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Extrusion and rarefaction of lubricant in boundary layer is the key processes of adhesive wear of highly loaded tribocontacts

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Abstract

A comprehensive analysis of the Adhesion-Deformation, Elasto-hydrodynamic and Hydrodynamic friction models is presented, which describe different modes of lubrication in accordance with the Stribeck curve. The main provisions of these models are considered in conjunction with the Langmuir-BET theory of adsorption and Hertz's elastic-deformation theory of curvilinear contacts. It is shown that the revealed contradictions require their resolution, and the discovered multiple effects need a scientifically based interpretation. It is proposed to develop a more generalized model of friction and wear based on naturally occurring processes that have been hidden from direct observations for a long time. These are: Extrusion of lubricating layers in the convergent elastically deformed and Rarefaction in divergent elastically deformed regions of tribo-contacts. Understanding these processes makes it possible to predict the localization sites and causes of the occurrence of primary subsequent acts of adhesion of friction surfaces and their wear in the following cycle: “rarefaction and desorption of lubricating layers, which leads to deformation destruction of oxide films and adhesion of juvenile surface areas, after which to tearing of a fragment material from the bearing and the neoplasm of the protrusion on the shaft - in the divergent elastically deformed areas of the contact. Then microcutting by this fragment of the bearing surface occurs with the release of the wear product in the convergent elastically deformed region, which accordingly leads to a change in the actual geometry and tension of the tribo-contact. Further, in other areas of the renewed contact, adhesive interaction occurs in other divergent areas according to the same mechanism. A deep understanding of the reasons for the desorption of lubricating layers will make it possible to develop and apply new highly efficient technological and material science methods in order to increase the resource of highly loaded tribo-systems of machines and mechanisms.

Keywords: Macro-friction models; adhesive wear; pressure gradient; cavitation

1. Introduction

One of the main fundamental tasks of modern tribology is a deep and comprehensive understanding of the relationship of all regular phenomena and processes occurring in tribo-contact. Solving this problem will effectively increase the reliability and service life of civil and military equipment, an integral part of which are highly loaded curvilinear tribo-contacts of power plants and other mechanisms.

Highly loaded modern tribo-contacts are elastically deformed curvilinear discrete micro- and macro-contacts of solid surfaces with layers of active components (surfactants and others) adsorbed on them that are present in modern oils. Obviously, increasing the efficiency of their work is a complex physical-mechanical and physico-chemical problem, the key issue in which is to elucidate the role of hydrodynamic processes in a wide range of tribo-contact operation. The solution of this problem is impossible without understanding the physical model of the key processes that cause wear.

Modern tribology does not consider these key processes and is based on the three most popular models that describe the corresponding load-speed ranges and tribo-contact lubrication modes:

(1) The mode of "boundary" lubrication or boundary friction at low speeds and high loads is described in terms of the Adhesion-deformation model of friction and wear. This model covers the left descending branch of

the Striebeck curve (Fig. 1) in the friction range from starting off, at ultra-low, low speeds to medium. At the same time, the Adhesion-deformation model denies any hydrodynamic processes in the boundary layers, which are visually observed and predetermine the occurrence of quasi-dry friction conditions, adhesive interaction of friction surfaces and their wear [1–10].

(2) The mode of "liquid" non-contact friction at high sliding speeds and low loads is described by the classical Hydrodynamic friction model [11-14] without wear, which reflects the operation of a radial plain bearing in the range of the right ascending branch of the Stribeck graph (Fig. 1). In this friction mode, in contrast to the Adhesion-Deformation model, elastic-deformation processes in tribo-contacts are unfairly ignored. It should be emphasized that in modern highly loaded tribo-systems, the implementation of the hydrodynamic effect of the "oil wedge" at high loads, at low sliding speeds is unlikely and practically unrealizable. Within the framework of the Hydrodynamic model, the regular processes of extrusion and rarefaction of lubricating layers were not identified and were not taken into account.

(3) Friction under conditions of mixed or transient lubrication between "boundary" and liquid lubrication modes is described by the elasto-hydrodynamic friction model [15-19], which reflects the transition region of the Stribeck curve between the left and right branches. In the elasto-hydrodynamic model, as well as in the hydrodynamic model of friction, the tribo-system is theoretically wear-free. Within the framework of this model, the elastic-deformed state of curvilinear tribo-contacts during friction and elasto-hydrodynamic processes in the lubricating elasto-hydrodynamic layer, which is endowed with a bearing capacity during friction, are considered. However, a number of the main provisions of the elasto-hydrodynamic model directly contradict the theory of adsorption–desorption of mono- and multimolecular layers on the surfaces of solids by Langmuir and BET [20–22], and also directly contradict the fundamental physical laws of Hooke and Pascal. In particular, it is incorrect to consider the pressure in the elasto-hydrodynamic film as identical to the contact stresses, which is fundamentally wrong from the standpoint of the BET theory. Moreover, it is assumed that no hydrodynamic processes occur during friction of the elasto-hydrodynamic tribo-contact in the elasto-hydrodynamic film; the elasto-hydrodynamic film is a kind of glassy substance in a highly compressed state [15–16]. In addition, the discontinuity of the continuity of the lubricating layer is fundamentally denied in the elastohydrodynamic model, and the natural phenomenon of EXTRUSION, hidden from the eyes, was not detected and was not studied.However, cavitation that occurs in the outlet region of tribocontacts in recent decades, as an effect, has become a research trend in modern tribology [23–46].

The choice of a friction model plays a leading role in understanding the processes that determine the efficiency of the friction unit [47]. An analysis of the adhesion-deformation, hydrodynamic, and elasto-hydrodynamic friction models leads to the conclusion that each of these models lacks important information about the hydrodynamic processes that naturally occur in the elastically deformed regions of the tribocontact. These are the extrusion of lubricating layers in a converging and their extrusion in an expanding elastically deformed contact area [48].

The purpose of the work is to establish, based on the analysis of existing phenomenological models of tribo-contact friction, the relationship between elastic-deformation and hydrodynamic processes of lubricant boundary layers and the primary adhesion of friction surfaces in order to develop a promising new tribocontact friction model, which takes into account the key processes of extrusion and rarefaction of lubricating layers in elastically deformed contact areas.

The Stribeck curve (Fig. 1) — perhaps the only fundamental experimental dependence recognized in tribology, which connects the friction force with the external friction parameters (load, speed) and lubrication conditions (lubricating medium viscosity) simultaneously.

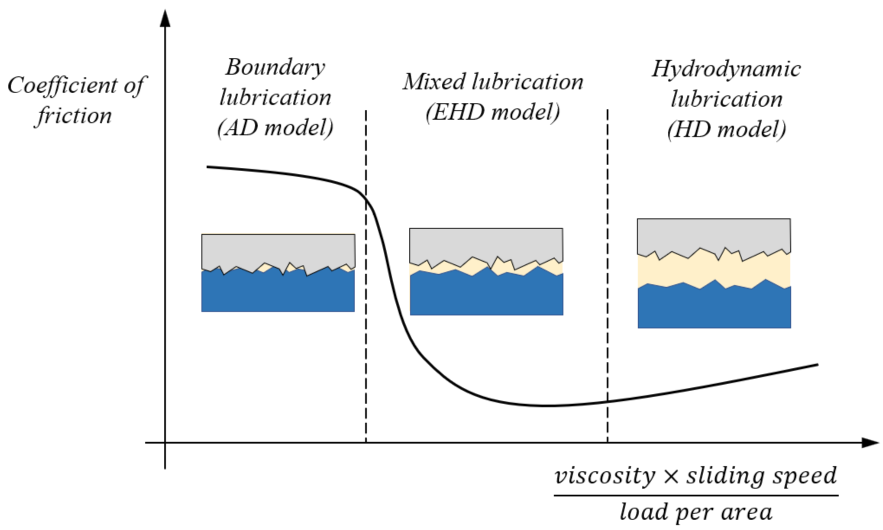

Fig. 1 Graphical illustration of different types of lubrication depending on friction modes

This remarkable dependence qualitatively illustrates the behavior of the tribo-system, covering the entire range of its friction during operation: from the starting stage of switching on to operating speeds and subsequent shutdown at each cycle of the entire machine. The generally recognized three characteristic regions of the Striebeck curve (Fig. 1) describe the corresponding modes of friction and lubrication by three phenomenological models: 1. contact "boundary" lubrication (the left descending branch of the Striebeck curve) is considered within the framework of the adhesion-deformation model of friction and wear at high loads and low speeds [2-5]; 2. Theoretically, non-contact "hydrodynamic" lubrication is described by the hydrodynamic friction model, respectively - the right ascending branch of the Stribeck curve [6-10]; 3. Theoretically, contactless "mixed" lubrication, including "boundary" and "hydrodynamic" types of lubrication in an elastically deformed friction contact at moderate speeds and loads, is described by the elasto-hydrodynamic friction model - the transition region of the Striebeck curve [15-19].

Adhesive interaction of friction surfaces with adsorbed lubricant layers and adhesive wear of surfaces, as the main negative process during friction, is considered only within the framework of the adhesion-deformation model. Adhesive interaction of surfaces, their submicro- and microsetting, the formation of secondary structures during friction, and wear kinetics are the main issues studied within the adhesion-deformation model.

Within the framework of Elasto-hydrodynamic and Hydrodynamic, the theoretically direct interaction of friction surfaces is prevented by a lubricating layer, the minimum thickness of which should exceed the sum of roughnesses. Therefore, Hydrodynamic and Elasto-hydrodynamic models are non-contact and wear-free. Therefore, the adhesion-deformation model of friction and adhesive wear of friction surfaces, which determines the resource and reliability of the entire mechanism, is of the greatest interest among designers and machine builders.

Depending on the operating mode of the tribosystem, the graph (Fig. 1) shows the contact patterns of two rough friction surfaces separated by a lubricant layer, the smallest thickness of which, in comparison with the roughness, characterizes one or another lubrication mode during friction. However, the minimum thickness of the lubricating layer in tribocontact is not reliably determined by direct measurement methods today, which prevents the establishment of clear boundaries between friction modes, types of lubricant and, accordingly, between the adhesion-deformation, elasto-hydrodynamic and hydrodynamic models that describe them.

Thus, the same bearing, depending on the friction rate, load and rheological properties of the lubricating medium (viscosity), is considered from the standpoint of these different phenomenological models and the corresponding basic provisions: from contact boundary friction at low speeds and high Amonton-Coulomb contact loads [47] to hydrodynamic non-contact friction at high speeds and low loads (classical hydrodynamic friction model with lubrication by O. Reynolds [11].

In practice, all critical and highly loaded friction units, which are usually enclosed in one lubrication system (lubrication systems for internal combustion engines, gas-turbine engines, transmissions and other power plants and mechanisms), as well as the entire machine, are periodically stopped and cyclically restarted for the duration of operation. to perform the following tasks. After parking for different durations in different conditions, when starting the machine, starting from the moment of starting, with acceleration, bypassing ultra-low speeds, all friction units reach operational speed, load and thermal conditions. When stopping, each tribosystem goes back: from operational load-speed and thermal modes of operation to a complete stop.

It turns out that completely different friction models are applied to the same tribosystem with lubrication operating in different speed, load and thermal conditions: Adhesion-Deformation, Elasto-hydrodynamic and Hydrodynamic, which have a limited area of use and do not have clear boundaries and applicability criteria. This indicates the need to search for those underlying dominant and key processes that occur during friction that have not come to the attention of tribologists in order to create a more complete and universal model of friction and wear. This work is aimed at such a search, and the presented results and conclusions can sufficiently meet the criterion of scientific novelty of this article.

2. Analysis of theoretical positions, experimental phenomena and effects of Adhesion-Deformation, Hydrodynamic and Elasto-hydrodynamic models of friction.

The left descending branch of the Stribeck curve (Fig. 1), when friction occurs at ultra-low-low-medium speeds and high loads, reflects the so-called friction mode under "boundary lubrication" conditions, which is described by the Adhesion-Deformation model of friction and wear (Bowden, Tabor, Kragelsky, Demin and others [1-10]). The beginning of the curve on the Stribeck diagram corresponds to the maximum value of the friction force that occurs at the moment the tribosystem leaves the state of rest at the very beginning of the relative movement of the surfaces at the points of contact - "rest friction". The law of friction, originally formulated by Leonardo da Vinci in 1508 in an implicit form as "Friction requires doubling the effort if the weight is doubled", 200 years later Amonton was proposed in a quantitative form [47]:

$$F_{friction} = f \cdot N \tag{1}$$

where the friction force $F_{friction}$ is directly proportional to the load or downforce (normal reaction) N, multiplied by the dimensionless friction coefficient *f*.

In 1875, Coulomb proposed a refinement of formula (1):

$$F_{friction} = A + f \cdot N \quad (2)$$

where A - a constant component that occurs in the process of adhesion microseizure of surfaces and is one of the reasons for the mismatch between static friction and sliding friction, that is, the presence of a jump at the beginning of sliding (Fig. 1).

The adhesion-deformation or molecular-mechanical model of friction is based on the concept of the dual nature of friction and the corresponding two-term friction force formula, which is essentially the Amonton-Coulomb law (Eq. (2)), where A is the adhesive or molecular component of the friction force, and the second term $\left(f \cdot N \right)$ reflects the deformation or mechanical component.

The study of the contribution of each of these components of the friction force to its total value according to the Eq. (2), carried out by V. Hardy [50] showed the following relationship: $\left(f \cdot N \right) : A = 1:10000$, while I. V. Kragelsky [1,10,50] values this ratio as 1:100. Consequently, the adhesive interaction of friction surfaces, which leads to adhesive wear, generates the most significant 99% component of the total friction force A, the reduction of which is the most important task of boundary friction tribology. Noteworthy is the absence in Eq. (1) and Eq. (2) of the friction velocity $\boldsymbol{\upsilon}$, which is the main condition for the friction process itself, as well as the compression force N.

2.1 Adhesion-Deformation model of friction and wear

The main provisions of the Adhesion-Deformation model:

(1) The main position of the Adhesion-Deformation model of friction and wear is based on the concept of the discreteness of the actual contact of rough elastically deformed surfaces compressed by an external force, which are visualized by the classic schemes (Fig. 2(a)). Since the surfaces of real solids always have irregularities, the nature of which depends on the material of the contacting surfaces and the method of their processing, the contact of such surfaces is discrete at relatively low compression forces. It is believed that in the elastic contact of the rough surfaces of a friction pair, only a small part of the protrusions of these surfaces is elastically deformed - up to 0.001 fraction of the contour area according to [2].That is, the contact area of the surfaces is a set of locally loaded curvilinear quasi-point and quasi-linear microcontacts, which, during friction, interact with each other according to the deformation and adhesion mechanisms, dynamically replacing one contact zone with another.

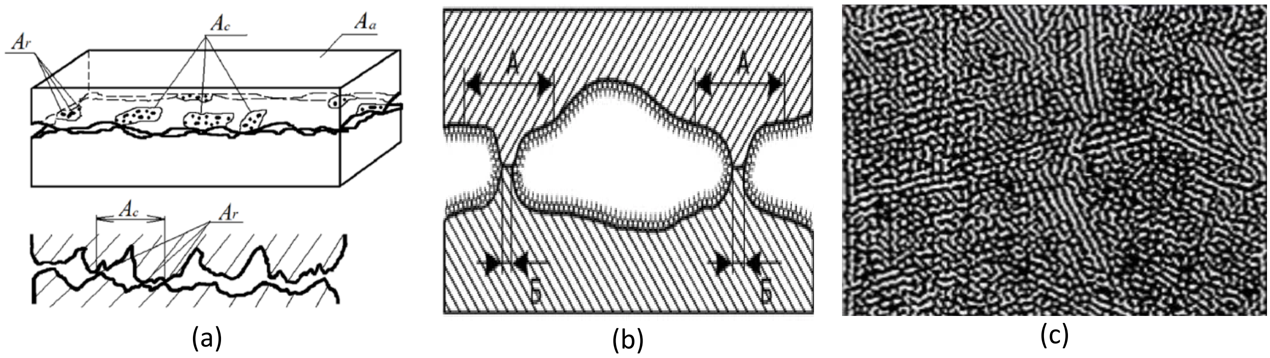

Fig. 2. Classical schemes of the adhesion-deformation model of friction of surfaces based on the concept of discreteness of actually contacting protrusions of real rough surfaces: 3D-scheme and 2D-scheme (I.V. Kragelsky, A.M. Demkin) [1,2] - (a); a schematic representation of the appearance of mutually elastically deformed sections A and the formation of adhesion sites of setting B after the destruction of the boundary monomolecular lubricant layer on the protrusions of rough surfaces (Bowden) [3] - (b); cellular structure of nanorough surfaces when the vertices merge into closed cells like honeycombs along the contour (dark areas) with multimolecular lubricant layers inside the contours (light areas) [48] - (c).

During the friction of such elastic discrete contacts of two surfaces with layers of lubricant adsorbed on them, between the contacting areas of the surfaces, adhesive interaction forces arise in the form of welding bridges (Fig. 2(b)) [3,5], which manifest themselves as microseizure, microseizing, tearing of fragments of a less durable material and neoplasms on a more durable one, followed by their cyclic cutting and formation of wear products.

This position is limited to reflect the contact of highly rough surfaces or the contact of polished surfaces under low compressive load. The fact is that the Adhesion-Deformation model of friction and wear originated in the USSR in the first half of the last century in 1939 (A. M. Ertel [15]) and was further developed in the USSR

(A. N. Grubin, 1947; A I. Petrusevich, 1951; Kragelsky, 1965 [1,16,18]), and in England (D. Dawson 1963, 1966, Bowden 1968, Tabor 1978) [3]. It should be noted that, at that time (1930-1970s), the metalworking industry was at the initial stage of its development and, accordingly, the parts of friction units were manufactured with large errors and deviations, and the surfaces themselves had a relatively large roughness with the parameter R_a = 0.25 ... 0.5 microns and more.

At the same time, a hypothesis arose about the process of running in new parts as a self-formation of a certain "optimal roughness" in the process of running in machines and mechanisms (at present, a fashionable name is often used - self-organization). Its essence lies in the fact that smooth polished friction surfaces should not be made, for example, a shaft and a radial plain bearing, since during their running-in the roughness will increase due to adhesive destruction with characteristic tears in the bearing and subsequent microcutting. Conversely, a large shaft roughness during friction will somewhat decrease due to deformation shearing, microcutting and adhesive wear. Losses for such self-formation of optimal roughness during running-in are inevitable - increased friction and high wear intensity with known negative consequences at the initial stage of product operation.

Currently, in the course of the revolutionary development of machine tool building, materials science, metalworking technologies, as well as methods for finishing friction surfaces in production, nanometer roughness is achieved with a parameter of R_a<0.1…0.02 mkm and even with Ra<10 nm. A constant increase in contact stresses and an increase in the nanometer quality of surfaces leads to a logical conclusion that with an increase in the compression force of the initially discrete contact areas of nanoprotrusions, their elastic deformation occurs and the dimensions of these actual compression areas will naturally increase. Therefore, the generally recognized discrete model of contacting highly polished nano-rough surfaces at constantly increasing contact stresses (for example, 1000 MPa and more in aviation tribo-contacts) should be supplemented with logical processes for increasing not only their actual contact areas, but also their natural and inevitable merging with the subsequent emergence of common for two surfaces of closed cavities (Fig. 2(c)).

That is, under static compression, in the course of deformation of the protrusions of surfaces with monomolecular layers adsorbed on them, closed contours will appear, inside which there are cavities common to two surfaces, such as many lakes with multimolecular layers of lubricant of different amounts (two, three, and more) according to the type of Fig. 2(b)) and 2(c)). Therefore, precision engineering is of particular interest to the behavior of boundary layers of lubricant between highly polished nano-rough surfaces with monomolecular boundary layers located on them with a comparable nanometer thickness both under static compression and friction.

(2) Within the framework of the Adhesion-Deformation model, the definition of the process of adhesive wear is given as a consequence of the atomic-molecular interpenetration of juvenile contact areas of friction surfaces with subsequent tearing out and transfer of a material fragment from one surface to another and its subsequent cutting with the formation of a free wear particle [1-6].

A prerequisite for the occurrence of adhesive wear is the absence of any films on friction surfaces such as oxides, secondary structures, including mono- or multimolecular lubricant layers [48]. It is believed that in real friction units, adhesive wear always manifests itself under the condition that the rate of attrition of various films on the friction surface exceeds the rate of its recovery due to the physical adsorption of the lubricant, or chemisorption, or oxidative processes.

Depending on the extensiveness of the contact of molecular interaction, adhesive wear is distinguished at the microlevel and macrolevel. At the micro level, the process of molecular adhesion develops on elastically deformable rubbing protrusions of microroughness of rough friction surfaces. Depending on the strength of the emerging molecular bonds, destruction is observed both in thin near-surface layers and deep tearing of the metal. In this case, processes of material transfer from one friction surface to another often occur, for example, technological methods of copper plating or friction brassing. The metal particles torn out and stuck to the friction surface are partially cut off or peeled off in subsequent interactions, forming adhesive wear products. The composition of the wear particles is in most cases identical to the base material. Adhesive wear at the macrolevel manifests itself in friction units in the form of seizing, deep tearing and seizing. This type of wear, as a rule, manifests itself at high contact stresses, with large tangential and normal forces in the contact zones, leading to desorption of the boundary layers.

The intensity of wear during adhesive wear is quite high, and the process itself is accompanied by high values of the friction coefficient, its oscillations with large amplitude and frequency, as well as significant heating of the parts of the friction unit.

(3) The third position is related to the ideas about the structure and properties of the boundary layers of lubricant on friction surfaces: the surfaces of solids are always in interaction with the environment, the nature of which depends on the physicochemical properties of the material of the solid and the medium. Boundary layers always appear on all lyophilic surfaces of solids wetted with a lubricating hydrocarbon medium, which traditionally contains both surfactants and dissolved molecules of environmental gases (from 8 to 12%) [49].

Within the framework of the Adhesion-Deformation theory, it is believed that mono- and multimolecular layers of lubricant with a thickness of 0.01...0.10 μm [1-10] are always formed on the oxide films of the friction surfaces of real parts according to the mechanism of physical and chemical adsorption. It is believed that when the boundary layers of the lubricant are strongly compressed by two surfaces, they acquire the properties of some kind of amorphous “glassy” or “third body” [16-17]. These properties are different from what they would have in an unlimited volume of lubricant. It is also generally accepted that the thinner the layer, the higher its elasticity. The strength of such layers in compression is very high, their modulus of elasticity exceeds the modulus of elasticity of structural materials of friction surfaces and even diamond, which is noted in the works of V. Hardy [50] and A.S. Akhmatov [10]. A schematic representation of the process of static compression of surfaces with such multimolecular layers of an Epitropic- Liquid Crystal structure [51] is shown in Fig. 3.

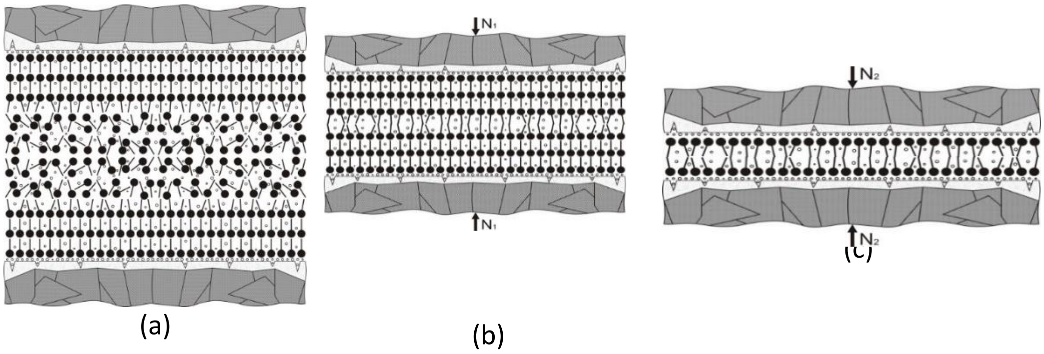

Fig. 3. Schemes of the structure of Epitropic-Liquid-Crystalline oil layers, taking into account active molecules adsorbed on the surfaces (plates with circles) and gases dissolved in oil (transparent circles) between two surfaces with a gap (a); layer-by-layer desorption of supra-monomolecular layers during their static compression by force N1 (b); as well as a diagram of the uniaxial stress state of monomolecular layers adsorbed on friction surfaces at higher compression loads N2 >> N1 (c) [10,50,51].

Fig. 3 shows a diagram of the appearance of contact between surfaces and monomolecular layers during compression in accordance with the concept of the epitropic liquid crystal structure of boundary layers of oils on lyophilic friction surfaces [51]. Such a representation is consistent with the theories of Langmuir and BET, namely: all multimolecular liquid crystal layers located above the monomolecular layer behave like fluid fragments of a liquid in a volume. Accordingly, when these fragments are deformed, they flow into the bulk phase in the region of lower pressure, under laboratory conditions, these are regions with ambient pressure - atmospheric pressure p_0 .

(4) Since the monomolecular lubricant layer is capable of withstanding, without collapsing, significant compression pressures [10], it is believed that at ultra-low friction velocities and high loads in the contact zone of surfaces with boundary layers of lubricant adsorbed on them, there are no hydrodynamic processes.

This dubious position is too categorical, it is not sufficiently substantiated, and here's why. In accordance with the well-known Langmuir and BET adsorption theories [20–22], monomolecular and polymolecular lubricant layers on lyophilic friction surfaces indeed have a high bond strength with the active centers of the surfaces and high elasticity under static compression. However, as extensive research results show, during friction with relatively small loads and low speeds, areas of adhesive setting of surfaces always appear, material is pulled out, which obviously must be preceded by the destruction of monomolecular adsorbed lubricant layers, that is, their desorption. Therefore, it is quite logical that the first act of adhesive interaction of surfaces with lubricant layers adsorbed on them or primary adhesion during friction of lubricated surfaces should be preceded by desorption of both multi- and monomolecular layers.

There is a main hypothesis about the tangential force destruction of lubricating layers at the tops of discrete contacts of rough surfaces under the action of tangential shear stresses [1-3] (Bowden, Tabor, Kragelsky, etc.), which is experimentally insufficiently confirmed. Moreover, A.S. Akhmatov in his famous work “Molecular physics of boundary friction” [10] focused on the fact that no one has reliably observed the destruction of boundary layers by such a tangential mechanism and is a very controversial point of view. Another hypothesis of thermal desorption of boundary layers [9] is based on the concept of local temperature flashes that occur at discrete contact areas during friction and can reach 600...1200°C. However, in the initial period of starting from a place, when primary adhesion of friction surfaces occurs, such flashes are not observed and are also insufficiently confirmed experimentally.

(5) The maximum contact stresses are calculated according to the Hertz formulas [56-57] without taking into account the roughness so that they do not exceed the yield strength of the material. In practice, this situation is extremely rare, in real friction units, as a rule, the maximum contact stresses of tribo-contacts are an order of magnitude less than the yield strength of structural materials.

Since the primary adhesive interaction of the surfaces of solids and their wear is possible only if there are no multi- and monomolecular layers of lubricant on them, as well as any oxidizing [4] and other films, it is obvious that the primary adhesive interaction of "juvenile" surfaces friction must be preceded by desorption of the boundary layer of the lubricant [58-59].

The cause-and-effect relationship between desorption of boundary layers and adhesion of friction surfaces is a key moment for understanding the mechanism of the occurrence of primary, secondary, and so on acts of adhesion in local areas of contacting friction surfaces.

Primary adhesion when the shaft rubs against the plain bearing leads to tearing out of a fragment of material from the surface of the bearing, then the formed build-up (new formation) on the shaft is cut off when it comes into contact with the formation of a free particle - a wear product, and the remaining build-up on the shaft is micro-cutting the bearing. Thus, it becomes obvious that it is the desorption of the boundary layers that causes the occurrence of conditions for local quasi-dry friction, as a process preceding the primary adhesive setting of the friction surfaces.

Primary adhesion leads to an instantaneous change in all contact conditions and violation of all initial friction conditions, which sharply reduces the possibilities of modeling within the Adhesion-Deformation model. And if we take into account the fundamentally random places of localization of adhesion during friction, as well as the fundamentally random behavior of the friction force due to adhesive wear, then the modeling of tribosystems becomes very far from practice.

Thus, from the above analysis of the Adhesion-Deformation model, it follows that a clear understanding of all the phenomena and patterns in the relationship that determine the desorption of lubricating layers during friction under boundary lubrication conditions is extremely necessary. The mechanism of the appearance of primary adhesion during friction of lubricated surfaces is an extremely urgent problem of modern tribology, the solution of which will largely determine the success in creating advanced technology for various purposes. Prevention of primary adhesion of friction surfaces requires a deep understanding of all the elementary processes that occur during friction in combination to create effective technological methods in the manufacture of tribosystems in order to reduce or completely prevent their wear during long-term operation.

(6) Adhesive wear is based on the concepts of molecular adhesion of friction surfaces or "welding", therefore it is quite obvious that this process must be preceded by the destruction of sufficiently strong mono- and multi-molecular lubricant layers, as well as oxidative layers of surfaces. Consequently, the organization of adsorption of boundary layers of lubricant on friction surfaces, the creation of frictional contact, the friction of boundary layers and their desorption, followed by adhesion of surface materials, are successive fast processes that can be represented as Fig. 4 shows:

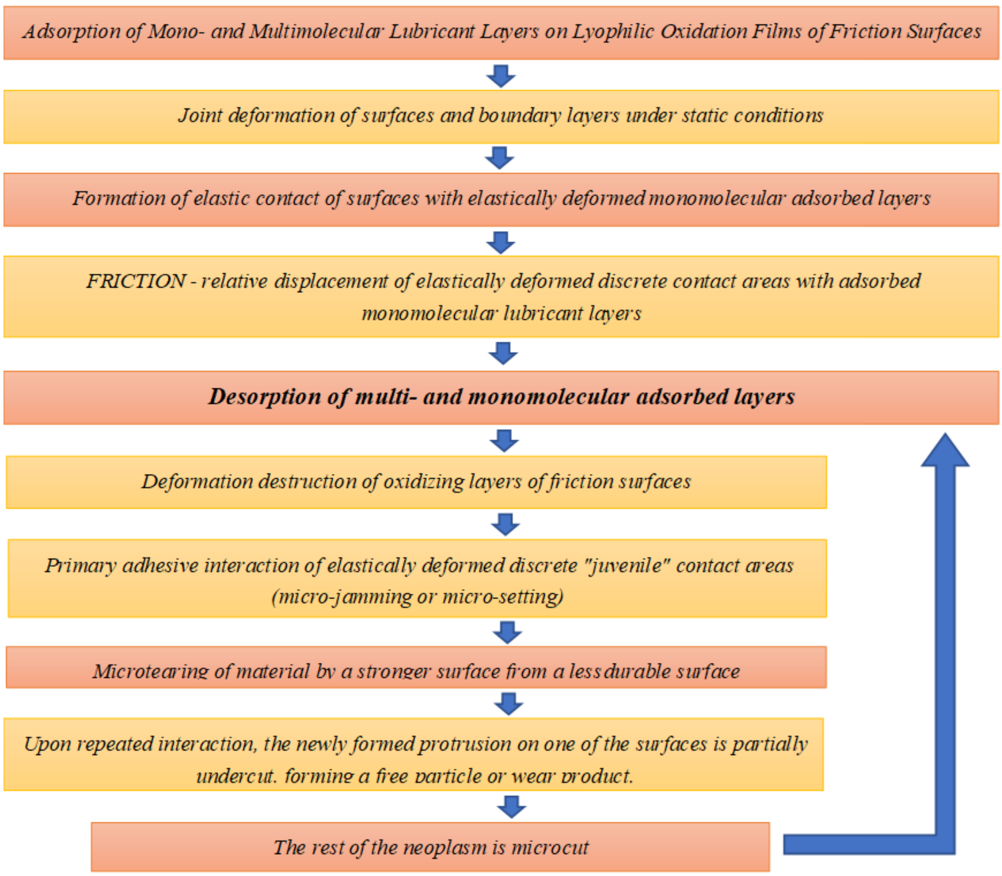

Fig. 4 Process of the destruction of lubricant layers and oxidative layers on surfaces

According to the authors, unreasonable ignoring the occurrence of any hydrodynamic processes during friction of surfaces with sufficiently strong mono- and multimolecular layers of a lubricating medium adsorbed on them at ultra-low and low speeds is the main obstacle to the further development of the Adhesion-

Deformation model.

Meanwhile, the Adhesion-Deformation model of friction and wear covers a number of well-known theories, phenomena and effects, among which the following stand out: the theory of oxidative wear of friction surfaces by B.I. Kostetsky [4], "The phenomenon of selective transfer of copper" or "the effect of wearlessness" and the theory of hydrogen wear by D. Garkunov and others [52-53], "The phenomenon of abnormally low friction" by Silin and others [54]. Also of interest is the effect of abnormally high wear resistance of a sliding friction pair: a polished shaft - a highly rough bearing (I.V. Kragelsky) [1]. The question of why rubbing a polished shaft against a highly rough lubricated bearing results in less wear on the latter than rubbing against a similarly polished surface also needs to be explained. In addition, the mechanism of the regular behavior of tribocontacts when starting from a place, when at the moment of the beginning of the movement of richly lubricated surfaces the static friction force significantly exceeds the motion friction force, is not fully disclosed, which also needs a deeper experimental and theoretical substantiation and interpretation.

The main drawback of the Adhesion-Deformation model of friction and wear is the denial of hydrodynamic processes in the boundary layers during friction. Therefore, there is no relationship between the elastic-deformation processes of curvilinear discrete micro-contacts and hydrodynamic processes in the lubricant adsorbed on them by mono- and multimolecular boundary layers during friction. Below [2-5] it will be shown that at the slightest displacement of such contacts, the extrusion of boundary layers immediately occurs in narrowing (converging in the direction of friction - the input contact area) and their rarefaction in expanding (divergent in the direction of friction - the output contact area) elastically - deformed contact areas.

2.2. Hydrodynamic model of lubrication without wear

The right ascending branch of the Striebeck curve (Fig. 1), when friction occurs at high sliding speeds in a viscous medium, and with a small radial load *N*, is referred to as a hydrodynamic or "liquid" friction mode, when smooth surfaces are completely separated by an oil layer of thickness *h* due to the occurrence it has a sufficiently high hydrodynamic pressure and the so-called "oil wedge" effect is realized. Such a friction mode is described by the Hydrodynamic friction model [11-14], the classical scheme of which is presented on the example of a radial plain bearing (Fig. 5). In accordance with the Hydrodynamic model of friction, surface wear is not considered in principle.

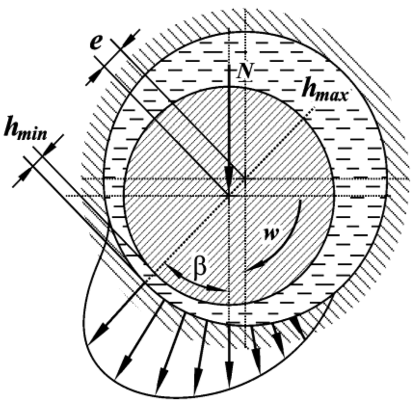

Fig. 5. Classical representation of the Hydro-Dynamic model of friction of a radial plain bearing, which illustrates the occurrence of pressure in the lubricating film (pressure diagram is shown as radial arrows, reflecting the effect of the "oil wedge"), at high shaft speeds ω, at low or moderate radial load *N* , where it is shown how the rotating shaft "floats up" and shifts in the direction of friction by the eccentricity *e*, and the axis of the shaft centers, on which the minimum $h_{\min}$ and maximum $h_{\max}$ clearances of the radial bearing are located, is shifted by an angle β.

Friction within the framework of the Hydrodynamic model is described by the equation of O. Reynolds (1886) [11] using various boundary conditions and algorithms in order to determine the load capacity of the lubricating film in the minimum gap between the rubbing surfaces $h_{\min}$. At the same time, the friction coefficient has significantly lower values compared to the "boundary" friction mode within the framework of the Adhesion-Deformation model. The basic equation of the Osborne Reynolds Hydrodynamic theory allows you to determine the pressure distribution in the lubricating film along the gap of a radial plain bearing according to the formula:

$$\frac{\partial p}{\partial x} = 6\mu\upsilon \frac{\left(h - h_{\min}\right)}{h^3} \tag{3}$$

where x – the coordinate in the direction of the shaft movement, h – the value of the current clearance, $h_{\min}$ – the minimum clearance between the shaft and the bearing; p - the pressure in the oil film, μ – the viscosity of the oil, υ – the peripheral speed of the shaft sliding along the bearing [11].

Noteworthy is the absence of load N in Eq. (2), which is usually taken into account through the minimum thickness of the lubricating film $h_{\min}$. In this case, any deformation of the surfaces is also not taken into account, as well as their stressed state: the bearing capacity of the lubricating film under pressure is estimated by the excess of the pressure in it, multiplied by the friction area, over the value of the radial load applied to the bearing. The main criterion of the Hydrodynamic model of friction is the value of the minimum thickness of the lubricating layer in the bearing clearance $h_{\min}$, which should not exceed the sum of the roughness of the shaft and bearing surfaces $\left(R_{a1}+R_{a2}\right)$. Unfortunately, today the minimum thickness of lubricating layers $h_{\min}$, as the most important criterion for their non-contact bearing capacity, is not reliably measured and in most cases is a hypothetical speculative value.

2.2.1 The main provisions of the Hydrodynamic model of friction of lubricated surfaces

(1) The gap of the radial bearing is completely filled with lubricant, there are no end flows of the lubricant, the viscous lubricating layer implements a laminar layer-by-layer Newtonian flow. O. Reynolds voiced this position at the very beginning of his work, analyzing the Beauchamp Tower experiments on the instructions of the Institute of Mechanical Engineers in 1883-1884[11-14];

(2) The surfaces do not come into contact, since the minimum thickness of the lubricating layer $h_{\min}$ exceeds the sum of the surface roughness $\left(R_{a1}+R_{a2}\right)$, therefore, wear of the surfaces is not assumed and is not considered;

(3) The bearing implements friction immediately at sufficiently high speeds, which is contrary to practice: all friction units, like all machines, go through the natural stages of starting off, accelerating to operating speeds and operating for the required time, followed by stages of deceleration to a complete stop.

(4) When the shaft rubs against the radial bearing, the pressure in the lubricating layer **p(x)** always exceeds the ambient pressure p_0;

2.2.2 Actual experimental and theoretical problems of the Hydrodynamic model of friction

(1) The mechanism of occurrence of the hydrodynamic effect of the “oil wedge” has not been fully disclosed, probably due to the lack of reliable and complete information about the regular extrusion flows of boundary layers, which O. Reynolds mentioned at the very beginning of his well-known work [11];

(2) Adhesive interaction of surfaces, their wear, as well as their contact, the Hydrodynamic model is not provided, although in practice all bearings wear out. It is believed that wear occurs only at the starting and stopping stages of operation. This assumption has not been experimentally proven;

(3) Calculations of the bearing capacity of lubricating layers based on Eq. (3) are not widely used in the design bureaus of machine-building enterprises - new friction units are created by improving tribosystems of previous generations using new structural and lubricant materials, coatings and additives, as well as by improving existing manufacturing technologies;

(4) In the main equation of hydrodynamic lubrication of the Hydrodynamic model (Eq. (3)), there is a sliding speed, medium viscosity, minimum thickness of the lubricating layer, but there is no load, which, like speed, is the second main condition for the implementation of the process of friction of solids. The absence in Eq. (3) of the value of contact stresses σ(x) of the surfaces in the area of the elastically deformed state of the contact $\left[-b;+b\right]$, which are set by the designer and are quite well calculated by the formulas of G. Hertz, is the main problem of the Hydrodynamic model. The fact is that bearings with a small contact load in modern mechanical engineering are of no practical value, just like a bearing without relative movement of compressed surfaces, that is, without speed (Amonton-Coulomb Eq. (2)).

(5) Problem of “negative pressure loop” by O. Reynolds and half Sommerfeld solution (half solution of A. Sommerfeld of hydrodynamic friction).

In the well-known works of O. Reynolds, when describing Fig. 10 [11, p. 275], it is noted that in the direction of rotation of the shaft relative to the minimum gap filled with oil, with a flat surface, two symmetrical regions arise: narrowing to the minimum gap - convergent, and subsequent expanding - divergent. O. Reynolds notes that "... if the load (on the liner with a rotating shaft) continues to increase, then the pressure at point A ([11, Fig. 1, p. 256]) decreases or even becomes negative." “Since the amount of negative pressure that the oil can withstand depends on unexplained reasons, then the ultimate load that provides complete lubrication should be considered one at which the smallest distance between the pin and the liner is equal to half the difference in radii.” Further, O. Reynolds notes: “fluid pressure on the right side removes the surfaces” (we are talking about the convergent area), but since “the pressure on the left side is negative” (divergent gap area), then “a moment is obtained that rotates towards the surface" shaft. Further, when considering Fig. 11 in [11], it is indicated that "the pressure is everywhere positive ... therefore, the pressure everywhere tends to push the surfaces apart." In further reasoning, starting from the description of Fig. 12, O. Reynolds, based on the experiments of B. Tower, only a few times mentions "negative pressure in the lubricating layers", which then leads to "rupture of the oil layer" [11], p. 284, it is interpreted as follows: "whether this negative pressure occurs or not depends on whether the lower end of the insert is completely immersed in the oil bath or not. Usually, such a case does not occur. If this happens, it still remains unclear how far the negative pressure can go ... we can only say that it is not lower than atmospheric pressure. Such reflections of the great O. Reynolds regarding the "negative pressure" in the divergent areas have not received a clear interpretation to this day.

Brilliant calculations of the pressure in the lubricating layer of a radial plain bearing by A. Sommerfeld [11] also indicate the occurrence of a "negative pressure" in the lubricating film in the divergent contact area. However, the categorical, initially introduced by O. Reynolds, the main position of the Hydrodynamic model of friction about a completely filled gap between the trunnion and the bearing, led to the beautiful centrally symmetric pressure distribution curve in the lubricating layer of A. Sommerfeld, with positive and "negative" values, the scientific community has recognized only half of this curve, where the pressures are always not "negative", that is, greater than "zero". Then the "negative" part of the local pressure distribution curve in the lubricating layers was equated to constant values of ambient pressure, that is, to atmospheric pressure.

The problem of the "negative friction loop in the lubricating layer" of Reynolds-Sommerfeld (according to A. Cameron [55]) remains relevant, as well as the accepted terms "half Sommerfeld solution" [11].

Looking ahead, the authors believe that the resolution of these issues lies in the absence of a connection between the stress state of the surfaces and the local pressure in the lubricating layers between them with the ambient pressure p_0. The fact is that the microvolume state of lubricating layers of real droplet liquids in contact under compression is naturally characterized by flow properties due to their practical incompressibility. Also, in accordance with the theories of adsorption by Langmuir and BET [20–22], under external influence, namely, when surfaces with multimolecular boundary layers are compressed, which is characteristic of friction, all over monomolecular layers of adsorbed liquid molecules (the second, third, and so on layers) behave like a liquid. Only a monomolecular layer is capable of withstanding colossal compressive loads, since their modulus of elasticity is commensurate with the modulus of elasticity of the strongest structural materials. But under conditions of local rarefaction, all lubricating layers, including the monomolecular layer, in accordance with adsorption-desorption isotherms, can easily evaporate from surfaces - that is, desorb, which causes the conditions of quasi-dry friction and primary adhesion.

(6) The phenomenon of asymmetric wear observed and described by A. Sommerfeld on the bearings of wheel pairs of trains after their long-term operation [11] lies in the fact that the greatest destruction and wear of radial plain bearings in the vast majority of cases occurred in the area where the shaft came out of contact, that is in the divergent area. When examining the bearings of the axle boxes of locomotives in the Wittenburg repair shops after their long work, I visually observed some regularity in the wear of the plain bearing shells. In the work "On the theory of friction during lubrication" [11], Sommerfeld wrote: "Out of 20 bearings, 16 were worn more in front (in the direction of rotation, that is, in the divergent contact area), only two were worn more in the back (that is, in the convergent), while for the remaining two liners, the position of the place of greatest wear was unclear. In another series of cases, when examining 20 bearings, the results were as follows: 14 were worn more in front, 5 more in the back, and for one liner, the position of the point of greatest wear was indeterminate. A. Sommerfeld attributed this interesting fact to the confirmation of the hypothesis of a significant decrease in the lubricating layer in the divergent area of the plain bearing shells. The mechanism of this phenomenon and the reasons for its occurrence have not been disclosed to this day.

2.2.3 The main disadvantages of the Hydrodynamic friction model

(1)Denial of wear of the friction surfaces in the implementation of the hydrodynamic effect of the "oil wedge". The authors believe that adhesive wear occurs during fluid friction, as well as in other modes of friction during lubrication, only with a lower intensity.

(2)The minimum thickness of the lubricating layer $h_{\min}$, as the main criterion of the Hydrodynamic model, has not been reliably measured to this day, so the model itself remains more theoretical.

(3)Lack of connection between the elastic-deformation state of tribo-contact surfaces and hydrodynamic processes in lubricants adsorbed by mono- and multimolecular boundary layers on friction surfaces.

It will be shown below that all tribo-contacts are subjected to elastic deformation, and especially highly loaded ones, in which, at the slightest displacement in the boundary layers of the lubricant, extrusion immediately occurs in the narrowing and rarefaction in the expanding elastically deformed contact areas.

2.3 Elasto-hydrodynamic friction model without wear

The transition area of the Stribeck curve (Fig. 1) between the Boundary (left branch) and Hydrodynamic (right branch) modes of friction with lubrication is described from the standpoint of the Elasto-hydrodynamic friction model [15-19], which is presented in the form of a scheme that has become classical (Fig. 6). This model arose as a result of an attempt to combine the Adhesion-Deformation and Hydrodynamic models into a certain generalized model and is very popular in modern tribology.

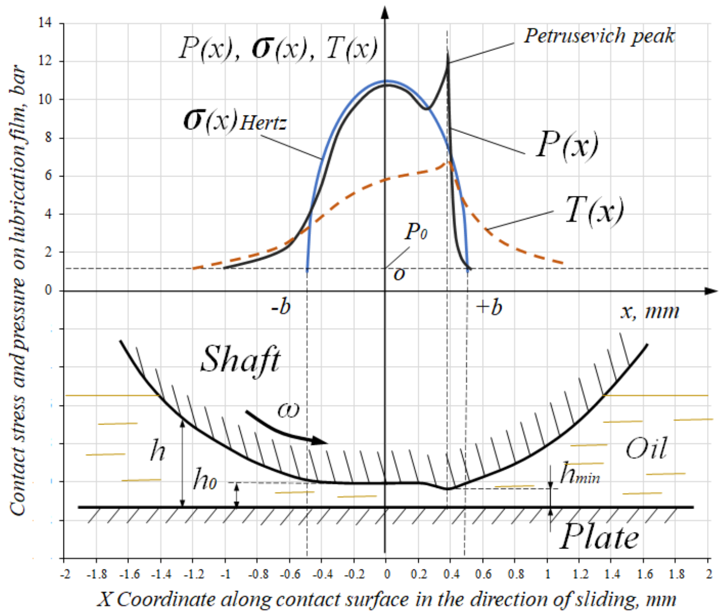

Fig. 6. Scheme of the Elasto-hydrodynamic friction model, where its main provisions are reflected: 1. The elastically deformed surfaces of the curvilinear contact are parallel; 2. The lubricating layers are strongly compressed from above and below by parallel compressing surfaces and have a constant thickness h_0; 3. The distribution of pressure in the lubricating film p(x) (EHL pressure) is everywhere greater than atmospheric p_0 and is identical to the contact stresses σ(x) according to G. Hertz, that is, p(x)=σ(x); 4. Contact stresses in the output region have a "Petrusevich peak", where the temperature T is maximum, and the thickness of the lubricating film is minimum $h_{\min}$, which should exceed the sum of surface roughness $\left(R_{a1}+R_{a2}\right)$; 5. Direct contact of the surfaces is not provided, therefore friction according to the Elasto-hydrodynamic model is theoretically wear-free.

The elasto-hydrodynamic friction model of Grubin, Ertel, Petrusevich and others [15-19] has become a classic and is presented in the form of a diagram (Fig. 6) in numerous textbooks on tribology for undergraduate and graduate students of higher technical educational institutions in the specialties "Tribology", "Tribo-technics", "Engineering".

2.3.1 The main points on which the Elasto-hydrodynamic model is built

(1) According to the scheme (Fig. 6), in curvilinear axisymmetric surfaces during their compression, the Hertzian contact stresses σ(x) are distributed axisymmetrically in the form of a semi-ellipse with respect to the vertical axis of symmetry OY. The half-width of the elastically deformed contact b, the distribution of contact stresses σ(x) of elastically deformed curved surfaces under the action of an external compressive force N within the contact width $\left[-b;+b\right]$, as well as the maximum contact stresses σ_{max} are calculated using the well-known formulas of G. Hertz. As applied to the Timken scheme "a model cylindrical shaft with radius R and length **l** – a flat surface", these formulas have the following form:

$$\sigma\left(x\right)=\sigma_{\max}\sqrt{1-\left(\frac{x}{b}\right)^{2}} \tag{4}$$

Where the maximum design contact stresses of elastically deformed curved surfaces σ_{max} under the action of an external compressive force σ_{max} within the contact width $\left[-b;+b\right]$ without taking into account roughness are calculated by the following formula:

$$\sigma_{\max}=0.418\cdot\sqrt{\frac{2\cdot E_{red}\cdot N}{l\cdot R}} \tag{5}$$

The half-width of a line contact is given by the following formula:

$$b = 2.15\sqrt{\frac{NR}{lE_{red}}} \tag{6}$$

where in Eq. (5) and Eq. (6) the reduced modulus of elasticity E_{red} is determined by the formula:

$$E_{red} = 2\frac{E_1(1-\mu_1^2)E_2(1-\mu_2^2)}{E_1(1-\mu_1^2)+E_2(1-\mu_2^2)} \tag{7}$$

where E_1 –the modulus of elasticity of the shaft material, E_2 - the modulus of elasticity of the material of the flat model bearing according to the Timken scheme, μ_1 and μ_2 – the Poisson's ratios of the shaft and flat bearing materials, respectively. Such formulas are of tremendous importance and are widely used in the strength departments of all design bureaus of modern aviation and other types of mechanical engineering.

According to Hertz, the main parameters of a stress-strain curvilinear contact are: contact half-width b, maximum contact stresses $\sigma_{\max}$ and their distribution $\sigma(x)$ in the form of a semi-ellipse. At the same time, it should be remembered that these parameters are calculated with a number of assumptions, such as: the surfaces are perfectly smooth, which is practically not achievable; contact is realized in absolute vacuum without any substances on them, which is also practically impossible.

Created on the basis of G. Hertz's formulas, modern software packages for calculating the stress-strain state of tribocontacts are practically not connected with any of the above-mentioned Adhesion-Deformation, Hydrodynamic and Elasto-hydrodynamic models due to the great inconsistency of their main provisions. In fact, calculation software packages, for example, for rolling bearings, are based on formulas obtained exclusively empirically, which include many coefficients that reflect both the influence of real roughness and the influence of the lubricating medium, temperature and other operational factors. Such software packages are of enormous value and are the subject of KNOW-HOW of well-known leading companies in the world of mechanical engineering. Therefore, these modern methods for calculating tribosystems are constantly being improved and still need to correctly reflect all the real processes in the lubricating film during friction of elastically deformed tribocontacts, which are not taken into account either in Adhesion-Deformation, or in Hydrodynamic, or in Elasto-hydrodynamic models of friction.

(2) The lubricating layer is limited by parallel elastically deformed surfaces and has a constant thickness h_0, and the pressure in the compressed lubricating layer p(x) in any contact area is constant and is always greater than atmospheric p_0;

(3) The lubricating film under the action of high (more than 400 MPa) contact stresses (according to Grubin) becomes a super viscous, “glass-like”, or “amorphous substance” [15-16], that is, not fluid, in which the pressure is identical to the contact stresses $p(x) \sim \sigma(x)$. Hydrodynamic processes in this so-called Elasto-hydrodynamic film are considered one-sidedly: only from the standpoint of pressure increases in the Elasto-hydrodynamic film, which determines its carrying capacity by balancing the load;

(4) Since, during friction, the gap h_0 of curved compressed surfaces is parallel to the plane, accordingly, the thickness of the compressed lubricating film is also practically unchanged with a slight thinning to $h_{\min}$ (Fig. 6) at the exit of the surfaces from the contact, which is explained by its heating during friction and a decrease in effective viscosity in this area.

(5) It is generally accepted that with increasing load, the area of the curvilinear contact with increasing contact width $\left[-b;+b\right]$ increases, the distributed contact stresses $\sigma(x)$ and their maximum values $\sigma_{\max}$ decrease, and therefore the bearing capacity of the lubricating layer in the Elasto-hydrodynamic film, increases.

(6) As in the Hydrodynamic model of friction, within the framework of the Elasto-hydrodynamic theory, it is assumed that the surfaces at different speeds never come into direct contact due to small surface roughness, the sum of which $\left(R_{a1}+R_{a2}\right)$ is less than the minimum thickness of the Elasto-hydrodynamic film $h_{\min}$. In this case, the lubricating medium is considered as a Newtonian homogeneous fluid. Accordingly, the practically observed wear of the surfaces is associated only with the starting modes of friction when starting from rest, which is described by the Adhesion-Deformation model of friction and wear. At subsequent stages of friction with acceleration at ultra-small, low-medium sliding speeds up to operational friction speeds, wear of the surfaces theoretically does not occur.

2.3.1. Problematic issues of Elasto-hydrodynamic friction models

(1) The Elasto-hydrodynamic model, like the Hydrodynamic model, is a model of wear-free friction. The Elasto-hydrodynamic models take into account the stress state of surfaces in contact one-sidedly from the

standpoint of an increase in the contact area due to elastic deformation and an increase in pressure in the lubricating Elasto-hydrodynamic film, which is incorrectly identified with contact stresses. Other processes, such as rarefaction, reverse flows and end lateral overflows, are not considered.

(2) Identification of the pressure in the Elasto-hydrodynamic film and the contact stresses is a very doubtful statement, since, to date, physical scanning and measurement of local pressure in the elastically deformed region of the tribo-contact during friction in dynamics has not been performed by anyone, with the exception of [48].

(3) The minimum thickness of the lubricating layer $h_{\min}$, which is the main parameter of the tribocontact and a criterion in the Elasto-hydrodynamic model, as well as in the Hydrodynamic model, is not reliably measured today, but is speculatively postulated in a certain range from fractions to tens of micrometers.

(4) It is noteworthy that in the fluid friction mode, in accordance with the Elasto-hydrodynamic and Hydrodynamic models, the surfaces do not wear out, however, operating practice indicates the opposite: all friction surfaces wear out and it is not a fact that adhesive wear occurs only in starting modes, which is described within the Adhesion-Deformation model.

2.3.2. Critical analysis of the main provisions of the Elasto-hydrodynamic friction model

The Elasto-hydrodynamic model arose as a result of a search for a model that would cover the entire range of bearing operation from ultra-low to high speeds and loads in an elastically deformed contact area and under different lubrication conditions. In the course of the analysis of the main provisions of the Elasto-hydrodynamic model, contradictions are found with the fundamental physical laws of Hooke's deformation of solids, Pascal's hydrostatics, and the theories of adsorption by Langmuir and BET [20–22].

Thus, according to the Elasto-hydrodynamic model, in the region of maximum contact stresses (Fig. 7), the elastic deformation of the surfaces is the same as on the periphery of the contact, where the stresses are very small. This contradicts Hooke's law, according to which the amount of deformation is directly proportional to the applied force, in our case, contact stresses. Consequently, the curvilinear tribo-contact, in accordance with Hooke's law, is clearly inhomogeneous (in the center, where the greatest stresses are, the elastic deformation will be significantly higher than in the edge regions. Therefore, the gap profile will obviously be axisymmetric in the form of a fillet tapering towards the middle on the left and right (Fig. 7(a)).

When choosing the direction and the beginning of friction, it is also obvious that three elastically deformed characteristic regions will appear (Fig. 7(b)): a narrowing or confusing elastically deformed region, a transition elastically deformed region and an expanding or diffuse elastically deformed region. Since adhesive wear occurs only in elastically deformed regions (wear in non-contact regions with a gap is unlikely), it is these regions that require close attention to determine the causes and patterns of primary adhesion. Having answered the questions: in which of the three elastically deformed regions (confusing elastically deformed region, diffuse elastically deformed region or transition elastically deformed region) and why does primary microadhesion occur, we will be able to counteract it more consciously and effectively.

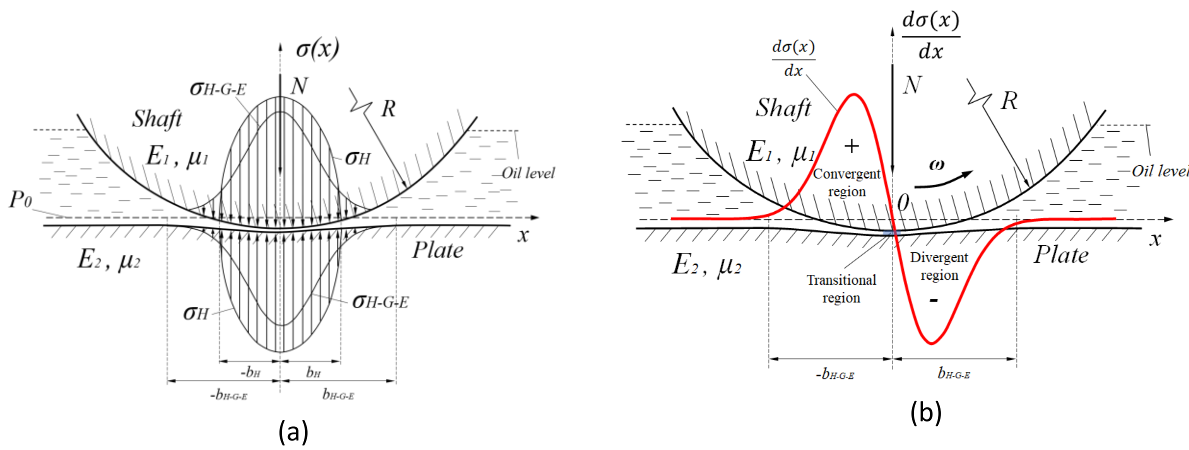

Fig. 7. Scheme of the formation of an elastically deformed linear contact of the shaft surface with a flat model bearing

Scheme of the formation of an elastically deformed linear contact of the shaft surface with a flat model bearing under static compression (without friction) with adsorbed layers of lubricant in the form of a fillet, as well as diagrams of contact stresses in the form of a semi-ellipse according to Hertz $\sigma^{H}(x)$ and in the proposed by the authors exponential distribution of Hertz-Gauss-Euler $\sigma^{H-G-E}(x)$ relative to the ambient pressure p_0 on the corresponding elastically deformed surfaces with the width of the curvilinear contact $[-b_H;+b_H]$ and

$\left[-b_{H-G-E};+b_{H-G-E}\right]$ (a). Gap profile in the form of a fillet and the appearance of three characteristic regions in an elastically deformed contact: convergent (confusing elastically deformed region), where $\frac{d\sigma(x)}{d(x)}>0$ transitional (transition elastically deformed region), where $\frac{d\sigma(x)}{d(x)}\sim 0$ and divergent (diffuse elastically deformed region), where $\frac{d\sigma(x)}{d(x)}<0$ at friction with shaft rotation frequency ω – (b).

Within the framework of the Elasto-hydrodynamic model, it is assumed that the thickness of the lubricating film in a curvilinear contact is constant and has the same value, which already contradicts the mono- and themselves as liquids. In accordance with this, even with static compression of surfaces with lubricating adsorbed layers, all supra-monomolecular layers will be squeezed out and flow into areas with lower pressure, and their microvolume pressure will tend to the oil pressure in the volume, in accordance with their main property - fluidity and Pascal's law. Therefore, in the case of a curvilinear contact, fragments of the boundary layers of the lubricant under the action of an external compression load in the convergent region realize extrusion. Looking ahead a little, we say that the extrusion of the lubricating layers will occur only in the narrowing convergent elastically deformed area of the tribocontact, where the lubrication fragments will be squeezed out in the opposite direction of friction and flow along the shortest path to the area where the oil pressure in the volume is minimal, which under normal conditions corresponds to atmospheric p_0.

In addition, within the framework of the Elasto-hydrodynamic model, a number of experimentally established effects were found that do not find their unambiguous explanation:

(1) Thermal effect of Murch and Wilson in the input area of the contact [19], which means that during friction of an elastically deformed curvilinear contact, the highest temperature occurs in front of the contact. The causes and mechanism of this effect have not been fully elucidated.

(2) Peak Petrusevich - the peak of the increase in contact stresses that occurs during friction in the output region of the elastically deformed contact, is associated with thinning of the lubricating layer in the "rear half of the contact", that is, in the output contact zone. This peak was observed at stresses of at least 200–300 MPa and is explained by its author [17-18] by the fact that the lubricating layers, passing through the contact zone, pass into a “quasi-plastic” or “amorphous” state with special properties of a certain “third body” according to Grubin [16]. In the input and central contact areas of the Elasto-hydrodynamic, the lubricant film heats up, its thickness decreases, and the contact stresses increase in the form of a peak, not exceeding the maximum values of the contact stresses determined by the formulas of G. Hertz (4-7) [17];

(3) The hydrodynamic effect of the “oil wedge” is also not fully disclosed, and such a main criterion for the bearing capacity of a lubricating Elasto-hydrodynamic film, such as its minimum thickness $h_{\min}$ has not been reliably measured for more than 100 years.

(4) Cavitation in lubricating layers of Elasto-hydrodynamic films, over the past 60 years, has become another effect in lubricating layers during friction, which has literally become a real trend in tribological research [23-46, 60-61]. During this time, the tribological literature on cavitation in lubricating layers during friction presents extensive results, most of which are aimed at determining the role and influence of cavitation bubbles on the bearing capacity of lubricating layers. Many researchers believe that frictional cavitation negatively affects the bearing capacity of lubricating Elasto-hydrodynamic films in accordance with the postulates of the Elasto-hydrodynamic model and can lead to adhesive wear. In widely known and popular works on this topic with extensive reviews and in-depth analysis of previous studies [60-61], as well as in reports from various research laboratories, friction cavitation is mostly explained from the standpoint of classical hydrodynamic cavitation at very high friction velocities. (700-15000 min^{-1}) [60-61]. A number of researchers associate the phenomenon of cavitation with dynamic oscillatory processes of surfaces in the radial direction due to the existing natural deviations of the shaft from the ideal geometric shape relative to the axis of rotation, which leads to oscillatory-pulsating radial loads per shaft revolution. The role of cavitation in tribological contacts has been discussed by many authors in relation to the bearing capacity of the lubricating film, namely its effect on the continuity and thickness of the Elasto-hydrodynamic lubricant film to determine the role of cavitation in frictional losses and wear of plain bearings [23-46].

2.4. The effect of cavitation during friction

The vast majority of works on cavitation during friction are in the nature of mathematical models and algorithms with different boundary conditions, by improving the Reynolds equations, to calculate the bearing capacity of the lubricating Elasto-hydrodynamic film in the gap of a radial plain bearing, taking into account cavitation phenomena. One of the very first works that led to the emergence of the cavitation direction in tribology was the work of Floberg [24, 60-61]. Many works devoted to the calculation of plain bearings and optimization of their parameters were aimed at determining the field of hydrodynamic pressures under the Swift-Stieber boundary conditions by integrating the generalized Reynolds equation. However, the application of the Swift-Stieber boundary conditions led to an imbalance in the consumption of the lubricant entering the lubricant layer from the source and flowing into the bearing ends. Therefore, the Jacobson–Floberg–Olson (JFO) boundary conditions became more correct, and the first efficient algorithm that indirectly included the JFO boundary conditions was developed by Elrod and Adamson [27] in 1974.This algorithm is based on the finite difference method (FDM) and divides the solution region into two - the active region, where there is no cavitation and the lubricant completely fills the gap, and the cavitation region, where the lubricating layer breaks. The division into active area and cavitation area is done by using the original switching function. The flow in the cavitation region was considered as two-phase, consisting of liquid and gas (vapor) phases, with a uniform density, and the flow in the active region was considered as compressible with a constant modulus of elasticity.

Despite the fact that Elrod's algorithm is successfully applied in the theory of lubrication, the problems of convergence of calculation results remain unresolved [28]. Many researchers have worked and continue to work on improving the Elrod algorithm, applying it to different tribo-systems. So in 1986, Brewe [29] applied the Elrod algorithm to the study of dynamically loaded plain bearings. Brewe used an implicit alternating direction method, then, together with Woods [30], applied multigrid methods to increase the speed of convergence of the calculation results.

In modern mechanical engineering and, in particular, in engine building, in order to increase the resource, studies of the influence of deviations of the micro- and macro-geometry of friction surfaces from the ideal geometric shape are widely used. In addition to the structurally specified, macro-deviations are associated with natural errors in the processing of friction surfaces, load and thermal deformations, as well as friction and wear processes during the operation of tribocouples in the composition of the product. Therefore, Vijayaraghavan and Keith [31-32] modified the Elrod algorithm using different finite difference schemes for two flow regions, which improved its numerical stability, which was applied by Qiu and Khonsari [23] for textured thrust bearings. Qiu and Khonsari used a multi-grid method with Seidel's iterative scheme, and later they [33] applied the algorithm in conjunction with Patir and Cheng's method to investigate the effect of roughness within textures.

Also in 1991, Kumar and Booker [34-35] proposed an algorithm for transients optimized for the finite element method, which was applied by the authors of [36] to textured surfaces, taking into account roughness. Other FEM-based models have been developed by Shi and Paranjpe [37], Hajjam and Bonneau [38].Another approach by Gherca et al. [39] was implemented to study textured flat bearings under stationary and transient conditions. The cavitation algorithm was developed by Payvar and Salant [40] in 1992. Based on Elrod's theory, they developed a finite difference version of the algorithm with optimized numerical stability. Their model was adapted for mixed lubrication by Wang et al. in 2003 [41] using the method of Peter and Cheng as a rough surface contact model under real misalignment conditions in radial and thrust plain bearings. In [42], the authors used the Payvar and Salant algorithm to develop a universal Reynolds equation that allows one to simultaneously calculate macro- and micro-cavitation in plain bearings with rough surfaces. The Payvar and Salant algorithm is being applied to study lip seals with mixed lubrication [43], textured thrust bearings, and textured seals with rough surfaces [44-45]. Xiong and Wang [46] carried out a detailed analysis of the numerical implementation of this model by the finite volume method.

It should be noted that almost all of the above works [23-46,60-61] on cavitation in the lubricating layers of plain bearings are of a fundamental theoretical nature, and it is too early to talk about their applied use.

About the probable occurrence of cavitation in lubricating layers during rolling friction, in the well-known work of D. Klamman (1988) [62], on page 50, only a hypothesis was put forward that cavitation may occur in the rolling contact, which can lead to the destruction of surfaces with scaly petal look and pitting. These surface damage in the form of petals externally look like flakes raised in the rolling direction precisely in the areas where the surfaces of the rolling bodies roll out, which we also observed, that is, in the divergent areas. Then the scientific editor of the translation sharply criticized this hypothesis: "this statement is the personal opinion of the author (D.Klamman) and is not confirmed by official science (tribology)."

To a lesser extent, experimental works have been published with a detailed description of the methodology for conducting experiments and tangible results, which makes it difficult to reproduce them. Thus, a visible violation of the continuity of the lubricant flow when the shaft slides along a glass radial bearing in the zone of minimum clearance, that is, cavitation in the form of air gaps between oil streams, was observed as early as 1920 [63] by Hyde. A glass bearing and fluorescent lubricating oil were used by Cole and Hughes [64] in the study of the "negative pressure loop" of O. Reynolds in the area of the shaft exit from the contact, where the pressure reached 35...70 kPa (at atmospheric pressure approximately 102 kPa). A.Cameron (1962) [55] compared such pressure with real surface contact stresses, which usually amount to tens and hundreds of MPa and are two orders of magnitude greater than the pressure in the lubricating layer. Therefore, A. Cameron concluded that these negligibly small quantities can be neglected.

Cavitation in lubricating media during friction has also become the subject of many inventions and patents for methods and devices for its detection in the USSR, the USA, and China [65–67]. However, the main questions: where and why does cavitation occur during friction and how is it related to the primary adhesive wear today remain unanswered. The main drawback of the Elasto-hydrodynamic friction model is the absence of a relationship between the elastic-deformation processes between the contact surfaces and the hydrodynamic processes in the mono-multimolecular boundary layers of the lubricant adsorbed on them during friction. The hypothesis about the Elasto-hydrodynamic film as some kind of amorphous substance is not valid, since it directly contradicts the BET theory and Pascal's law. It will be shown below that the elastic deformation of surfaces and especially highly loaded tribocontacts at the slightest displacement immediately leads to the appearance of extrusion in narrowing and rarefaction in expanding elastically deformed contact areas.

2.5 Adsorption

All three characteristic areas of the Stribeck curve (Fig. 1) for real highly loaded friction units have a common condition - the obligatory presence of lubricating layers on the friction surfaces. Therefore, the above analysis of the main friction of Adhesion-Deformation, Elasto-hydrodynamic and Hydrodynamic models should be supplemented with information about the surface adsorption of lubricants as the main process that ensures the very presence of boundary lubricant layers on friction surfaces.

Adsorption, as a process of concentration of a substance in the boundary layer at the contact boundary of different phases, as applied to tribology, covers two groups of phase separation - such as "solid - liquid" and "solid - gas". The last group is not considered within the framework of this article, due to the fact that we are talking about highly loaded friction units, which cannot work for quite a long time without a lubricant in real technology. At the same time, the friction surfaces are an adsorbent, and surfactants are an adsorbate, which are always present in a dissolved form in modern lubricants, which are adsorbents.

It is known that the adsorption of a lubricant boundary layer on friction surfaces, depending on the nature of the acting forces, proceeds through two main mechanisms, which are distinguished as physical adsorption and chemisorption. During physical adsorption between the molecules of the adsorbate and the active centers of the adsorbent, van der Waals forces act mainly, which have a dispersion, orientation, and induction character. The peculiarity is that these forces act between molecules or atoms that are in different phases. Physical adsorption, in contrast to chemisorption, is reversible and is most characteristic of lyophilic friction surfaces of real parts when they interact with modern lubricants, in which the most active adsorbate is surfactant molecules. At the same time, one cannot ignore the formation of secondary structures on friction surfaces in lubricating media as a result of chemisorption, which play an important role in the friction process and especially during long-term operation of tribo-systems.

The theory of adsorption of a monomolecular Langmuir layer [20–22] on a friction surface describes well the dependence of the adsorbate concentration on pressure using adsorption isotherms with increasing pressure. Obviously, as the pressure on the surface decreases, the adsorption isotherms become desorption isotherms. The BET multilayer theory of adsorption (Fig. 8) includes five main types of isotherms, which, depending on pressure, make it possible to predict the rate of adsorption-desorption and the thickness of adsorption layers, depending on the nature and properties of both the lubricating medium and the surface on which adsorption occurs. The modern literature contains tens of thousands of adsorption-desorption isotherms obtained for a wide variety of solids and liquids. However, most of the physical adsorption-desorption isotherms according to the IUPAC classification are based on the classical adsorption isotherms from I to V (BET), and according to the classification first proposed (1935-1940) by S. Brunauer, L. Deming , W. Deming and E. Teller (BDDT) already have them VI. These types of isotherms are shown in Fig. 9. Type I characterizes adsorption on surfaces with micropores, which are close to real rough friction surfaces. Type II and III isotherms are characteristic of macroporous materials often used in plain bearings. Isotherm types IV and V, which have a hysteresis loop, reflect capillary condensation in narrow gaps of near-contact elastically deformed sections of discrete protrusions of friction surfaces with adsorbed layers, that is, near actual contacts with a microrelief that is always present during compression of friction surfaces. The type VI isotherm describes the layer-by-layer filling of surfaces with adsorbate molecules and is extremely rare, for example, when nitrogen is adsorbed on the surfaces of some types of activated carbon. Common to all types of adsorption isotherms is that when the pressure is lowered, the opposite to the adsorption process occurs - the desorption process, when lowering the pressure to ultra-low values leads to the fact that the concentration of adsorbate molecules tends to zero (Fig. 9).

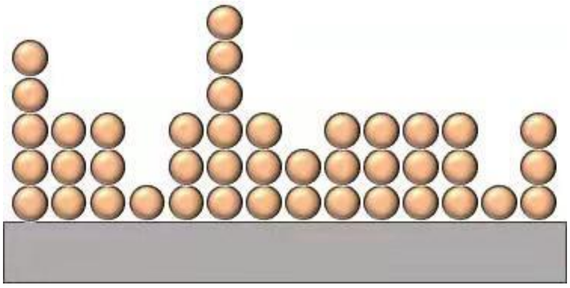

Fig. 8. BET multilayer adsorption model or random distribution of surfactant oil molecules (adsorbate) on the friction surface (adsorbent) in the form of circles included in one, two, three, and so on layers.

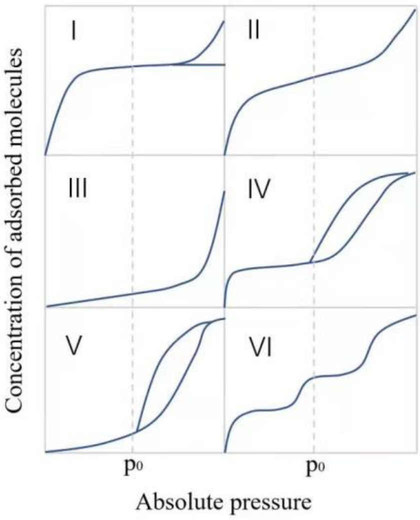

Fig. 9. Types of adsorption-desorption isotherms according to IUPAC and BDDT classification.

Another of the most important provisions of the BET theory for tribology is the conclusion that all supra-monomolecular adsorbed layers (second, third, etc.) under external force, including static compression, which is typical for all tribocontacts, lead themselves like liquid molecules. Thus, in the case of static compression of two surfaces with lubricating layers under the action of contact stresses, only the monomolecular layer is able to perceive and transmit uniaxial compression loads, and all other layers located above it (above the first monomolecular layer) will flow into areas of reduced pressure in accordance with the fundamental Pascal's hydrostatic law (Fig. 5).

However, in practice, during long-term operation of precision friction units, such as spool, plunger and other friction pairs with a roughness of Ra <0.02 μm, for example, fuel or hydraulic equipment with small contact stresses in contact of friction surfaces (6 ... 20 MPa), such tribosystems are often subject to adhesive micro-setting, seizing, jamming and wear. Therefore, naturally, the question arises why monomolecular layers on nano-rough surfaces, which under static conditions withstand high compression loads, are destroyed by friction. The mechanisms of desorption of monomolecular adsorbed lubricant layers on surfaces are experimentally little studied, not fully disclosed, and therefore this issue is an extremely urgent task for modern tribology. It should be emphasized that only in the BET theory there is information about the high sensitivity of multi- and monomolecular adsorbate layers to pressure reduction. So, all isotherms of the dependence of the adsorbate concentration on the adsorbent surface on pressure (Fig. 9) are also desorption isotherms, which indicate that with a local and rapid decrease in pressure at the adsorbent surface, the desorption of the adsorbate layers proceeds very intensively.

Thus, it is obvious that it is the desorption of multi- and monomolecular layers that is the key process that precedes the adhesive interaction of surfaces and adhesive wear. But the processes leading to the destruction of boundary layers and their desorption during friction are not fully understood.

2.6 General conclusions from a comprehensive analysis of the Adhesion-Deformation friction and wear model, Hydrodynamic and Elasto-hydrodynamic models of friction with lubrication and the elastic-deformation theory of solids by H. Hertz, the law of elastic deformation of solids by Robert Hooke and the law of hydrostatics by Blaise Pascal

Summing up the above analysis of Adhesion-Deformation, Hydrodynamic and Elasto-hydrodynamic friction models, the authors came to the conclusion that each of these models lacks some very important information about regular and objectively occurring processes in the elastically deformed regions of the tribocontact. These are the extrusion of the lubricating layers in the narrowing and their VACUUM in the expanding elastically deformed contact areas.

If each of the known Adhesion-Deformation, Hydrodynamic, and Elasto-hydrodynamic models is supplemented with regular and obvious processes of extrusion and rarefaction in combination with the adhesive interaction of working surfaces, then a generalized model will very likely arise, for example, the model of friction and wear proposed in [48].

Conclusions

1. The presented comprehensive analysis of the Adhesion-Deformation, Elasto-hydrodynamic and Hydrodynamic friction models, which describe different lubrication regimes during friction in different load-velocity ranges, in combination with the Langmuir-BET adsorption theory and the elastic-deformation theory of

curvilinear Hertz contacts made it possible to identify the following main contradictions: within the framework of the Adhesion-Deformation model of friction and wear, hydrodynamic processes in the boundary layers of the lubricant at low speeds and high loads are unlawfully denied; in the framework of the Hydrodynamic and Elasto-hydrodynamic friction models, the elastic-deformation state of the tribocontact is considered one-sidedly, only from the standpoint of a geometric increase in the contact area, which increases the bearing capacity of the Elasto-hydrodynamic lubricant film. At the same time, the wearlessness of friction surfaces is postulated unconvincingly and unjustifiably at low loads and at moderate to high friction velocities. At the same time, the minimum thickness of the lubricating film, which is the main criterion for non-contact and wear-free friction, is in fact not currently determined or measured by direct methods.

2. The revealed contradictions between Adhesion-Deformation, Hydrodynamic and Elasto-hydrodynamic models can be resolved if we take into account naturally occurring processes that have been hidden from direct observations for a long time - Extrusion of lubricating layers in convergent elastically deformed and Rarefaction in divergent elastically deformed regions of tribocontacts. Thus, a real opportunity arises to create a new, more complete model and theory of friction and wear, where extrusion and rarefaction of boundary layers play a dominant role in the occurrence of quasi-dry friction conditions that cause adhesive wear of various tribocontacts, including highly loaded tribosystems.

3. The Hydrodynamic and Elasto-hydrodynamic models of friction are mostly of a theoretical speculative nature, since they do not touch upon the topical issues of wear and service life of tribosystems. However, the practice of operation and the personal experience of the authors testify to the opposite: adhesive wear of surfaces of lightly and highly loaded tribosystems always occurs during friction: in all lubrication modes and in the entire range of loads and speeds, just the wear intensity will naturally be different.

Nomenclature

N axial load of the shaft on the plain bearing, N

h(x) respectively, the gap between the sliding friction surfaces of the shaft and the bearing in the current coordinate (x) in the direction of friction and its minimum value $h_{\min}$ or the minimum thickness of the lubricating layer according to Reynolds

O_1 the center of the axis of rotation of the plain bearing shaft

O_2 the center of the axis of the radial plain bearing

e the eccentricity between the central axes of symmetry of the rotating shaft O_1 and the cylindrical friction surface of the bearing O_2, which occurs during friction

p_0 ambient pressure: atmospheric pressure or pressure of the lubricating medium in the lubrication system

$\sigma_{\max}$ maximum contact stresses, determined by the formulas of G. Hertz

b the width of the linear contact, determined by the formula of G. Hertz

l length of the linear contact of the radial plain bearing

r radius of plain bearing shaft

R radius of plain bearing

υ Linear sliding speed

ν Kinematic viscosity of the lubricating medium

k_1, k_2 Rheological coefficients

x the coordinate of the contact points of the abscissa corresponding to the direction of sliding

CED convergent elastically deformed area of tribocontact

DED divergent elastically deformed area of tribocontact

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References

1. Kragelskiy I V, Dobychin M N, Kombalov V S. Basics of friction-wear calculations. Moscow: Mashinostroenie, 1977. (in Russian)
2. Demkin N B, Ryzhov E V. Surface quality and machine parts contact. Moscow: Mashinostroenie, 1981. (in Russian)
3. Bowden F P, Tabor D. Friction and Lubrication of Solids. Oxford (UK): Oxford University Press, 2001.
4. Kostetskiy B I. Wear resistance of machine parts. Moscow : MASHGIZ, 1950. (in Russian)
5. Khebdy M, Chichinadze A V. Handbook of Tribotechnics. Moscow: Mashinostroenie, 1989. (in Russian)
6. Kragelskiy I V, Alisin V V. Friction, Wear and Lubrication: A Handbook. Moscow: Mashinostroenie,

1978. (in Russian)
7. Garkunov D N. Tribotechnics. Moscow: Mashinostroenie, 1989. (in Russian)
8. Mashkov Y K. Tribology of Structural Materials: Tutorial. Omsk: OmSTU Press, 1996. (in Russian)
9. Luzhnov Y M, Aleksandrov V D. Basics of Tribotechnics: Tutorial. Moscow: MADI, 2013. (in Russian)
10. Akhmatov A S. Molecular Physics of Boundary Friction. Moscow: Fizmatgiz, 1963. (in Russian)
11. Leybenzon L V. Hydrodynamic theory of lubrication. Moscow: State Technical and Theoretical Publishing House, 1934. (in Russian)
12. Petrov N P. Friction in machines and the effect of lubricating fluid on it. Moscow: Trudy, 1883. (in Russian)
13. Reynolds O. On the theory of lubrication and its application to Beauchamp Tower's experiments including an experimental determination of the viscosity of olive oil Philos. London: London Ser, 1886.
14. Stribeck R. Die wesentlichen Eigenschaften der Gleitund Rollenlager. Berlin: Springer, 1902: 341. (in German)
15. Ertel A M. Hydrodynamic calculation of lubrication contact of curved surfaces. Moscow: CNIITMASH, 1945. (in Russian)
16. Grubin A N. Contact stresses in toothed and worm engagements // Fundamentals of the hydrodynamic theory of lubrication of heavily loaded cylindrical surfaces / Trudy CNIITMASH, book 30. Moscow: Mir, 1949. (in Russian)
17. Petrusevich A I. The main conclusions from the contact-hydrodynamic theory of lubrication. Izvestiya AN SSSR: OTN 2: 209-223 (1951). (in Russian)
18. Petrusevich A I. The main conclusions from the contact-hydrodynamic theory of lubrication. Izvestiya AN SSSR: OTN 2: 209-216 (1951). (in Russian)
19. Murch L E, Wilson R D. A thermal elastohydrodynamic inlet zone analysis. ASME J. Lubr Technol 97(2): 212-216 (1975).
20. Gregg S J, Sing K S W. Adsorption, Surface Area and Porosity: 2. Auflage. London: Academic Press, 1982. (in German)
21. Fenelonov V B. Introduction to the physical chemistry of the formation of supramolecular structure of adsorbents and catalysts. Novosibirsk: SO RAN, 2002. (in Russian)
22. Prodan V D. Tightness of detachable connections of equipment operated under the pressure of the working medium:tutorial. Tambov: FGBOU VPO 'TGTU' Press, 2012. (in Russian)
23. Qiu Y. On the prediction of cavitation in dimples using a mass- conservative algorithm. J Tribol 131(4): 1–11 (2009).
24. Jakobsson B, Floberg L. The finite journal bearing, considering vaporization. Sweden: Tran Chalmers University of Tech Gothenburg, 1957.
25. Olsson K O. Cavitation in dynamically loaded bearings. Sweden: Tran Chalmers University of Tech Gothenburg, 1965.
26. Elrod H G. A Cavitation Algorithm. Journal of Tribology 103(3): 350–354 (1981).
27. Elrod H G. A computer program for cavitation and starvation problems. Cavitation and related phenomena in lubrication: 37-41(1974).
28. Fesanghary M, Khonsari M M. A modification of the switch function in the Elrod cavitation algorithm. J Tribol 133(2) (2011).
29. Brewe D E. Theoretical modeling of the vapor cavitation in dynamically loaded journal bearings. J Tribol 108: 628–637 (1986).
30. Woods C M, Brewe D E. The solution of the Elrod algorithm for a dynamically loaded journal bearing using multigrid techniques. Tribology conference in Maryland : 302-308 (1989).
31. Vijayaraghavan D, Keith Jr T G. Development and evaluation of a cavitation algorithm. Tribol Trans 32(2): 225–233 (1989).
32. Vijayaraghavan D, Keith Jr T G. An efficient, robust, and time accurate numerical scheme applied to a cavitation algorithm. J Tribol 112: 44–51 (1990).
33. Qiu Y. Performance analysis of full-film textured surfaces with consideration of roughness effects. J Tribol 133(2): 021704 (2011).
34. Booker J F. A finite element cavitation algorithm. J Tribol 113(2-4): 276–284 (1991).
35. Kumar A, Booker J F. A finite element cavitation algorithm: application/validation. J Tribol 113(2): 255–260 (1991).
36. XIE Y. A mass - conservative average flow model based on finite element method for complex textured surfaces. Sci China Phys Mech Astron 56: 1909–1919 (2013).
37. Shi F. An implicit finite element cavitation algorithm. Comput Model EngSci 3(4): 507–515 (2002).
38. Hajjam M. A transient finite element cavitation algorithm with application to radial lip seals. Tribol Int 40(8): 1258–1269 (2007).
39. Gherca A. Effects of surface texturing in steady-state and transient flow conditions: Two-dimensional numerical simulation using a mass-conserving cavitation model. J Tribol 229(4): 505–522 (2014).
40. Payvar P. A computational method for cavitation in a wavy mechanical seal. J Tribol 114: 199–204 (1992).

41. Wang Y. Mixed lubrication of coupled journal-thrust-bearing systems including mass conserving cavitation. J Tribol 125(4): 747–755 (2003).
42. Harp S R. An average flow model of rough surface lubrication with interasperity cavitation. J Tribol 123(1): 134–143 (2000).
43. Shi F. A mixed soft elasto-hydrodynamic lubrication model with interasperity cavitation and surface shear deformation. J Tribol 122(1): 308–316 (1999).
44. Zhang J. Direct observation of cavitation phenomenon and hydro - dynamic lubrication analysis of textured surfaces. TribolLett 46(2): 147–158 (2012).
45. Brunetiere N. Numerical analysis of a surface-textured mechanical seal operating in mixed lubrication regime. TribolInt 49: 80–89 (2012).
46. Xiong S. Steady-state hydrodynamic lubrication modeled with the PayvarSalant mass conservation model. J Tribol 134(3): 1–16 (2012).
47. Koronatov V A. On the correct application of Coulomb's law when using experimental friction characteristics. Technology 3 (43): 35-43 (2019). (in Russian)
48. Stelmakh O U. Physical and technological bases of control of dynamic processes in lubricating layers to improve their performance of tribosystems. Khmelnitskiy (Ukraine): Khmelnitskiy National University, 2015. (in Russian)
49. Pinkus O. Thermal Aspects of Fluid Film Tribology. New York: ASME Press, 1990: 317-326.
50. Hardy W I. Collected Scientific Papers. London: Càmbridge, 1936.
51. Altoiz B A, Bondarev V N, Shatagina E A, Kiriyan S V. Model of organization of an epitropic liquid-crystal phase. Zh. Tekh. Fiz 84: 58-61 (2014). (in Russian)
52. Garkunov D N. Selective transfer in friction nodes. Moscow: Transport, 1969. (in Russian)
53. Garkunov D N, Kragelskiy I V. Wearlessness effect. USSR Patent 41: 12 Nov. 1956.
54. Dukhovskoy E A, Onischenko V S, Ponomarev A N, Silin A A, Talroze V L. The phenomenon of abnormally low friction in a vacuum. USSR Patent 121: 16 Sep. 1969.
55. Cameron A. Basic Lubrication Theory. Moscow: MASHGIZ, 1962. (in Russian)
56. Kravchuk A S, Kravchuk A I. Applied contact problems for a generalized rod model of coating. St. Petersburg: Naukoyemkie tekhnologii, 2019. (in Russian)
57. Johnson K. Mechanics of contact interaction. Moscow: Mir, 1989. (in Russian)
58. Kravchuk A S, Kravchuk A I. Mechanics of contact interaction of bodies with circular boundaries. Minsk: Tekhnoprint, 2000. (in Russian)
59. Zolotarevskiy V S. Mechanical properties of metals. Moscow: Metallurgiya, 1983. (in Russian)
60. Braun M J, Hannon W M. Cavitation formation and modelling for fluid film bearings: A review. J Eng Tribol 224(9): 839-863 (2010).
61. Information on https://oaktrust.library.tamu.edu/bitstream/handle/1969.1/93246/Notes06%20Liquid%20cavitation%20model.pdf?sequence=1&isAllowed=y.
62. Klamann D. Lubricant and related products. Moscow: Khimiya, 1988. (in Russian)
63. Donkin S B. Report of the Lubricants and Lubrication Inquiry Committee. London: Department of scientific and industrial research, 1920.
64. Cole J A, Hughes C J. Oil flow and film extent in complete journal bearings. Proceedings of the Institution of Mechanical Engineers 170(1): 499-510 (1956).

Олександр Стельмах, Хун'ю Фу, Іцяо Гуо, Сіньбо Ван, Хао Чжан, Павло Каплун. Екструзія та розрідження мастила в граничному шарі як ключьові процеси адгезійного зношування високонавантажених трибоконтактів.

Представлено комплексний аналіз адгезійно-деформаційної, пружно-гідродинамічної та гідродинамічної моделей тертя, які описують різні режими змащування відповідно до кривої Штрібека. Основні положення цих моделей розглядаються в поєднанні з теорією адсорбції Ленгмюра і пружно-деформаційною теорією криволінійних контактів Герца. Показано, що виявлені протиріччя потребують свого вирішення, а виявлені множинні ефекти потребують науково обґрунтованої інтерпретації. Пропонується розробити більш узагальнену модель тертя та зношування на основі природних процесів, які довгий час були приховані від прямих спостережень. Це: Екструзія змащувальних шарів у збіжних пружно деформованих і Розрідження у розбіжних пружно деформованих областях трибоконтактів. Розуміння цих процесів дає змогу передбачити місця локалізації та причини виникнення первинних наступних актів зчеплення поверхонь тертя та їх зношування за таким циклом: «розрідження та десорбція мастильних шарів, що призводить до деформаційного руйнування оксидних плівок та зрощення ювенільних ділянок поверхні, після чого до відриву осколкового матеріалу від підшипника та новоутворення виступу на валу – у розбіжних пружно деформованих ділянках контакту. Далі відбувається мікрорізання цим фрагментом опорної поверхні з виходом продукту зношування в конвергентну пружнодеформовану область, що відповідно призводить до зміни фактичної геометрії та натягу трибоконтакту. Крім того, в інших областях відновленого контакту адгезивна взаємодія відбувається в інших розбіжних областях відповідно до того самого механізму. Глибоке розуміння причин десорбції мастильних шарів дозволить розробити та застосувати нові високоефективні технологічні та матеріалознавчі методи з метою підвищення ресурсу високонавантажених трибосистем машин і механізмів.

Ключові слова: моделі макротертя; адгезійний знос; градієнт тиску; кавітація