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Analytical evaluation of materials wear resistance and tribotechnical coupling resource

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Abstract. When predicting the durability of various friction units at the design stage of the machine, an experimental-analytical standard of wear resistance is used. Its definition requires long-term and expensive experimental studies of samples or full-scale analogs. In order to reduce these costs, in this work, the fundamental theoretical dependences are derived for evaluating the wear resistance and the life of tribocoupling operating in stationary conditions of friction and fatigue wear. The basis for building the desired dependencies is a thermodynamic analysis of the steady-state process of friction, as a dual, molecular-mechanical process. The verification of the obtained theoretical dependences was carried out by comparing the calculated and experimental values of the wear resistance of samples abraded by the FCM-1 friction commercial machine according to the standard "steel roller - cast iron pad" under various conditions of friction interaction. The obtained results showed a fairly high adequacy of analytical dependencies. The proposed equations can be used to calculate the expected life of various tribocoupling at the design development stage of machines or their reconstruction.

1. Research target setting

Traditionally, the tribocoupling resource at the design stage of a machine is determined by the criteria of their wear resistance [1 - 4]. According to GOST 23.001 and GOST 27.674 [5, 6], the coupling wear resistance, as a property of triboelements materials resisting wear in certain friction conditions, is evaluated by a semi-empirical indicator:

$$И = L \,/\, y \;= V_{ск} / \dot{y}\,, \qquad (1)$$

where $L_t = V_{ск} \cdot t$ - the calculated relative slip path of the triboelements being worn over the surface of the solid counter body associated with it at a rate $V_{ск}$ in time t;

$y = \dot{y} \cdot t = y_1 + y_2$ - measured linear wear over time t with average speed $\dot{y}$;

However, the need for long-term and expensive experiments, in determining this indicator, leads to the problem of estimating the design life of a machine because of the large number of studied tribocouplings that limit its durability.

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In order to solve this problem, this paper proposes the method for analytic design assessment of wear resistance and coupling life without conducting experimental studies.

2. The derivation of the basic equations for the design assessment of tribocoupling durability and resource

In order to simplify the mathematical calculations, we will consider stationary coupling operating under steady-state temperature and force conditions of friction and fatigue wear.

The basis for building the desired dependencies is a thermodynamic analysis of the steady-state process of friction, as a dual, molecular-mechanical process. From an energetic point of view, a tribocoupling operating under steady-state conditions can be viewed as a stationary thermodynamic system, and the energy conservation law in tribocoupling can be written in the form of the energy balance equation [7]:

$$A_{tr} = F_t \cdot L = f \cdot F_n \cdot L = \Delta U_1 + \Delta U_2 , \tag{2}$$

where f, F_t, F_n - friction coefficient, friction force and normal load in coupling respectively;
$L = V_{sk} \cdot t$ - sliding distance, V_{sk}, t - sliding speed and coupling time.

The change in the total internal energy (ΔU_1 and ΔU_2) of the material contact volumes of each coupling element can be expressed by the corresponding coefficient of external energy absorption A_{tr}:

$$\nu_1 = \Delta U_1 / A_{tr} \qquad \text{and} \qquad \nu_2 = \Delta U_2 / A_{tr} , \tag{3}$$

and the values ν_1 and ν_2 can be determined using B.V. Protasov's energy distribution coefficient between the elements of the tribocoupling $\varepsilon = \Delta U_1 / \Delta U_2$:

$$\nu_1 = \varepsilon/(1+\varepsilon) \qquad \text{and} \qquad \nu_2 = 1/(1+\varepsilon) , \tag{4}$$

where $\varepsilon = \dfrac{((1-\mu_1^2)/E_1)^{2/3} \cdot R_{a1}^{1/3}}{((1-\mu_2^2)/E_2)^{2/3} \cdot R_{a2}^{1/3}}$;

μ_1, μ_2, E_1, E_2 – Poisson's coefficients and elastic moduli of the surface layer materials of the first and the second coupling elements;

R_{a1}, R_{a2} – the arithmetic average height of microroughnesses in the profile of both surfaces.

Solving equations (3) for ΔU_1 and ΔU_2, we obtain the expression of the energy balance for each element of the tribocoupling:

$$\Delta U_1 = \nu_1 \cdot A_{tr} \qquad \text{and} \qquad \Delta U_2 = \nu_2 \cdot A_{tr} . \tag{5}$$

In turn, according to V.V. Fedorov's ergo dynamic theory of plastic deformation and destruction of solids [1, 7 - 9], the change in total internal energy ΔU_1 and ΔU_2 can be represented as the sum of two components:

$$\Delta U_1 = \Delta U_{e1} + Q_1 \qquad \text{and} \qquad \Delta U_2 = \Delta U_{e2} + Q_2 , \tag{6}$$

where ΔU_{e1} and ΔU_{e2} – changing the latent (potential) energy of the crystal structure of materials of contact volumes of friction pair elements;

Q_1 and Q_2 – friction heat effect.

Expanding in equations (5) the quantities ΔU_1 and ΔU_2 in accordance with (6), we obtain the energy balance equations for each element of the tribocoupling in the form of the first law of thermodynamics:

$$\nu_1 \cdot A_{tr} = \nu_1 \cdot F_n \cdot f \cdot L = \Delta U_{e1} + Q_1 \quad \text{and} \quad \nu_2 \cdot A_{tr} = \nu_2 \cdot F_n \cdot f \cdot L = \Delta U_{e2} + Q_2 . \tag{7}$$

The physical meaning of the components of equation (7) is defined by S.V. Fedorov [7].

One part of the friction work is transformed into a change in the latent potential energy ΔU_{e1} and ΔU_{e2} of various kinds of elementary defects and damage to the microstructure of deformable volumes of the surface layers of both bodies, determining the measure of their strain hardening, damage and destruction. It is this part of the external energy that reflects the true resistance of the deformable volumes of materials of the contact layers to the relative displacement of surfaces. It is accumulated - it is “destroyed” by friction, “being accumulated” in the structure of these volumes in the form of the potential energy of elastic distortions of the crystal lattice.

The second fraction of the friction work is converted into internal energy Q_1 and Q_2 of the vibration, thermal atoms motion of the surface layers of both elements, defining the “thermal effect of friction”. A smaller part of this effect is transformed into the energy of thermal motion of atoms of the crystal structure of deformable volumes. It causes softening and release of the latent energy of defects, an increase in the intensity of atomic vibrations and an increase in the temperature of the surface layers in stationary conditions to values of $T_1 = const$ and $T_2 = const$. Most of the thermal effect is dissipated into the environment.

If expressions (7) are solved with respect to the friction coefficient, we obtain the binomial energy balance equations for each element of the tribocoupling:

$$f = \frac{\Delta U_{e1}}{\nu_1 \cdot F_n \cdot L} + \frac{Q_1}{\nu_1 \cdot F_n \cdot L}; \qquad f = \frac{\Delta U_{e2}}{\nu_2 \cdot F_n \cdot L} + \frac{Q_2}{\nu_2 \cdot F_n \cdot L}. \tag{8}$$

From the standpoint of I.V. Kragelsky’s theory of friction [2], the friction force has a dual, molecular-mechanical nature, and the work of the friction force on the contact can be expressed by the sum of the mechanical F_{mech} and molecular F_{mol} components of the total friction force F_t:

$$A_{tr} = F_t \cdot L = F_{tmech} \cdot L + F_{tmol} \cdot L = f_{mech} \cdot F_n \cdot L + f_{mol} \cdot F_n \cdot L. \tag{9}$$

From equation (9) it follows that the total coefficient of friction is the sum of the mechanical and molecular components:

$$f = f_{mech} + f_{mol}. \tag{10}$$

From a comparison of expressions (8) and (10), which determine the friction coefficient from the energy and molecular mechanical positions, respectively, we can estimate the components of the friction coefficient in the form:

- a mechanical component

$$f_{mech} = \frac{\Delta U_{e1}}{\nu_1 \cdot F_n \cdot L}; \qquad f_{mech} = \frac{\Delta U_{e2}}{\nu_2 \cdot F_n \cdot L}; \tag{11}$$

- molecular component

$$f_{mol} = \frac{Q_1}{\nu_1 \cdot F_n \cdot L}; \qquad f_{mol} = \frac{Q_2}{\nu_2 \cdot F_n \cdot L}. \tag{12}$$

Due to the fact that irreversible distortions (defects) of the surface layers structures are determined by the amount of accumulated potential energy ΔU_{e1} and ΔU_{e2}, to calculate the characteristics of their destruction (wear), you can use equations (11), in the following form:

$$\Delta U_{e1} = \nu_1 \cdot f_{mech} \cdot F_n \cdot L; \qquad \Delta U_{e2} = \nu_2 \cdot f_{mech} \cdot F_n \cdot L. \tag{13}$$

Expressing in each equation the total potential energy (ΔU_{e1} and ΔU_{e2}) accumulated on the friction path $L = V_{sk} \cdot t$ (during operation time t) by the materials of the surface layers of each body as a product of the corresponding energy density (Δu_{e1} and Δu_{e2}) and the deformable volume of the surface layer (V_{cL1} and V_{cL2}), we obtain:

$$\Delta u_{e1} \cdot V_{cL1} = \nu_1 \cdot f_{mech} \cdot F_n \cdot L\,; \qquad \Delta u_{e2} \cdot V_{cL2} = \nu_2 \cdot f_{mech} \cdot F_n \cdot L\,. \tag{14}$$

The mechanical component of the friction coefficient under the conditions (13) can be determined by N.M. Mikhina's method [10] depending on the contact type:

$$f_{mech} = a \cdot \alpha_g \cdot \Delta \cdot (p_a \cdot \theta)^b\,, \tag{15}$$

where α_g, θ, p_a - hysteresis loss coefficient, the material elastic constant of the element being worn, nominal pressure on the contact, respectively;

a, b - constants corresponding to a particular contact type;

Δ - complex parameter of the surface roughness of the counter element, determined from reference data for run-in surfaces [2, 10, 11].

According to the ergo dynamic theory of the destruction of solids, those local volumes of the surface layer ΔV_1 and ΔV_2 of all triboelements, in which by the time moment t, the values Δu_{e1} and Δu_{e2} reach the limiting value:

$$\Delta u_{e1} = \Delta u_{e1}^* \qquad \text{and} \qquad \Delta u_{e2} = \Delta u_{e2}^*\,, \tag{16}$$

where $\Delta u_{e1}^* = \Delta H_{S1} - u_{e1}(0) - \Delta u_{T1}$ and $\Delta u_{e2}^* = \Delta H_{S2} - u_{e2}(0) - \Delta u_{T2}$ - critical density of the internal potential energy of defects;

ΔH_{S1} and ΔH_{S2} - the fusion enthalpy of triboelements materials in the liquid state;

$u_{e2}(0)$ and $u_{e2}(0)$ - density of the potential component of the internal energy of the materials of the surface layers in the initial state, determined as a function of surface hardness by V.V. Fedorov's equation [8, 11];

$\Delta u_{T1} = \rho_1 \cdot c_1 \cdot T_1$ and $\Delta u_{T2} = \rho_2 \cdot c_2 \cdot T_2$ - a density change of the kinetic component of the internal energy of destroyed local materials of the surface layers by the time moment t in triboelements heated to temperatures T_1 and T_2;

ρ_1, c_1 and ρ_2, c_2 - density and heat capacity of triboelements materials.

In this case, the volumetric wear of each element of the stationary coupling in the friction path (or during operation time t) will be:

$$\Delta V_1 = \frac{\nu_1 \cdot F_n \cdot f_{mech} \cdot V_{sk} \cdot t}{\Delta u_{e1}^*}\,; \qquad \Delta V_2 = \frac{\nu_2 \cdot F_n \cdot f_{mech} \cdot V_{sk} \cdot t}{\Delta u_{e2}^*}\,. \tag{17}$$

Linear wear of triboelements and coupling in general:

$$y_1 = \frac{\Delta V_1}{A_{T1}}\,; \qquad y_2 = \frac{\Delta V_2}{A_{T2}}\,, \qquad y = y_1 + y_2\,. \tag{18}$$

Linear wear rate of triboelements and coupling in general:

$$\dot{y}_1 = \frac{\Delta V_1}{A_{T1} \cdot t} = \frac{y_1}{t}\,; \qquad \dot{y}_2 = \frac{\Delta V_2}{A_{T2} \cdot t} = \frac{y_2}{t}\,; \qquad \dot{y} = \dot{y}_1 + \dot{y}_2\,. \tag{19}$$

Here A_{T1}, A_{T2} - the friction area of the first and second elements of the pair, respectively.

Analytical indicators of triboelements wear resistance and coupling in general:

$$И_1 = \frac{V_{sk}}{\dot{y}_1}\,; \qquad И_2 = \frac{V_{sk}}{\dot{y}_2}\,; \qquad И = \frac{V_{sk}}{\dot{y}}\,. \tag{20}$$

The expected resource of the tribocoupling is determined by the specified limit value of its linear wear $[y]$:

$$t_{pr} = [y] / \dot{y}. \tag{21}$$

3. Verification of theoretical results

The reliability of the proposed technique was checked by comparing the calculated and experimental values of wear resistance of samples abraded by the FCM-1 friction machine according to the roller-block scheme under various conditions of frictional interaction (at a relative slip speed $V_{sk} = 1{,}5\, m/\sec$ and normal pressing force of the samples $P = F_n = 200\, H$) with intensive cooling of the contact with running water.

The tests consisted of 7 series of experiments, with 4-fold repetition in each series. In a separate series of experiments, pads of cast iron (GG-30) were worn out on a roller of steel 45. The series differed from each other in the duration of the tests t_i, where $i = 1-7$, the values of which are presented in the Table.

Experimental average values of wear resistance of cast iron pads $И_{ch}$, steel rollers $И_{st}$ and coupling $И$ as a whole in each series of experiments were determined by losing their mass by weighing on the electronic discharge scales IU 215S of the 1st category before and after the tests (see the Table). In tests, the contact temperature was recorded with a CONDTROL IR-T4 infrared thermometer, and the surface roughness of the samples was measured with a Perthometer S2 MAHR instrument before and after the experiment.

The comparison results for all series of experiments were determined by the magnitude of the calculation errors $\delta_И$ and are presented in the Table.

Table. Comparison of experimental and calculated values of sample wear resistance

Experiment series №	Test Period, sek		Wearability indicator values			Miscalculations		
			$И_{ch} \cdot 10^9$	$И_{st} \cdot 10^9$	$И \cdot 10^9$	δ($И_{ch}$)	δ($И_{st}$)	δ($И$)
1	7200	Exp.	0,5466	2,836	0,4583	8,89	21,04	2,60
		Theor.	0,5952	2,239	0,4702			
2	14400	Exp.	0,4918	2,110	0,3988	20,08	5,33	16,98
		Theor.	0,5906	2,222	0,4666			
3	21600	Exp.	0,6961	2,609	0,5495	27,51	28,06	27,63
		Theor.	0,8876	3,341	0,7013			
4	28800	Exp.	0,895	2,930	0,6856	1,58	16,63	4,74
		Theor.	0,9091	3,417	0,7180			
5	36000	Exp.	1,026	3,132	0,7728	24,96	54,02	31,06
		Theor.	1,282	4,823	1,013			
6	43200	Exp.	1,164	3,178	0,8518	14,07	56,81	23,06
		Theor.	1,327	4,983	1,048			
7	50400	Exp.	1,327	7,538	1,129	45,43	10,76	32,90
		Theor.	1,931	6,726	1,500			

The results of the comparative analysis given in the Table show that the error in predicting the values of wear-resistance of triboelements and coupling in general are in the range: for cast-iron pads - $\delta(И_{ch}) = (1{,}5-45)\ \%$; for steel rollers $\delta(И_{st}) = (5-54)\ \%$ and $\delta(И) = (2-33)\ \%$, respectively.

The presented results of the analysis testify to a fairly high accuracy of theoretical developments in the calculation of the wear resistance and durability of the triple conjugations according to equations (20) and (21). This, in turn, makes it possible to evaluate the technical condition of the tribocoupling and its level of performance (reliability) at the design stage under the expected operating conditions at

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any fixed point in time in the future or the maximum duration of its operation (time to failure, resource).

4. Conclusion
Thus, the proposed method makes it possible to evaluate the tribocoupling wear resistance and life at the design development stage without laboratory or industrial experiments.

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