplitudes is determined by formula (22);
- 3) or, for the section on the envelope curve from  $\phi_k$  to  $\phi_{k+N}$ , the absorption coefficient is computed by formula (23).

Example of determining the coefficient of energy absorption at friction.

It is required to determine the coefficient of energy absorption in the contact between a sliding bearing and a lubricant at the following initial conditions on the pendulum:  $l=350$  mm;  $Q=2$  H;  $R=15$  mm;  $B=20$  mm;  $h=0.05$  mm.

It is found from the experiment on the envelope curve:

$$T=1 \text{ s}; \frac{\phi_k}{\phi_{k+1}} = 1,19.$$

Find by formula (24):

$$n = \frac{1}{T} \ln \frac{\phi_k}{\phi_{k+1}} = \ln 1,19 = 0,1739 \text{ s}^{-1}.$$

The absorption coefficient is determined according to formula:

$$\psi = 2 \ln \frac{\phi_k}{\phi_{k+1}} = 0,3479.$$

Based on the obtained results, we shall also examine the procedure for determining the coefficient of fluid friction (viscosity).

If, in the course of experiment, the validity of dependence of the type (17) is established, and the oscillatory period  $T$  is found, then we receive from (18):

$$n = \frac{1}{T} \ln \frac{\phi_k}{\phi_{k+1}}. \tag{24}$$

We obtain from dependence (9):

$$n = \frac{\pi \mu R^3 B}{h m l^2} = \frac{\pi \mu R^3 B g}{h Q l^2}, \tag{25}$$

where  $g=9810$  mm/s<sup>2</sup>.

The magnitude of dynamic viscosity is determined from (25):

$$\mu = \frac{nhQl^2}{\pi R^3 B g}. \tag{26}$$

At a small load, the thickness of oil film can be accepted equal to the gap:

$$\mu = \Delta = R_2 - R_1,$$

where  $R_2$  is the radius of bushing;  $R_1$  is the radius of shaft.

*Example of determining the dynamic viscosity of lubricants.*

The tests were conducted on the pendulum device whose design is shown in Fig. 2.

The suspension of pendulum 1 in the upper part was fixed to cylindrical sliding bearing 2 with disposable inserts 3 made of the examined materials. The lubricant under pres-

sure was fed through the lubricator in the bearing housing into the gap between the insert and shaft 4. Load 5 was attached to the lower part of the suspension. Indicator 7 on graduated dial 8 controlled the angle of pendulum deflection. Scale value of the dial is 1°.

Parameters of the pendulum device are as follows:

- length of pendulum  $l=350$  mm;
- weight of pendulum  $G_1=2$  H;  $G_2=4$  H;  $G_3=2$  H;
- radius of bearing  $R=15$  mm;
- width of bearing  $B=20$  mm;
- thickness of the layer of the lubricant (it is accepted equal to the gap)  $h=0.05$  mm.

During testing, we employed four types of materials of the inserts and three types of lubricants (Table 1).

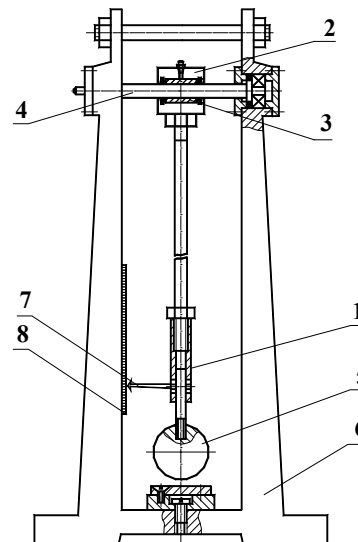


Fig. 2. Design of the pendulum device: 1 – suspension; 2 – sliding support; 3 – insert; 4 – shaft; 5 – load; 6 – housing; 7 – indicator; 8 – dial

Table 1

Results of determining the characteristics of viscosity of the lubricants

Bearing material		Viscosity characteristics at loading								
		Motor oil SAE 15W-5 (Agrinol, Ukraine)			Motor oil Formula Q8 (Q8Oils, Kuwait)			Solidol-S (Agrinol, Ukraine)		
		2H	4H	6H	2H	4H	6H	2H	4H	6H
Organic glass	N	22	26	28	18	22	26	7	11	16
	n	0.24	0.19	0.18	0.29	0.24	0.19	0.82	0.49	0.33
	$\mu$	2.83	4.48	6.47	3.43	5.65	6.71	9.66	11.6	11.6
Copper	N	13	15	18	10	13	16	5	9	14
	n	0.41	0.35	0.29	0.55	0.41	0.33	1.22	0.61	0.38
	$\mu$	4.85	8.32	10.3	6.47	9.71	11.6	14.4	14.4	13.4
Brass	N	18	24	28	14	19	22	6	12	16
	n	0.29	0.21	0.18	0.38	0.27	0.24	0.99	0.45	0.33
	$\mu$	3.43	5.06	6.47	4.48	6.48	8.48	11.7	10.6	11.6
Bronze	N	14	17	19	12	14	17	5	9	13
	n	0.38	0.31	0.27	0.45	0.38	0.31	1.22	0.61	0.41
	$\mu$	4.47	7.28	9.72	5.29	8.95	10.9	14.4	14.4	14.6

Note:  $N$  – number of oscillations to the full stop of a pendulum;  $n$  – coefficient of viscous resistance of a lubricant;  $\mu$  – dynamic viscosity of a lubricant, Pa·s

Initially, the pendulum deflected from the horizontal axis by 90°. Next, the dial (Fig. 2) registered a decrease in the deflection angle of the pendulum until its full stop. The accuracy of angle measurement reached 1° by the scale of the dial. Thus we obtained the experimental sample of values of the adjacent deflection angles of the pendulum  $\varphi_k$  and  $\varphi_{k+1}$  consisting of N values (number of oscillations until a full stop). Then we found their ratios  $\varphi_k/\varphi_{k+1}$ . The mean value of the obtained sample of values of the adjacent deflection angles was calculated by formula

$$\frac{\sum_1^N \left( \frac{\varphi_k}{\varphi_{k+1}} \right)}{N}$$

The coefficient of variation in ratio  $\varphi_k/\varphi_{k+1}$  for the entire array of tests was within the limits  $v=0.15...0.25$ . The coefficient of liquid resistance of lubricant n was calculated by formula (24) and next we calculated the value of dynamic viscosity  $\mu$  by (26). Results of the calculation of these characteristics are given in Table 1.

An analysis of the obtained results (Table 1) reveals that all tested types of design materials, when lubricated with liquid oils, demonstrated an increase in the values of the indicator of dynamic viscosity  $\mu$  at increasing load (Table 1). When lubricated with a grease lubricant, the indicator of dynamic viscosity changed insignificantly (not exceeding 10 %), which indicates a better load-bearing capacity of grease lubricants.

The obtained values of viscous characteristics of the liquid lubricants depend on the type of material, which an insert of the sliding support was made of. Viscosity varies from the lowest to the largest value for the tested oils in the following order: organic glass – brass – bronze – copper. This is related to a different adhesive degree of the lubricants and surfaces of the examined materials. The degree of adhesion can be indirectly assessed by a contact-wetting angle of the surface of materials. The adhesion is stronger with better wettability.

The contact-wetting angle  $\Theta$  was measured in the following way (Fig. 3). The drop of examined fluid 5 was applied by a pipette to the surface, then it was illuminated by strong luminous source 1 (we recommend using a point source) and projected onto screen 4. A hard surface was put up in parallel to the illuminating cone of rays so that the projection of the hard surface was projected onto the screen in the form of a thin line. The image of the drop on the screen was outlined or photographed 4.

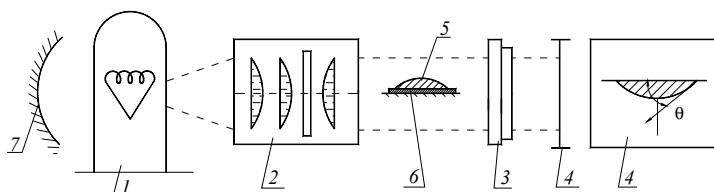


Fig. 3. Schematic of wetting angle measurement: 1 – source; 2 – condenser (illuminating lens); 3 – projection lens; 4 – screen; 5 – drop of oil; 6 – surface of material; 7 – reflecting mirror

In the point of intersection of the drop’s projection and the surface we built a tangent and determined the magnitude of contact angle.

As an example, we measured the angle of wetting the surface of samples made of organic glass, brass, bronze and copper with the motor oil SAE 15W-5 (Agrinol, Ukraine).

For statistical representativeness, each experiment on applying the drop of oil at the appropriate surface of the sample was repeated 5 times. Table 2 gives the averaged values, obtained when measuring the angle of wetting. The value of variation coefficient that characterizes the spread of obtained measurement results was within the limits  $v=0.15...0.2$ , which indicates sufficient uniformity of the experimental sample.

Table 2

Results of determining the angle of wetting

Material	Organic glass	Brass	Bronze	Copper
Contact wetting angle $\Theta$ , degrees	41	29	23	19
$\cos \Theta$	0.755	0.875	0.921	0.945

Based on the analysis of measurement results of the contact-wetting angle (Table 2), it is possible to draw a conclusion that the contact oil – copper demonstrates the best wettability and the largest adhesion. The wetting angle decreases in the following order: bronze, brass, organic glass. Adhesive properties depend on the forces of intermolecular interaction between a lubricant and a hard surface. In this case, the stronger adhesion of the molecules of oil with the surface leads to an increase in the forces of viscous resistance to the displacement of boundary layers of the lubricant (viscosity).

## 6. Discussion of results of determining the characteristics of viscous friction using the method of pendulum in the sliding supports

The obtained results allow us to define certain problems in the tribology of oil layers in contact.

In accordance with the Newton concept, a complete cohesion of surfaces occurs in the contact of solid body and fluid. In this case, the speed of points at the surface of fluid and the points at the solid body surface coincide. In this case, a solid body, when moving over the surface of a fluid, must drag along all the points of the surface with which it comes into contact.

In reality, contact points of the surface of a fluid come into contact with a solid body only in a certain limited section, and then they go out of the contact and slide down from the surface. Thus, the surface of fluid slides over the solid body surface and so the external friction occurs between the solid body and surface of the fluid. This is not accounted for by classical hydrodynamics.

Let us denote the tangential forces between the layers of fluid through  $\tau_f$ ; those between a solid body and a fluid –  $\tau_s$ .

Then two cases are possible:

- 1)  $\tau_f < \tau_s$  – sliding occurs between the layers of fluid – this is the internal friction in a fluid;
- 2)  $\tau_f > \tau_s$  – sliding occurs between a solid body and the surface of a fluid – this is the external friction.

It is obvious that these are the different types of resistance to the motion of a body on the fluid. In a general case of the motion of a solid body on a fluid, these two types of resistance are combined, and it is not easy to separate them.

At the same time, such separation is possible when conducting the specialized experiments.

In some methods of determining the viscosity, the volumetric deformation of a lubricant occurs. When determining the kinematic viscosity, the viscosimeters of variable section are used. When a lubricant passes through such a viscosimeter, the lubricant undergoes volumetric deformations in the transition to a narrow cross section and shearing strains while passing a narrow section. In this case, the number of viscosity consists of two components: volumetric deformations and shearing strains. Solving the problem on the separation of these components looks promising for the theory of lubricant.

Thus, it is possible to assume that the phenomenon of viscosity is composed of the three components:

- 1) internal friction from the shift between the layers of a lubricant;
- 2) external friction between a solid body and a lubricant;
- 3) internal friction and resistance from the volumetric deformation of a lubricant.

A lubricant possesses simultaneously the properties of a solid deformed body (frame) and a fluid (oil).

In the existing examination of mechanical properties of a lubricant, especially in thin layers, the characteristics of rigidity are not actually considered. This limits solving the problems on the contact interaction of bodies, separated by a layer of lubricant. Studying the normal and tangential rigid properties of thin films in a contact between solid bodies being deformed is one of the basic problems in the tribology of oil films.

## 7. Conclusions

1. It is theoretically obtained that for a pendulum in the lubricated sliding supports, the process of oscillations is described by a second order differential equation with viscous resistance, proportional to the deflection velocity of a pendulum. An analysis of the exact solution of this problem reveals that the ratio of adjacent amplitudes of damped oscillations is a constant magnitude. Hence, it follows that the absorption coefficient is constant over the entire process of oscillations.

2. The methods of determining the basic characteristics of viscous friction are proposed based on the model of pendulum oscillations: coefficient of energy absorption and dynamic viscosity. It is established that for the case of viscous friction, the absorption coefficient is precisely equal to the doubled logarithmic damping decrement and is determined either by one or a cycle of oscillations. The obtained method for calculating the indicator of dynamic viscosity of a lubricant according to the decrement of pendulum oscillation damping is important for the practical application.

3. Research into the influence of different types of lubricating and design materials on the characteristics of viscous friction allowed us to define certain problems in the issues of contact interaction between the lubricated materials. It is in particular established that the actual viscosity of lubricants is composed of the three components:

- internal friction from the shift between the layers;
- external friction between a solid body and a lubricant;
- internal friction and resistance from the volumetric deformation of a lubricant.

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