Vol. 45, No. 4 (2023) 729-741, DOI: 10.24874/ti.1581.11.23.12

Tribology in Industry

www.tribology.rs

RESEARCH

Calculation of Wear of Metallic Surfaces Using Material's Fatigue Model and 3D Texture Parameters

Guntis Springis[a,*], Irina Boiko[a], Oskars Linins[a]

[a]*Institute of Mechanics and Mechanical Engineering, Faculty of Mechanical Engineering, Transport and Aeronautics, Riga Technical University, Kipsalas 6B, Riga, Latvia.*

Keywords:

Friction
Wear
Wear calculation
3D surface roughness
Material's fatigue, kinematic
Constructive parameters

* Corresponding author:

Guntis Springis
E-mail: guntis.springis@rtu.lv

Received: 15 November 2023
Revised: 29 November 2023
Accepted: 14 December 2023

A B S T R A C T

Today, when it comes to manufacturing parts and components for various mechanisms, the tendency is to use both approaches – the latest technologies and new material combinations- to achieve a longer product's lifetime. That is why the issue of machine parts' working capacity criteria, one of the most important of which is wear and its prediction, remains vitally important. Having studied the prediction theories of the wear process that have been developed over time, one can state that each has shortcomings that might strongly impair the results, thus making unnecessary theoretical calculations. Also, predicting wear based on lengthy, time-consuming, costly experiments is still prevalent. The article discusses a new wear calculation model, which is based on the application of theories from several branches of science. This model considers the surface texture (3D) parameters' values in modelling the surface's micro-topography with the random field theory while the friction surfaces' destruction – with the fatigue theory. The new wear calculation model is synthesised based on a developed friction surface contact model, providing a more complete surface description, which is essential for wear calculation and gives more accurate results. The proposed new wear calculation formula includes parameters that can be easily determined using modern measurement methods, thus speeding up the product design process and significantly contributing to sustainable development. Experimental studies on the steel-bronze sliding friction pair validated the analytical wear calculations' results.

1. INTRODUCTION

With the increasing development of machine parts' manufacturing technologies and the range of materials used, the lifetime of different products, their prediction, and the factors that influence it have always been a particularly topical issue. Over time, a number of wear calculation models have been developed which can be used to predict approximately the service life of a particular product. Due to the variety of wear processes, many parameters influence the wear process: geometry of the surface asperities (roughness, waviness, shape deviation, etc.), physical-mechanical conditions of the surface, material of the parts, wear temperature, wear regime, etc. It is not possible to consider all these factors acting in the process analytically, and therefore wear calculations developed based on several theories which take into account the complex of influencing variables.

One of the most popular theories for calculating wear was developed by the British scientist J.F. Archard. This theory is based on the idea that the volume of material worn Q (mm^3) is proportional to the sliding distance L (mm), and the normal load N (N), and inversely proportional to the hardness of the material concerned H [1]:

$$Q = k \cdot \frac{N \cdot L}{H}. \tag{1}$$

where k - wear coefficient.

A literature review [1-11] showed that Archard's expression for the calculation of wear, in some cases modified or updated to suit an object's geometry and materials, is still widely used today. Although simple enough, it has its drawbacks: the coefficient k has to be initially determined experimentally, there is insufficient information on the contact surfaces, only the hardness of the material is taken into account as a material characteristic, and there is no information on the surface roughness.

There are also mixed wear calculation models that integrate several approaches, which should be noted as wear calculation models based on mass and energy balance [12-14]. Mass balance models were used for describing the formation of iron sulphide layers in a lubricated tribosystem by using radioactive sulphur and sulphur compounds, for describing the behaviour of metal transfer and oxidation in the wear process, for modelling the wear behaviour of granular matter in the frame of the third body concept [12]. Concerning wear prediction based on energy balance, it is stated that mechanisms and processes of energy balance involve transfer, storage, emission, and dissipation of mechanical work [12].

One of the most prominent scientists who linked the wear rate to the specific pressures and the relative sliding velocity of a friction pair in a calculation method was A.Pronikov. The scientist's offered methodology allows determining the wear (change in linear dimensions of the body) and the shape of the worn surface. These calculations are based on materials' wear behaviour and consider the joined surfaces' configuration [15]. Still, the wear resistance parameters included in the equation are only determined by long-term experimentation, so there is no point in carrying out wear calculations in advance.

Researchers in other fields use calculation methods that already include the friction pair's structural characteristics, the friction material's physics-mechanical parameters, and the surface's geometrical parameters. The most famous scientist belonging to this group is I.Kragelsky. The wear calculation model developed by I.Kragelsky includes the friction pair's structural characteristic quantities and physical and mechanical parameters of the material of friction components as well as geometrical parameters of the components' surfaces. This model takes into consideration not only the impact of the material hardness and load on the friction pair but also characteristic quantities of a definite material's flexibility, mode of component operation (load, velocity, temperature), external conditions (lubrication, environment), and the constructive peculiarities of the friction pair [16]. The shortcoming of this calculation model is that at the characterisation of the friction component's surface parameters, the non-standard roughness parameters are used, which implies additional calculations.

At the moment, the model that can be considered as an almost complete model for the wear calculation of sliding-friction surfaces was introduced by scientist J.Rudzitis. In comparison with the previous wear calculation model (offered by I.Kragelsky), in addition to the wear calculation parameters mentioned above, it takes into consideration the standardised profile roughness values at modelling the surface's micro-topography with the random field theory while the friction surfaces' destruction – with the fatigue theory [17]. The most crucial drawback of this model is the use of surface roughness profile parameters, which still do not provide complete information about the real micro-topography of the friction surface, which may also reduce the accuracy of the wear calculations.

One of the most important scientific achievements in product inspection is related to the introduction of the standard for surface texture (3D) parameters in 2012 (ISO 25178). It should be noted that this has had a significant impact on the approach to further manufacturing and scientific research, allowing the accuracy and quality of the results to be improved, as well as providing wide opportunities at a fundamental science level to integrate the new parameters into the processing and analysis of research. Studies by several researchers [18-21] show that surface texture (3D) parameters provide more detailed information about the real surface topography than profile parameters, allowing to process research results with greater accuracy, which is also an essential prerequisite for this research. Based on the above, the 3D texture parameters necessary to define a rough surface are integrated into the surface contact model discussed below and applied to the wear analytical calculations to obtain more accurate and realistic calculation results.

2. MATHEMATICAL MODEL OF SURFACE AND WEAR CALCULATION

2.1 Surface roughness modelling

For studying the irregular surface roughness the random function theory is efficient, thus the surface micro-topography can be described by a 2D random function, i.e. random field $h(x, y)$ with two variables (x and y) [22].

The random field at worn surfaces is assumed to be normal, i.e. the ordinates of such a field are distributed according to the normal distribution law, characterised by the height parameter Sq (root mean square deviation from the mid-plane).

An important characteristic of the random function is the correlation function, which indicates the relationship between the points of the random function, so the faster the correlation function decreases, the more chaotic is the random field. The correlation function depends on two variables τ_1 and τ_2, where τ_1 and τ_2 are the projections on the abscissa and ordinate axis of the vector τ connecting two points on the surface in the Cartesian coordinate system. Thus, the following definition can be applied: the surface roughness is described by a normal uniform two-variable random field $h(x, y)$ possessing ergodic properties and whose correlation function is uninterrupted and has uninterrupted derivatives. The mean value of random field is constituted by a plane, which can be called a mid-plane. Thus, to describe a normal random field, the mathematical expectation of this field and correlation function should be known [22].

We can assume that the area has been set when its dispersion and standardised correlation function are known. The requirement to find the area dispersion leads to finding Sa (the standard arithmetic deviation from the mid-plane) for the surface, and the requirement to solve the problem $\rho(\tau_1, \tau_2)$ – to the determination of the corresponding roughness step parameter RSm_1 (a step perpendicular to the processing trace direction; is taken from the surface texture) and RSm_2 (a step towards the processing trace along; is taken from the surface texture) (Fig.1). The surface texture parameters Sa and Sq are linked by the following expression [22]:

$$Sa = \sqrt{\frac{2}{\pi}} \cdot Sq. \tag{2}$$

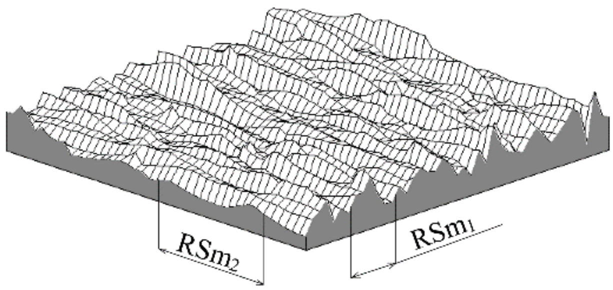


Fig. 1. Step parameters for irregular surface roughness.

The step parameters RSm_1 and RSm_2 measured from surface texture allow the determination of the anisotropy coefficient Str:

$$Str = \frac{RSm_1}{RSm_2}. \tag{3}$$

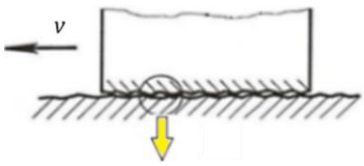
The anisotropy coefficient Str varies from 0 to 1. At Str=1 the area is isotropic, while at Str=0 it is maximum stretched.

Thus, a surface texture can be described in height using the parameter Sa, and in steps - longitudinally RSm_2 and transversally RSm_1.

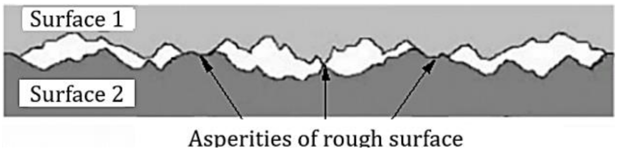
It should be noted that the selected surface texture (3D) parameters are technologically feasible during the surface preparation process and can be easily determined with modern measuring equipment.

2.2 Formation of the wear

In previous research works [24, 25], the fatigue character of the wear process has been proved. This means that the wear of contacting materials results in the formation and spreading of cracks, which finally leads to the separation of material particles. When two rough surfaces are brought together with a definite force F at a definite velocity v (Fig.2.), the contacting asperities of a rough surface generate stresses which create the conditions for the material to be destroyed.



(a)



(b)

Fig. 2. Schematic contact of two rough surfaces.

The process of calculating linear wear can be described by the following equation [22]:

$$U_n = V_\Sigma \cdot \frac{N_{cf}}{N_c}. \quad (4)$$

where

V_Σ - deformed volume over the entire friction surface;

N_{cf} - the actual number of cycles applied to the surface asperities during the friction process;

N_c - number of cycles leading to the destruction of the surface asperities.

The calculation of the above parameters will be the subject of discussion below.

2.3 Wear particle's formation of rough surfaces

Based on [22], N_{cf} is calculated as follows:

$$N_{cf} = \frac{L_p}{RSm_2^a}. \quad (5)$$

where

L_p is the length of the friction path;

RSm_2^a - the mean step of surface roughness towards friction for an active surface (i.e. for the surface that causes wear of another surface).

Using the linear summation of stress [24] and assuming that its amplitude distribution complies with that of the roughness peaks, the average number of cycles for material destruction can be determined by the following formula:

$$N_c \frac{N_0}{5m!} \cdot t_\sigma^m. \quad (6)$$

where

t_σ is the non-dimensional stress relation;

N_0 - the number of material durability cycles at a non-symmetrical load;

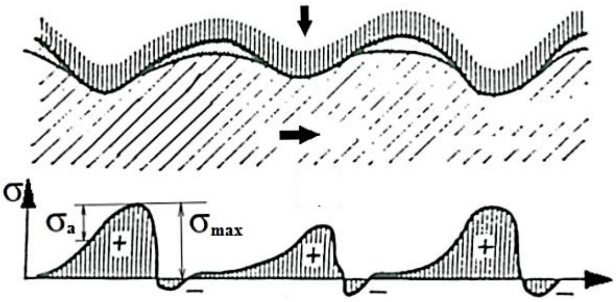
m - the degree of the fatigue curve equation.

In turn, t_σ is calculated using the following coherence [24]:

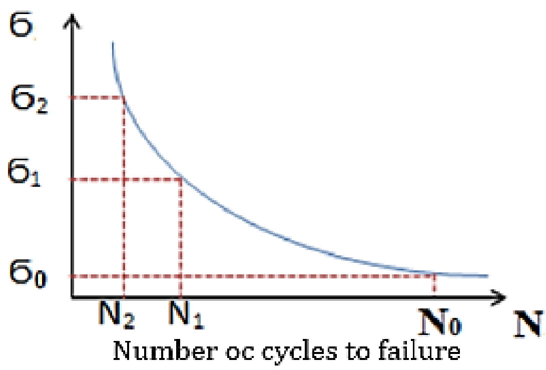
$$t_\sigma = \frac{\sigma_0}{\sigma_a}. \quad (7)$$

where σ_0 is the material durability limit.

In motion, any asperity whose height exceeds a certain level, determined by the position of the opposite asperity, deforms that asperity, generating a stress field. Fig. 3 shows that the stress of asperities of two sliding surfaces changes follow a non-symmetrical cycle.



(a)



(b)

Fig. 3. Irregular rough surfaces: loading diagram of the interaction of the asperities and stress variation diagram (a); cycle number curve (b).

Parameters (N_0, m, σ_0) in formulas have the mean values for definite material type.

From formula (7) it follows that the number of material destruction cycles is connected with the stress amplitude σ_a. Based on the equations given in [17,24], integrating surface roughness parameters and by mathematical operations, the stress amplitude σ_a absorbed by a single ellipsoidal asperity is calculated according to the following formula:

$$\sigma_a = \frac{\pi^2}{\sqrt{2}} \cdot \frac{E}{[K(e)]^{1/2}} \cdot \frac{Sa}{RSm_1}. \tag{8}$$

where

π is the mathematical constant;

E - modulus of elasticity of the material;

$K(e)^{1/2}$ - elliptic integral.

By inserting formula (8) into the basic equation (6), we obtain the final formula for calculating the number of cycles required for material destruction:

$$N_c = \frac{N_0}{5m!} \cdot [\frac{\sqrt{2}}{\pi^2} \cdot \frac{\sigma_0 \cdot RSm_1 \cdot K(e)^{1/2}}{E \cdot Sa}]^m. \tag{9}$$

The next step explains the volume of asperity removal from the rough surface due to friction and wear.

2.4 Calculation of the rough surface particle volume separated in the wear process

Since the irregular character of surface roughness in the given model is described with a normal random field $h(x, y)$, high peaks of this field can be shown by elliptic paraboloids with the segment volume V_p [22]:

$$V_p = \frac{\pi \cdot h_0^{\ 2}}{K_i^{1/2}}. \tag{10}$$

where

h_0 is the height of the paraboloid segment measured from the top (thickness of the separated particle);

K is Gauss' bending of the roughness peak.

Parameter h_0 (Fig. 4.) is the thickness of a particle separated in the wear process.

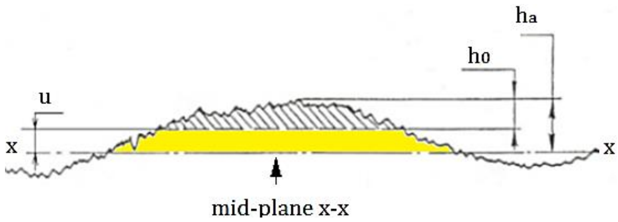


Fig. 4. Possible separation of wear particles from the surface's asperity (h_a -rough surface asperity's total height).

Thus, the h_0 value depends on the situation with the upper layer and other physical and mechanical factors that determine the particle formation during a friction process. Taking into consideration the requirements of the moving contact model, the wearing can proceed according to the following scheme: at the cyclic loading of peak tops (with account taken for imperfection of the material) a crack is being formed in the subsurface layers of the material. Under load the cracks merge, grow, and the particles separate in the form of h_0 thick scales. The h_0 value should be estimated based on the analysis of the upper layer's condition [24].

Considering the relationships found in the literature [17,25] and performing mathematical calculations, the average value of the volume separated by the *i*-th asperity can be determined as follows:

$$V_i = \frac{Sq}{2 \cdot \gamma^2 \cdot \pi \cdot n_1(0) \cdot n_2(0)}. \tag{11}$$

where

γ is the relative height of the cut ($\gamma = u/Sq$; u-the level to which deformation of the surface asperity occurs);

$n_1(0)$, $n_2(0)$ are the numbers of zeros in two mutually perpendicular directions of surface cuts x and y (i.e. in the longitudinal and transversal roughness directions of the surface).

To determine the total volume V_Σ, it is necessary to multiply the volume V_i of one asperity by the number N_γ of deformed asperities.

2.5 Determination of the number of deformed asperities of 3D surface texture

One of the most important parameters in the wear process is the number of asperities on the contacting surfaces. The surface asperity is the part of a rough surface above the level u (the normalized value of the level u is $\gamma=u/Sq$). As shown by other researchers (J. Rudzitis et.al.), in practice under real loads, the deformation of the rough surface's asperities occurs mostly at the level $\gamma \geq 2$. In this case according to [23] the number of asperities N_γ per unit area involved in friction and wear process can be calculated as follows:

$$N_\gamma = \frac{1}{5} \cdot n_1(0) \cdot n_2(0). \quad (12)$$

Expressing root mean square deviation from the mid-plane Sq from formula (2) and inserting it into expression (11), carrying out the mathematical calculations, the total volume of the asperities V_Σ per unit area separated by friction is calculated by the following expression:

$$V_\Sigma = \frac{Sa \cdot \sqrt{\pi}}{10 \cdot \sqrt{2} \cdot \gamma^2}. \quad (13)$$

2.6 Summary of the wear calculation equation

Based on formula (4) and inserting equations (5) and (9) into it, we obtain the formula for calculating the linear wear over the service life:

$$U_n = V_\Sigma \cdot \frac{5m!}{N_0} \cdot \left(\frac{E}{\sigma_0}\right)^m \cdot \left(\frac{Sa}{RSm_1}\right)^m \cdot \left(\frac{\pi^2}{\sqrt{2} \cdot K(e)}\right)^m \cdot \frac{L_p}{RSm_2^a}. \quad (14)$$

Formula (14) contains, in addition to the physical-mechanical parameters and surface texture (3D) parameters also the parameter γ, which is determined by two surface contact as the relative surface deformation rate. This level is determined for the surface subjected to wear using the contact theory equations [25]:

$$q = \frac{k_q \cdot Sa}{RSm_1 \cdot \theta} \cdot F_1(\gamma). \quad (15)$$

where

q is the pressure on contacting surfaces in elastic contact;

k_q - the coefficient depending on the roughness anisotropy coefficient Str;

$F_1(\gamma)$ - the relative surface deformation level's function;

θ - the constant of the material's elasticity.

The values to be specified in this formula are $F_1(\gamma)$. Experimental studies have shown that for friction surfaces, after the period of running-in stage, γ is in the range of ≈1.4 to ≈2.7, so we can assume γ=2 as an average value.

According to [17,26] for friction surfaces at $\gamma = 2$:

$$F_1(\gamma) = \frac{1}{40 \cdot \gamma}. \quad (16)$$

Thus, by inserting the resulting relationships into formula (14), linking linear wear to the motion parameters of the friction surfaces (relative velocity v and time t) and making mathematical calculations, we obtain the average linear wear formula for the normal service life (normal wear stage):

$$U_n(t) = 32 \cdot \frac{m!}{N_0 \cdot k_q^{\ 2}} \cdot \frac{E^{(m-2)}}{\sigma_0^{\ m}} \cdot \frac{Sa^{(m-1)}}{RSm_1^{\ (m-2)}} \cdot \left(\frac{\pi^2}{\sqrt{2} \cdot K(e)^{1/2}}\right)^m \cdot q^2 \cdot \frac{v \cdot t}{RSm_2^a}. \quad (17)$$

The parameters Sa and RSm_1 are for the wearing surface, but RSm_2^a - for the surface that causes wear of another surface).

The next step is to verify the correlation of analytically calculated wear values with experimental data.

2.7. Application of the wear calculation model

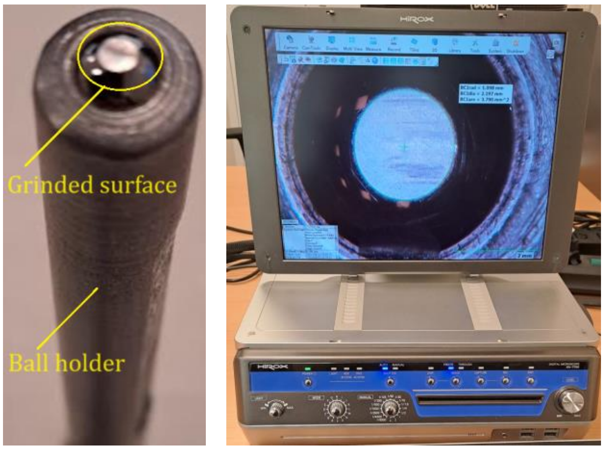
Summarising the information and formulae presented in the previous chapters, one can see that the wear calculation equation (17) proposed by the authors includes the following parameters: structural-kinematic (v, t, q), material fatigue and physico-mechanical parameters (N_0, m, σ_0, E) of the friction pair, as well as standardised surface texture (3D) parameters Sa, RSm_1 and RSm_2^a (according to EN ISO 25178). This set of parameters was not taken into account in any of the wear calculation models discussed above - Archard analysed only load, distance and material hardness; Pronikov considered speed and load without describing surface roughness; Kragelsky described the physical phenomena of the friction process but used non-standardised surface roughness parameters, while Rudzitis applied roughness parameters to the profile rather than the surface when calculating wear, thus incompletely describing the actual surface. It can be concluded that the wear calculation model proposed by the authors is simple enough and can be used for wear calculations in engineering tasks, as all the parameters included are easy enough to determine using technical literature and modern measuring equipment.

3. MATERIALS AND METHODS

3.1 Selection of sample materials and sample preparation

The following specimens and their materials were selected for the experimental studies:

1. A steel ball with a diameter of 6 mm. The material of the ball - 102Cr6 (EN 1.2067). The ball was machined prior to the experiment to produce a plane with a defined area by grinding and the diameter of the grinded surface was measured (Fig. 5).



(a) (b)

Fig. 5. A ball with a grinded surface: (a) ball in holder; (b) measurement of the diameter and surface area of the ball.

This option was chosen on the basis that the wear model requires a constant nominal area of the friction pair's contact during the wear process, which is not possible for a round ball without surface grinding. The surface area of the ball ground - 2,1 mm.

2. The disc with a diameter of 40 mm and a thickness of 5.5 mm. The disc material is bronze CW307G (EN 12163).

Before the experiment, the samples were treated with sandpaper with different abrasive grain gradations, thus achieving the required roughness of the contact surfaces of the samples (*Sa*<0,1). After the surface grinding operations were completed, both samples were cleaned with a wipe dipped in ethanol, removing metal chips and abrasive sandpaper from the surface of the samples. After grinding, control measurements of the surface texture (3D) parameters were carried out on both samples according to EN ISO 25178.

3.2 Equipment and software used for measurement and data processing

The scheme of the tested sample's contact - "ball (grounded contact plane) - rotating disc" (Fig.6).

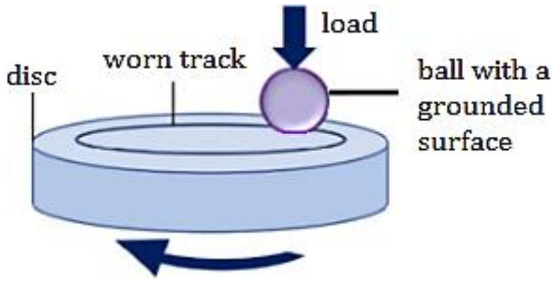
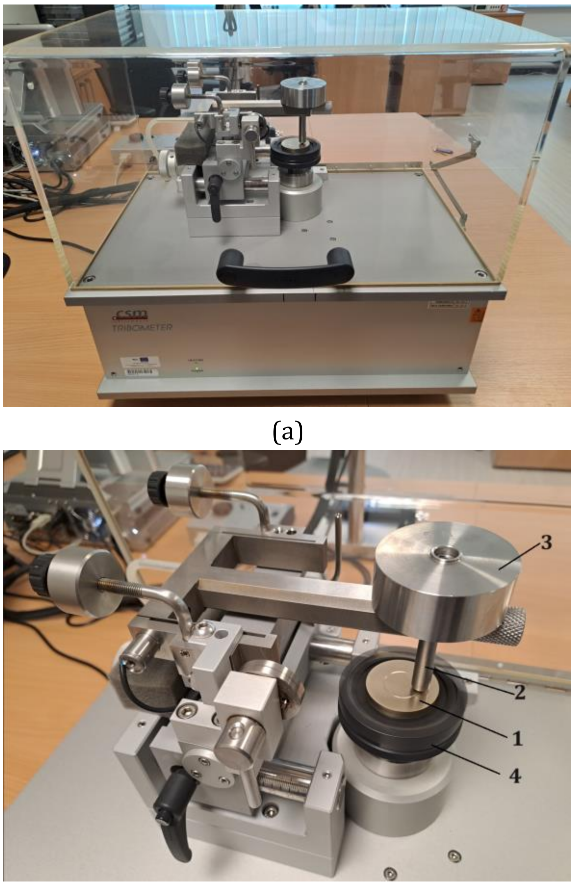


Fig. 6. Wear test scheme.

For the experimental investigations, the CSM tribometer (Fig. 7) (CSM Tribometer, Switzerland, maximum loading force - 10 N) was used.



(a)

(b)

Fig. 7. CSM tribometer: 1 - rotating disc; 2 - ball holder; 3 - loading weights; 4 - disc self-centring fixing chuck.

In order to set the load force, the linear velocity of the friction pair, etc., as well as to collect and process the friction coefficient's data and wear time measurements during the experiment, special software (InstrumX) was used (Fig. 8).

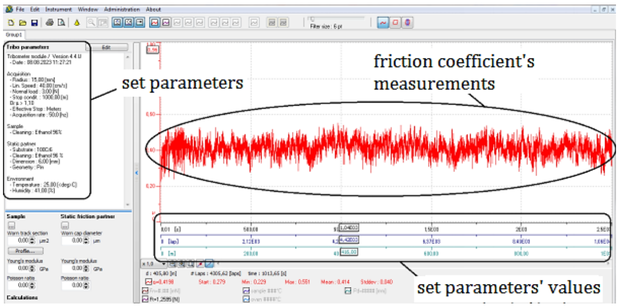


Fig. 8. Interface of the InstrumX software.

The parameters set for each experiment are shown in Table 1.

The surface texture (3D) parameters were measured using the AVANT 3D contour and surface roughness measuring system (Mitutoyo, Japan) (Fig. 9). The S-3000 roughness detector module featured a 0.75 mN detector and a standard stylus 12AAC731 with a cone angle of 60 degrees and a tip radius of 2 micrometres.

Fig. 9. AVANT 3D contour and surface roughness measuring system.

Table 1. Values used in wear calculations and experiments.

Parameter	Designation		Value	Units
Load	q		1) 0.58; 2) 0.87; 3) 1.45.	MPa
Linear velocity	v		1) 700; 2) 450; 3) 300.	mm/s
Wear (sliding) distance			1) 6000; 2) 4000; 3) 4000.	m
Surface contact	Dry friction (in all cases)			
Hardness of the disc	HRC 17			
Hardness of the ball	HRC 62			
Fatigue failure parameters of the disc material:	Degree of the fatigue curve equation	m	4	-
	The material's durability limit	σ_0	300	MPa
	The number of material's durability cycles	N_0	$5\text{x}10^6$	-
Modulus of elasticity of the disc's material	E		$1.15\text{x}10^5$	MPa
Surface texture (3D) parameters (after running-in stage)	The standard arithmetic deviation from the mid-plane	Sa	1) 0.79; 2) 0.83; 3) 1.8.	µm
	The step perpendicular to the wear trac	RSm_1	1) 12; 2) 17; 3) 49.	µm
	The step parallel to the wear trac	RSm_2^a	1) 61; 2) 65; 3) 120.	µm
	Surface anisotropy parameter	Str	<0.05	-
	Coefficient (depends on surface anisotropy)	k_q	0.15	
Wear after running-in stage	U_p		1) 2,61; 2) 2,88; 3) 5,44.	µm

The profilometer Mitutoyo SURFTEST SJ-500 (Mitutoyo, Japan) was used to determine the width and cross-sectional area of the worn track (Fig. 10).

Fig. 10. Mitutoyo Surftest SJ-500 profilometer.

The diameter of the grounded ball's plane area was measured before/after the experiment using a Hirox digital microscope (Hirox, Japan).

Further processing of the data was carried out using MCube Map Ultimate 8.0 software, TalyMap Gold software and Microsoft Excel.

4. RESULTS AND DISCUSSION

Each experiment was repeated three times at the given values (speed *v* and load *q*), and the average values were calculated from separate measurements for statistically reliable data from the three experiments in each group. The results are plotted in Table 1 and graphs 15, 16 and 17.

In order to avoid overlapping curves in the graphs of the wear results, average values were calculated for each set of parameters.

During the experiments, the wear was measured on the bronze disc. It is assumed that the ball wears minimally, and that is why it was not considered in this case, although surface scratches oriented toward friction were observed on the contact surface of the ball already after the running-in stage. Based on the equation (17), it is necessary to determine values of *Sa* and RSm_1 for the wearing part of bronze disc and the RSm_2^q - for the ball's surface after running-in stage. For illustration one measurement (at *v*=450 mm/s, *q*=0,87 MPa) is shown in Fig. 11 and Fig. 12. The test specimens were fixed to the tribometer mounting points so that the displacement of the specimens after the measurements was minimal, thus ensuring minimal variations in the coefficient of friction before and after the measurements.

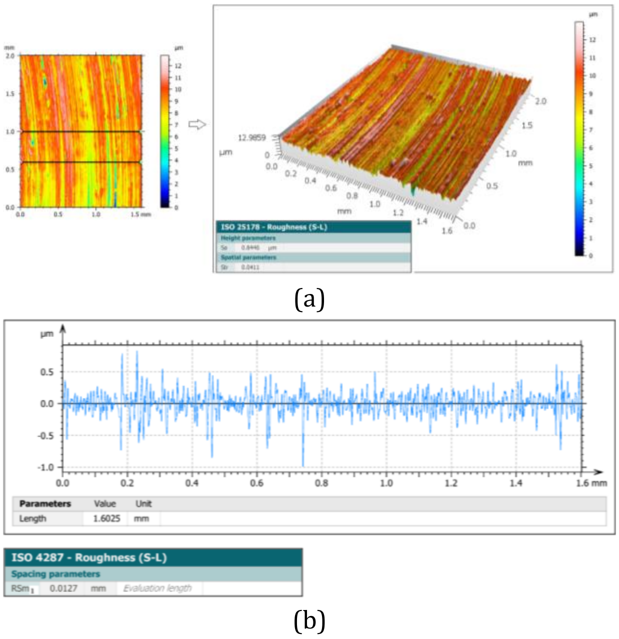


Fig. 11. Determination of *Sa* (a) and RSm_1 (b) for the bronze disc after running-in stage.

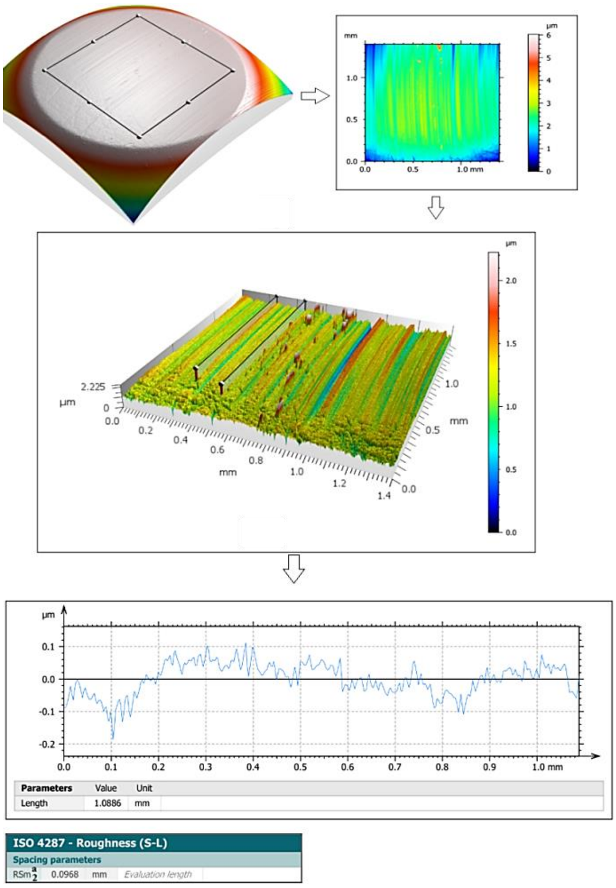


Fig. 12. Determination of RSm_2^q for the ball after running-in stage.

Due to the fact that a number of parameters, particularly the surface roughness parameters, change their values rapidly during the running-in stage, the surface texture (3D) measurements were taken on both samples after the running-in stage had finished and the wear values (experimental and analytically calculated) shown in Fig. 15, 16 and 17 are for normal (stable) wear stage. The end of the running-in stage was monitored by the stabilisation of the friction coefficient during the wear process (Fig. 13).

Fig. 13. Determination of the running-in stage for the wear process.

During the experiment, the wear values for bronze disc samples were measured every 500 meters. At each wear measurement stage, the worn track of the disc was measured at 4 locations (every 90 degrees), thus calculating the average linear wear for each stage. A sample of one measurement of the worn track is shown in Figure 14.

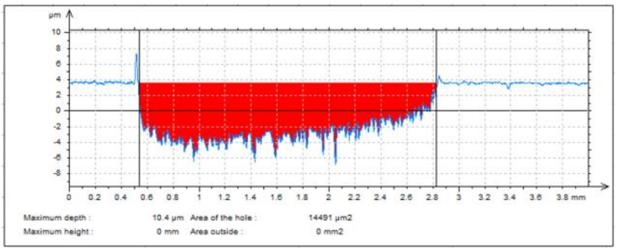


Fig. 14. Measurement of the cross-sectional area of a worn track on the bronze disc after 2500 metres (at *v*=450 mm/s, *q*=0,87 MPa).

The graphs of analytical and experimental linear wear are shown in Figure 15, 16, 17.

Figure 15 shows the wear values at linear velocity *v*=0.7 m/s and load *q*=0.58 MPa. The average experimental wear for a given group of specimens after 6000 metres of friction is 12.95 microns, and the analytical wear is 12.28 microns. One can see that the analytically calculated wear values for the bronze disc material are slightly lower than the average wear measured in the experiments (the average wear was calculated from the results of three experiments). The largest difference between the experimental and theoretical wear values is 17% at 500 metres of friction path, while the smallest difference is 3.7% at 3000 metres and 4.6% at 4000 metres.

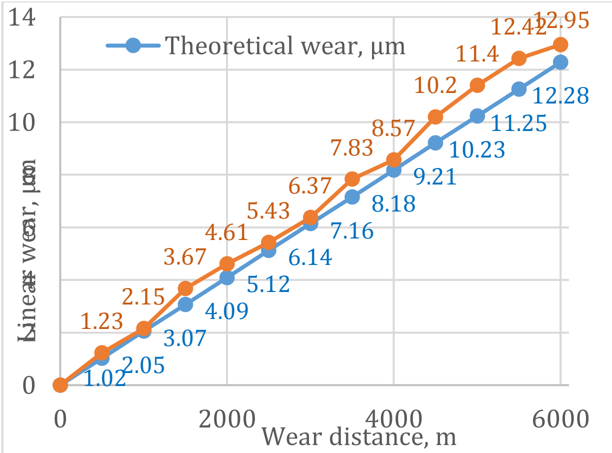


Fig. 15. Wear values of analytical calculations and experiments at *v*=0,7 m/s, *q*=0,58 MPa.

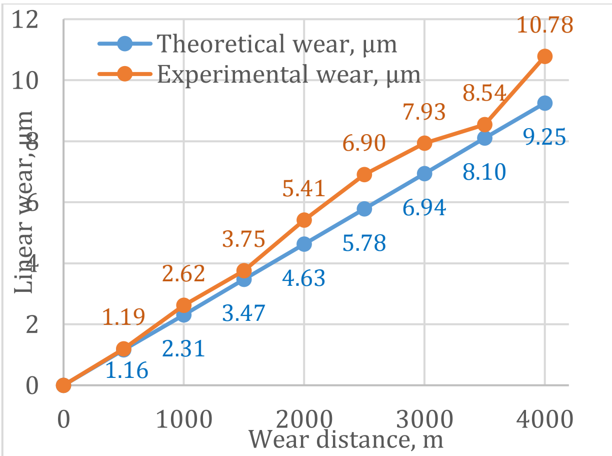


Fig. 16. Wear values of analytical calculations and experiments at *v*=0,45 m/s, *q*=0,87 MPa.

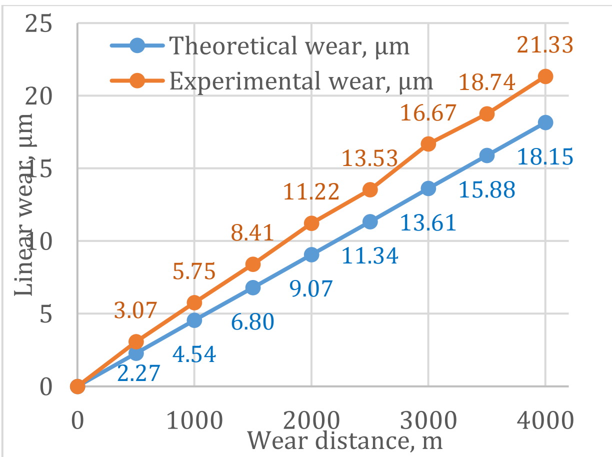


Fig. 17. Wear values of analytical calculations and experiments at *v*=0,3 m/s, *q*=1,45 MPa.

Figure 16 shows the wear values at linear velocity v=0.45 m/s and load q=0.87 MPa. The average experimental wear at the given parameters for the given group of specimens after 4000 metres of friction is 10.78 microns while the analytical wear is 9.25 microns. This graph shows that the analytically calculated wear of the bronze disc material is also lower than the average experimental wear. The largest difference is 16% at 2500 metres of friction path, while the smallest difference is 5.2% at 3500 metres.

A fairly similar situation is observed in Figure 17 at linear velocity v=0.3 m/s and load q=1.45 MPa. In this case the average experimental wear after 4000 metres of friction is 21.33 microns while the analytical wear is 18.15 microns. It should be noted that the difference in results is greater here than in the previous two cases. The largest difference is 26% at 500 metres of friction, while the smallest difference is 14.9% at 4000 metres and 15% at 3500 metres. This could be explained by the fact that, in this speed and load regime, a partial transfer of the bronze disc material to the steel ball contact surface was observed after the running-in period, creating conditions for both changes in the surface texture (3D) parameters and changes in the material properties of the friction surfaces, which could lead to an unstable wear process and permanent changes in the parameters included in the wear calculation formula.

Approximating the results of wear values from analytical calculations and experiments with corresponding equations, the coefficient of determination has been found to be at least 0.9.

By analysing the obtained average wear values for a given sliding friction pair at given kinematic, applied load, surface texture (3D) and fatigue parameters, it can be concluded that the proposed wear calculation model is valid for wear calculations in practical engineering tasks. Considering the wear values obtained in experimental studies and comparing them with those obtained in analytical calculations, as well as taking into account the set of parameters involved in the wear process, it can be seen that the proposed wear calculation model allows to obtain reliable wear values in a sufficiently simple way, thus saving time and technical resources required for long-term experiments. At the same time, the authors emphasise that further research is needed to analyse the wear model in more detail in order to determine the influence of the parameters included in the equation on the wear process and to verify the model by experiments with other materials/other loads and speeds/other surface texture (3D) parameters in order to confirm or reject the applicability of the model in certain regimes.

5. CONCLUSIONS

1. A new surface contact model between friction parts based on normal random field theory is offered. The model uses the surface texture (3D) parameters: a height parameter (Sa) and two surface roughness step parameters (RSm_1 and RSm_2) to describe the friction surface. These three parameters provide a complete micro-topographic description of the friction surface.
2. A new model for calculating the wear process for sliding-friction pairs is proposed. An experimental-theoretical calculation principle can characterise it, according to which a running-in stage is necessary for selecting the input data, and the calculation is performed for the following normal (stable) service life. The wear calculation model can be applied at any point in the service life, ensuring that the running-in stage has been completed.
3. The wear values obtained with the new calculation model were compared with the material wear values obtained in the experimental studies for the sliding friction pair steel 102Cr6 (EN 1.2067) - bronze CW307G (EN 12163) at three different speed and load values.
4. Considering the wear values obtained in the experimental studies and comparing them with those obtained in the analytical calculations, it can be seen that the new wear calculation model allows the obtaining of reliable wear values in a sufficiently simple calculation and can be applied to solve practical engineering problems.
5. By varying parameters included in equation 17, it would be necessary to investigate the correlation between the initial values of the surface texture (3D) parameters and the values of these parameters after running-in stage, thus saving the time and resources needed to measure the surface parameters.

Acknowledgement

This work has been supported by the European Social Fund within the Project No 8.2.2.0/20/I/008 «Strengthening of PhD students and academic personnel of Riga Technical University and BA School of Business and Finance in the strategic fields of specialization» of the Specific Objective 8.2.2 «To Strengthen Academic Staff of Higher Education Institutions in Strategic Specialization Areas» of the Operational Programme "Growth and Employment".

REFERENCES

[1] V. L. Popov, "Generalized Archard Law of Wear Based On Rabinowicz Criterion Of Wear Particle Formation," *Facta Universitatis*, vol. 17, no. 1, p. 39, Mar. 2019, doi: 10.22190/fume190112007p.

[2] W. Xu, W. Li, and Y. Wang, "Experimental and theoretical analysis of wear mechanism in hot-forging die and optimal design of die geometry," *Wear*, vol. 318, no. 1–2, pp. 78–88, Oct. 2014, doi: 10.1016/j.wear.2014.06.021.

[3] A. A. Schmidt, T. Schmidt, O. Grabherr, and D. Bartel, "Transient wear simulation based on three-dimensional finite element analysis for a dry running tilted shaft-bushing bearing," *Wear*, vol. 408–409, pp. 171–179, Aug. 2018, doi: 10.1016/j.wear.2018.05.008.

[4] S. Reichert, B. Lorentz, S. Heldmaier, and A. Albers, "Wear simulation in non-lubricated and mixed lubricated contacts taking into account the microscale roughness," *Tribology International*, vol. 100, pp. 272–279, Aug. 2016, doi: 10.1016/j.triboint.2016.02.009.

[5] W. Cha, T. Hammer, F. Gutknecht, R. Golle, A. E. Tekkaya, and W. Volk, "Adaptive wear model for shear-cutting simulation with open cutting line," *Wear*, vol. 386–387, pp. 17–28, Sep. 2017, doi: 10.1016/j.wear.2017.05.019.

[6] D. Gao, L. Sun, and L. Jihong, "Prediction of casing wear in extended-reach drilling," *Petroleum Science*, vol. 7, no. 4, pp. 494–501, Nov. 2010, doi: 10.1007/s12182-001-0098-6.

[7] J. A. Brandão, R. C. Martins, J. Seabra, and J. Castro, "Calculation of gear tooth flank surface wear during an FZG micropitting test," *Wear*, vol. 311, no. 1–2, pp. 31–39, Mar. 2014, doi: 10.1016/j.wear.2013.12.025.

[8] K. Frischmuth and D. Langemann, "Numerical calculation of wear in mechanical systems," *Mathematics and Computers in Simulation*, vol. 81, no. 12, pp. 2688–2701, Aug. 2011, doi: 10.1016/j.matcom.2011.05.011.

[9] J. Mukherjee *et al.*, "Enhanced nano-mechanical and wear properties of polycarbosilane derived SiC coating on silicon," *Applied Surface Science*, vol. 325, pp. 39–44, Jan. 2015, doi: 10.1016/j.apsusc.2014.11.086.

[10] I. Khader, D. Kürten, and A. Kailer, "A study on the wear of silicon nitride in rolling–sliding contact," *Wear*, vol. 296, no. 1–2, pp. 630–637, Aug. 2012, doi: 10.1016/j.wear.2012.08.010.

[11] B. Dirks and R. Enblom, "Prediction model for wheel profile wear and rolling contact fatigue," *Wear*, vol. 271, no. 1–2, pp. 210–217, May 2011, doi: 10.1016/j.wear.2010.10.028.

[12] F. Lyu *et al.*, "Research on wear prediction of piston/cylinder pair in axial piston pumps," *Wear*, vol. 456–457, p. 203338, Sep. 2020, doi: 10.1016/j.wear.2020.203338.

[13] H. Kloß and R. Wäsche, "Analytical approach for wear prediction of metallic and ceramic materials in tribological applications," *Wear*, vol. 266, no. 3–4, pp. 476–481, Feb. 2009, doi: 10.1016/j.wear.2008.04.034.

[14] R. Tandler, N. Bohn, U. Gabbert, and E. Woschke, "Analytical wear model and its application for the wear simulation in automotive bush chain drive systems," *Wear*, vol. 446–447, p. 203193, Apr. 2020, doi: 10.1016/j.wear.2020.203193.

[15] A. Pronikov, *Parametricheskaya nadezdnostj mashin,* Moscow: Bauman Moscow State Technical University, 2002.

[16] I. V. Kragelsky, M. N. Dobychin, and V. S. Kombalov, *Friction and wear: Calculation Methods*. Elsevier, 2013.

[17] J. Rudzitis, *Surface contact mechanics: Sliding surface wear calculations*, Riga: RTU, 2007.

[18] E. Jansons, J. Lungevics, K. A. Gross, "Surface Roughness Measure that Best Correlates to Ease of Sliding," *Engineering for Rural Development*, pp.687-695, May 2016.

[19] N. Bulaha and J. Rudzītis, "Analysis of model and anisotropy of surface with irregular roughness," *Engineering for Rural Development*, May 2017, doi: 10.22616/erdev2017.16.n241.

[20] N. Bulaha, O. Liniņš, and A. Avišāne, "Application of 3D roughness parameters for wear intensity calculations," *Latvian Journal of Physics and Technical Sciences*, vol. 58, no. 5, pp. 27–37, Oct. 2021, doi: 10.2478/lpts-2021-0037.

[21] N. Bulaha, “Calculations of surface roughness 3D parameters for surfaces with irregular roughness,” *Engineering for Rural Development*, May 2018, doi: 10.22616/erdev2018.17.n256.

[22] G. Spriņģis, J. Rudzītis, A. Avišāne, and A. Leitāns, “Wear calculation for sliding friction pairs,” *Latvian Journal of Physics and Technical Sciences*, vol. 51, no. 2, pp. 41–54, Apr. 2014, doi: 10.2478/lpts-2014-0012.

[23] G. Springis, J. Rudzitis, E. Gerins, A. Leitans, “Rough Surface Peak Influence on the Wear Process of Sliding-Friction Pairs,” Engineering for Rural Development, vol, 15, pp. 1430-1436, May 2016.

[24] E. Students, *Wear Calculation of Sliding Friction Surfaces*, PhD thesis, Riga Technical University, 1996.

[25] G. Konrads, Wear on sliding surfaces of machine parts, Riga: Riga Technical University, 2006. (in Latvian).

[26] J. Rudzitis, Contact mechanics of surfaces. 2nd part, Riga: Riga Technical University, 2007. (in Russian).