Wear Calculation Approach for Sliding - Friction Pairs

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Abstract. One of the most important things how to predict the service life of different products is always connected with the choice of adequate method. With the development of production technologies and measuring devices and with ever increasing precision one can get the appropriate data to be used in analytic calculations. Historically one can find several theoretical wear calculation methods but still there are no exact wear calculation model that could be applied to all cases of wear processes because of difficulties connected with a variety of parameters that are involved in wear process of two or several surfaces. Analysing the wear prediction theories that could be classified into definite groups one can state that each of them has shortcomings that might impact the results thus making unnecessary theoretical calculations. The offered wear calculation method is based on the theories of different branches of science. It includes the description of 3D surface micro-topography using standardized roughness parameters, explains the regularities of particle separation from the material in the wear process using fatigue theory and takes into account material’s physical and mechanical characteristics and definite conditions of product’s working time. The proposed wear calculation model could be of value for prediction of the exploitation time for sliding friction pairs thus allowing the best technologies to be chosen for many mechanical details.

1. Introduction

With the development of production technologies and materials the task concerning prediction of service life of fitted components and factors influencing it is of crucial value. Historically, several models of the wear process were developed which allow approximate prediction of the friction pairs service life. Due to the fact that the wear process is multiflorous it is affected by different parameters: the geometry of surface peaks (roughness, buckles, shape deviations, etc.), physically-mechanical conditions of the upper layer, material of contacting components, frictional heating during the wear process, wear conditions, etc.. These factors are impossible to take into consideration in the analytical calculation; therefore, the wear calculation development in the course of time went in different directions taking into account the main influencing factor complexes.

There are several wear calculation models based on the application of theories from several branches - the probability theory when predicting the fittings service life [6], the laws of classical physics [7], the analysis of various geometrical forms of fittings [8] and combined sliding-friction pairs calculation method [9], but none of these models contains technologically manageable surface roughness parameters.

The article deals with the wear calculation model in which the wear process is characterized by three stages seen in figure 1 [5].
Figure 1. Wear U dependent from duration.

Figure 1 (a) shows three stage wear. I stage is the run in process, II stage is normal wear and III stage is the critical wear (b) shows the nominal wear period.

According to figure 1:

$$U_{\text{max}} = U_p + U_n$$

where $U_p$ and $U_n$ is average wear value for run-in and normal exploitation stages.

In the given model run-in process wear ($U_p$) is determined experimentally by measuring it. It is so because during the run-in process there are large amount of parameters who are constantly changing and they cannot be calculated. For the offered model, run-in wear $U_p$ will be determined experimentally and normal exploitation time wear $U_n$ - by calculations.

That is the reason why this methodology is experimentally theoretical. Previously mentioned facts reflects in figure 1 (b) where average wear in normal exploitation time is linear.

This process is described with equation:

$$U_n = h_{\text{def}} \frac{N_{cf}}{N_c}$$

where $h_{\text{def}}$ is friction surface deformed layer height (see figure 2), $N_{cf}$ is number of cycles that cams are exposed to, $N_c$ is the number of cycles that surface cams can withstand before shearing off.

Figure 2. Potential wear particle separation from surface where $u$ is unevenness deformation level, $h_v$ - unevenness height.

2. Wear calculation principles

Wear calculations are based on following principles:

- Wear pair surface roughness is described with normal random field $h(x,y)$ (Roughness characteristics).
- Wear process is conducted with the principles of the theory of fatigue (Surface fatigue wear principles).
- Roughness unevenness contact correspond to Hertz theory (Roughness contact).

2.1. Roughness characteristics
Random function is a function the value of which at each specified argument (or several arguments) value is a random size. When the random function depends on several parameters it is called a random field. The random field in case of threadbare surfaces can be normal, i.e. ordinates of such field are distributed according to the normal distribution law which is characterized by height parameter \( \sigma \) (standard deviation). Besides the random field can be uniform (stationary) and non-uniform. The random field is called stationary if its average size is constant and the correlation function depends on the difference between the surface points. 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, standard deviation \( \sigma \) and correlation function should be known [1]. According to normal random field principles, surface characteristic parameters are calculated [1]:

**Average arithmetic deviation \( Sa \):**

\[
E\{Sa\} = \sqrt{\frac{2}{\pi}} \sigma
\]

where \( E\{Sa\} \) is mathematical expectation value.

**Surface anisotropy coefficient \( c \) see figure 3 [2]:**

\[
c = \frac{E\{RS_{m1}\}}{E\{RS_{m2}\}} = \frac{E\{n_x(0)\}}{E\{n_y(0)\}}
\]

Where \( RS_{m1} \) is a step perpendicular to the processing trace direction along midline and \( RS_{m2} \) is a step towards the processing trace along the midline, \( n_x(0), n_y(0) \) are the numbers of zeros in two mutually perpendicular directions of surface cuts \( x \) and \( y \).

![Figure 3. Irregular surface roughness spacing parameters.](image)

**Peak count per area:**

\[
E\{N\gamma\} \approx \frac{\pi E\{n_x(0)\} E\{n_y(0)\}}{2\sqrt{2\pi}} e^{-\frac{\gamma^2}{2}}
\]

where \( \gamma \) is surface roughness relative deformation level (\( \gamma = u/\sigma \)).

2.2. **Surface fatigue wear principles**

Various independent researchers have shown that material fatigue is one of the main reasons that cause wear of the surfaces. Fatigue crack formation is inevitable due to the mutual movement of two body’s that causes the fracture of the surface asperities. In order to better understand the process it is useful to take a look at two body contact model see figure 4.

![Figure 4. Two body asperity contact model. (a) is at macro scale and (b) is at micro scale.](image)

According to Hamilton’s theory [3] contacting surface asperities are exposed to tension witch is the cause of material upper layer collapse. If Huber - Mises - Genki’s criterion is used one can calculate
material yield strength. The following figure shows tension field for one surface asperity where contact area ellipse semi-major axle is represented as $a$ and the tension is represented as contact maximums pressure.

![Tension field figure](image)

**Figure 5.** Surface tension field whit friction coefficient (a), (b).

Figure 5 shows that if coefficient of friction tension maximum is approximately located at distance $0.5a$ from the contact surface but if tension maximum moves towards the material surface. Besides that increase also causes formation of second (submaximum) pressure point see figure 5 (b). If contacting bodies move along each other, tension fields change their location. As far as possibility that during wear process only two asperities are in contact is impossible one must analyse system as complex surfaces with many asperities as shown below.

![Tension fields](image)

**Figure 6.** Irregular roughness surface where (a) asperity contact and (b) tension fields.

During such surface movement, every asperity, which is high enough to reach some other asperity from second surface, is deformed thus creating tension field. Figure 5 (b) shows that surface tension in such model changes according to asymmetric cycle. According to fatigue theory for asymmetric cycles, elastic limit $\sigma_r$ is $\sigma_0$ because index $r$ is calculated as follows:

$$ r = \frac{\sigma_{\text{min}}}{\sigma_{\text{max}}} = 0 $$

(6)

Acquired equation allows calculating maximum amount of asperity bending cycles $N_c$ until the material cracks [5]:

$$ E\{N_c\} = \frac{N_0}{5m} t_o^m $$

(7)

where $N_0$ is number of asperity bending cycles until cracks, $m$ is indicator of fatigue curve degree, $t_o$ is dimensionless tension ratio.

Parameters $N_0$, $m$ and $\sigma_0$ are chosen as average values for specifically used material that receives tension.

2.3. Roughness contact.

Friction surface asperities projects as ellipses. According to Hertz theory [10]:

*...*
$$b_i = \left[ \frac{3E(e)\theta P_e}{2(1-e^2)H} \right]^{\frac{1}{e}} \quad (8)$$

$$a_i = b_i(1-e^2)^{\frac{1}{e}} \quad (9)$$

$$\alpha_i = \frac{3}{2} K(e) \frac{\theta P_e}{b_i} \quad (10)$$

where, $K(e)$, $E(e)$ are first and second degree elliptical integrals, $a_i$, $b_i$ are elliptical contact area semi-minor and semi-major axis, $a_i$ is deformation of $i$ asperity, $e$ is eccentricity of elliptic contact area ($e = \sqrt{1-e^2}$), where $c$ is anisotropy coefficient (see equation (4)), $\theta$ – deformation level ($\theta = \frac{1-\mu^2}{\pi E}$), $E$ – modulus of elasticity, $\mu$ - Poisson’s ratio, is average curvature of asperities:

$$H=(k1+k2)/2 \quad (11)$$

3. Summary of wear calculations

Based on previously viewed principles and equation (2) linear wear can be determined:

$$E(h_{as}) \approx E\{V_i\} E\{N_i\} \approx \frac{\sigma}{6\pi r^2} \quad (12)$$

where $E\{V_i\}$ is separated volume average value for one i-asperity.

$$E\{V_i\} \approx \frac{\sigma}{\gamma^2 \pi E(n_i(0)) E(n_i(0))} \quad (13)$$

Based on literature [4]:

$$\gamma^2 = \frac{1}{5} \left( \frac{S a}{R S m_i} \right)^{\frac{1}{3}} \left( \frac{E}{q} \right)^{\frac{2}{3}} \quad (14)$$

where $q$ is pressure on contacting surfaces.

Required cycles for material destruction are calculated [5]:

$$E\{N_c\} = \frac{N_0}{5m!} \left( \frac{\sqrt{2} \sigma e R S m_1 k^{1/2}(e)}{E \cdot S a} \right)^m \quad (15)$$

Actual number of cycles $N_{cf}$ is calculated:

$$N_{cf} = \frac{L_b}{S_m^{a-2}} \quad (16)$$

where $L_b$ is friction path length and $S_m^{a-2}$ is surface roughness average pitch in friction direction for the part than promote wear.

Inserting obtained coherences in formula (2) average linear wear can be calculated:

$$E\{U_n\} \approx k_{e-m} \cdot k_{r} \cdot k_{f-m} \cdot \left( \frac{q}{E} \right)^{\frac{2}{3}} \cdot S a \cdot \frac{L_b}{S_m^{a}} \quad (17)$$

where $k_{e-m}$ is coefficient dependent from surface anisotropy ($e^2 = 1 - e^{8/5}$) and fatigue curve parameters:

$$k_{e-m} = \sqrt{\pi m!} \frac{\pi^4}{N_0} \left( \frac{2K(e)}{2} \right)^{\frac{a}{2}} \quad (18)$$
$k_r$ is surface roughness parameter complex:

$$k_r = \left( \frac{S_a}{RS_{m1}} \right)^{\frac{2}{3}}$$  \hspace{1cm} (19)

$k_{f-m}$ is physical and mechanical parameter complex:

$$k_{f-m} = \left( \frac{E}{\sigma} \right)^m \hspace{1cm} (20)$$

Average linear wear in normal exploitation conditions can be calculated using equation (17). These conditions should be met:

- friction coefficient $f \leq 0.1$;
- both friction surfaces must be flat.

Total wear is determined using equation (1).

Equation (17) can be rewritten when taking into account that:

$$L_b = vt$$  \hspace{1cm} (21)

where $v$ is friction pair speed, $t$ – friction pair duration.

Then $E\{U_n\}$ is:

$$E\{U_n\} \approx k_{e-m} \cdot k_r \cdot k_{f-m} \cdot \left( \frac{q}{E} \right)^{\frac{2}{3}} \cdot \frac{Sa}{S_{m2}} \cdot vt$$  \hspace{1cm} (22)

### 4. Experimental studies.

To verify the theory an experimental study was carried out using “pin-on-disc” type tribometer. A special disk (38KHN3MA steel) with diameter 100mm and thickness of 6mm was manufactured to work as a tribometers’s disc and bronze (BrOF7-0.2) rod with a diameter of 6.5mm was used as pin. 3D surface roughness was determined with “Taylor Hobson Form Talysurf Intra 50” profilometer. Experimental study parameters are shown in table 1.

| Table 1. Experiment parameters |
|--------------------------------|
| Kinematic parameters | Physical and mechanical parameters | Roughness parameters |
| q=11 MPa | m=4 | S$_a$=0.6 µm |
| v=5 m/s | $\sigma_1$=600 MPa | RS$_{m1}$=0.05 mm |
| $t_{\max}$=272 h | $N_0$=5*10$^6$ | RS$_{m2}$=0.5 mm |
| | E=1.05*10$^5$ MPa | |
Figure 7. Linear wear experimental and theoretical data comparison where (a) total experimental data and (b) experimental and theoretical data comparison for normal exploitation stage.

Experiment results are summarized in figure 7 where (a) shows both experiment stages - run-in and normal wear. Total duration time of the experiment took 272 hours. Experiment was paused after 16 hours (run-in) to measure surface roughness parameters used in further calculations. In figure 7 (b) one can see comparison of experimental data after the run-in process and theoretically calculated curve. Strong correlation is obtained for experimental and theoretical curves with result scatter less than 10%.

5. Conclusion

Paper offers experimental-theoretical methodology to determine slide frictions wear. It is carried out in a sequence:

- Determine starting data that are necessary for calculations:
  - kinematic properties: load ($q$), speed ($v$), travelled distance ($L_b$), duration ($t$);
  - material physical properties: material fatigue fracture parameters ($m$, $\sigma_0$, $N_0$);
  - material mechanical properties ($E$, $\mu$)

- Determine parameters after run-in:
  - Surface roughness parameters ($S_a$, $R_{Sm1}$, $R_{Sm2}$, $S_{m2''}$);
  - run-in wear ($U_p$) and its duration ($T_p$).
6. Literature

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