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WEAR RESISTANCE OF STEEL IN LINEAR CONTACT ON COMPOSITE MATERIALS

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Generally one could make a tribological characterization of frictional couples based on a theoretical contact model and some numerical results obtained on an experimental model built on the theoretical basis. This procedure offers characterization data only for one couple, with the same kind of contact and the same loading conditions, without any possibility of generalization. The present study introduces the comparative wearing coefficient concept as criteria for the evaluation of the behavior of tribological couples with the same kind of contact (linear one) and obtained from the same category of materials (e.g. composite thermoplastics/steel) as a function of the length of frictional path.

Keywords: friction, wear, composite thermoplastics, comparative wearing coefficient.

1. INTRODUCTION

As it is already known [2], the characterization of the wearing rate of a material could be made by a wearing factor k . This factor is defined by relation:

$$V_u = kNvt \tag{1}$$

where: V_u - the volume of the wear material (cm^3); N - the test load (daN); v - the relative sliding speed (cm/s); t - the test period ($hours$); k - wearing factor ($cm^3 s/daNmh$).

Dividing the both terms of relation (1) by nominal contact area A, we obtain:

$$V_u / A = kvtN / A \tag{2}$$

It means:

$$h_u = kpvt \tag{3}$$

where: h_u - the depth of wear material (cm); p - the pressure on the nominal contact area (daN/cm^2).

The relation (3) expresses a general law of the wear as function of the contact pressure p and the length of the wearing path, so that $L_f = vt$.

We could then write:

$$k = h_u / pvt = h_u / pL_f \qquad \left(cm^2/daN\right) \tag{4}$$

respectively:

$$k = V_u / Nvt = V_u / NL_f \qquad \left(cm^2/daN\right) \tag{5}$$

Considering the large area of the load (N) or pressure (p) and the relative speed values used in tests for the evaluation of wearing behavior of the metallic counter-pieces of the frictional couples, we will use comparative wearing coefficients K and K^* , defined by:

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$$K = V_u / L_f = kN \qquad \left(cm^3/cm\right) \tag{6}$$

respectively:

$$K^* = h_u / L_f = kp \qquad \left(cm/cm\right) \tag{7}$$

These wearing coefficients were considered with respect to the period in which the frictional couple functions at different sliding speeds, under certain loading conditions (pressure).

The wear tests that were made have as main objectives the determination of the volume of material removed by wearing, the mean depth of the wearied layers, the frictional factors and coefficients, for different loading conditions.

2. ANALITICAL METHOD

The Timken frictional couple (with linear contact) under loading reveals the appearance of some wearing trace on the plane surface of the metallic material. The wearing trace is produced by the penetration of the plane semi couple material by the cylindrical bush (fig.1).

Theoretically, considering the bush as rigid and accounting for the generally low non-uniformity of the imprint, this could be considered as being formed by a series of cylindrical sectors having the length q .

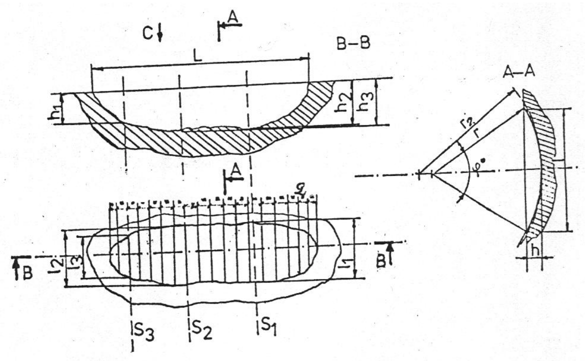

Fig.1. The aspect of wear imprint of metallic surface. Analytical method application to volume calculus of wear imprint.

Assuming that, the area of the lateral surface of the cylindrical sector is a circle segment, it results:

$$S_i = 0.5r^2\left(\pi\varphi_i^0/180° - \sin\varphi_i\right) \tag{8}$$

where: S_i - lateral surface of the cylindrical sector; φ_i - the angle; r - the circle radius

The radius r could not be identified with the cylindrical bush radius for the plastic/metal couples. And this fact is possible due to the elastic deformation of the bush under loading conditions, which has as effect the increment of the radius in the contact area. We illustrate this by the sketch plotted in figure 2.

Using r_1 for the undeformed bush radius and r_2 for the radius – in the contact area – of the deformed bush, we could notice from figure 1b that $r_2 > r_1$.

Increasing the bush radius in the contact area conducts to the decrease of the depth of the wearing trace from h_1 - (figure 2a), which would appear if the elastic deformation of the bush would be neglected, to the value h - (figure 2b), with the quantity h_2 :

$$h_2 = h_1 - h \tag{9}$$

Using l for the width of the wear imprint, from ΔABC , it results:

$$\left(2r_1 - h_1\right)h_1 = l^2/4$$

Because the value of the depth h_1 is very small, the term h_1^2 is neglectable and we could write:

$$h_1 = l^2/8r_1 \tag{10}$$

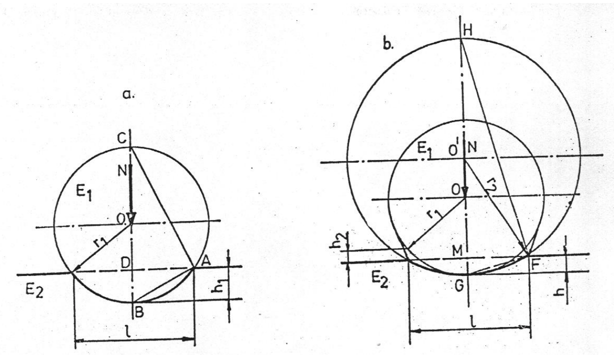

Fig.2. The elastic deformation of the cylindrical bush in the contact area for Timken frictional couples (a – theoretical; b – practical)

Similary, in ΔFGH , we have:

$$(2r_2 - h_2)h_2 = l^2/4$$

Using the same assumption, for the term h_2,we obtain:

$$h_2 = l^2/8r_2 \tag{11}$$

Introducing (10) and (11) in (9) it results:

$$h = l^2(1/r_1 - 1/r_2)/8 = l^2(r_2 - r_1)/8r_1r_2 = l^2/8r \tag{12}$$

where r is the equivalent curvature radius given by:

$$1/r = 1/r_1 - 1/r_2 = (r_2 - r_1)/r_1r_2 \tag{13}$$

From (12) it results:

$$(r_2 - r_1)/r_1r_2 = l^2/8h_2 \tag{14}$$

Considering that the frictional couple is loaded in the elastic domain with an elliptic distribution of stresses, the Hertz formula for computing the width of the wear imprint is:

$$l^2/4 = 8Nr(1-\nu^2)/\pi EL \tag{15}$$

where: ν - Poisson ratio; L - the length of the wear imprint; E - equivalent Young modulus.

Using index *1* for quantities related to the cylindrical bush, and index *2* for those related to plane half-couple, the equivalent elasticity modulus is given by:

$$1/E = 0.5\left[(1-\nu_1^2)/E_1 + (1-\nu_2^2)/E_2\right] \tag{16}$$

Because the numerical values of ν_1 and ν_2 are between 0.3 and 0.32 the equivalent elasticity modulus could be approximated by:

$$E = 2E_1E_2/0.91(E_1 + E_2) \tag{17}$$

From (15) we could express the width of the wear imprint:

$$l = 4\left[2Nr\left(1-\nu^2\right)/\pi EL\right]^{1/2} \tag{18}$$

Introducing in (18) the equivalent elasticity modulus and the equivalent radius expressions, the numerical value of Poisson ratio, one could obtain:

$$h_2 = 0.527N\left(E_1 + E_2\right)/LE_1E_2 \tag{19}$$

Considering the relations (10) and (19), we have for the depth of the wear imprint the expression:

$$h = \left(l^2/8r_1\right) - 0.527N\left(E_1 + E_2\right)/LE_1E_2 \tag{20}$$

Assuming that the wear imprint is the sum of some cylindrical sectors, expanding in series the relation (8), neglecting the high-order terms and reducing the similar terms we could obtain, for the area of the lateral surface of a sector, the expression:

$$S_i = r^2\phi^3/12 \tag{21}$$

Replacing in the relation above the angle ϕ with the ratio l/r and accounting for (13) and (14), we obtain:

$$S_i = l^3\left(r_2 - r_1\right)/12r_1r_2 = 2lh_2/3 \tag{22}$$

Replacing the value of h_2 obtained from (20) in (22) we could obtain the expression for the area of lateral transversal surface of a cylindrical sector:

$$S_i = 0.35l\left(E_1 + E_2\right)Nl/E_1E_2L \tag{23}$$

The volume of weared metalic material will be:

$$V_u = \sum_{i=1}^{n}\left(S_iq_i\right) = 0.351\left(E_1 + E_2\right)Nl_m/E_1E_2 \tag{24}$$

where l_m is the mean width of the wear imprint.

Practicaly, it is needed to measure the width of wear imprints in three points established before, computing then the mean value of this width. With this value we could obtain the volume of weared metalic material V_u and the mean of the depth of removed layer h_{mu}.

3. EXPERIMENTAL PROCEDURE

The study of wearing of the metalic component of the frictional couple was made on a Timken machine, with linear contact, presented in fig.3. Almost all tests are made without lubricating the frictional surfaces, but there are also tests with micro-lubrification.

In order to calculate the wear of the metalic component the method described above was used.

The equations (20), (23) and (24) have for the materials studied particular forms obtained by introducing the numerical values of the interferring parameters, obtaining for a mean depth h_{mu} and an weared material volume V_u the following relations:

- polyamide Nylonplast AVE +30% glass/steel:

$$h_{mu} = l_m^2/8r_1 - 6.84\cdot10^{-5}N \qquad (mm) \tag{25}$$

$$S = 4.55\cdot10^{-5}\ Nl_m \qquad \left(mm^2\right) \tag{26}$$

$$V_u = 4.55 \cdot 10^{-4} N l_m \qquad \left(mm^3\right) \tag{27}$$

- polycarbonate Lexan +20% glass/steel:

$$h_{mu} = l_m^2 / 8r_1 - 6.38 \cdot 10^{-5} N \qquad (mm) \tag{28}$$

$$S = 4.25 \cdot 10^{-5}\ N l_m \qquad \left(mm^2\right) \tag{29}$$

$$V_u = 4.25 \cdot 10^{-4} N l_m \qquad \left(mm^3\right) \tag{30}$$

- polyamide Noryl +20% glass/steel:

$$h_{mu} = l_m^2 / 8r_1 - 11.96 \cdot 10^{-5} N \qquad (mm) \tag{31}$$

$$S = 7.97 . \cdot 10^{-5}\ N l_m \qquad \left(mm^2\right) \tag{32}$$

$$V_u = 7.97 \cdot 10^{-4} N l_m \qquad \left(mm^3\right) \tag{33}$$

Fig.3. Experimental device for tribological studies

The studies concerning the metallic semi-couple wear are generally based on the elastic contact hypothesis. For these plane half-couple the values for the equivalent elasticity modulus are:

A. polyamide Nylonplast AVE +30% glass; $E_{2A} = 17{,}1 \cdot 10^4\ daN/cm^2$.
B. polyamide Noryl +20% glass; $E_{2B} = 9{,}8 \cdot 10^4\ daN/cm^2$.
C. polycarbonate Lexan +20% glass; $E_{2C} = 18{,}4 \cdot 10^4\ daN/cm^2$.

Assuming that the plastic bush does not crush, we imposed the condition $p_{max} < 0.5H$, where H stands for the Brinell hardness. The required condition allows us to establish the following values of the maximum of the loads of the couple:

$p_{A1} = 163{,}0\ daN/cm^2$; $p_{A2} = 230{,}5\ daN/cm^2$; $p_{A3} = 282{,}3\ daN/cm^2$; $p_{A4} = 326{,}0\ daN/cm^2$;
$p_{A5} = 364{,}4\ daN/cm^2$; $p_{B1} = 123{,}4\ daN/cm^2$; $p_{B2} = 174{,}4\ daN/cm^2$; $p_{B3} = 213{,}5\ daN/cm^2$;

$$p_{B4} = 246{,}4\ daN/cm^2;\ p_{B5} = 275{,}7\ daN/cm^2;\ p_{C1} = 169{,}1\ daN/cm^2;\ p_{C2} = 239{,}1\ daN/cm^2:$$
$$p_{C3} = 292{,}9\ daN/cm^2;\ p_{C4} = 338{,}2\ daN/cm^2;\ p_{C5} = 378{,}1\ daN/cm^2.$$

The experimental tests were performed in large domains for variation of relative speed and normal loadings, or contact pressures. Couples with bushes made from thermoplastic material with linear contact on a steel surface (C120, Rp3, etc.) were used. In table 1 are presented for example the results of experimental tests made on two frictional couples, for one of the 8 different relative sliding speeds used.

Tab.1. The results of the experimental tests performed in order to determinate the wear rate of metalic component. Frictional couple: Polyamide Nylonplast AVE +30% glass / C120; $v = 18.56\,cm/s$

N (daN)	*t (h)*	Width of the wear imprint *l(mm)*				l_1^2 (mm^2)	l_2^2 (mm^2)	l_3^2 (mm^2)	Lm^2 (mm^2)	*S* (10^{-5} mm^2)	h_u (10^{-4} mm)	V_u (10^{-4} mm^3)	Wear rate	
		l_1	l_2	l_3	l_4								h_{mu} (10^{-4} mm/h)	$\overline{V_u}$ (10^{-6} cm^3/h)
1 1	1 1	0,208 0,307	0,304 0,304	0,307 0,318	0,300 0,310	0,083 0,095	0,092 0,092	0,094 0,101	0,090 0,096	1,365 1,410	0,9316 0,9982	1,365 1,410	0,9649	0,1387
2 2	1 1	0,472 0,478	0,489 0,489	0,484 0,491	0,482 0,486	0,223 0228	0,239 0,248	0,234 0,241	0,232 0,239	4,386 4,423	2,4409 2,5187	4,386 4,423	2,4798	0,4404
3 3	1 1	0,592 0,658	0,641 0,595	0,703 0,497	0,645 0,583	0,350 0,433	0,411 0,353	0,494 0,247	0,418 0,345	8,804 7,958	4,4392 3,6281	8,804 7,958	4,0336	0,8381
4 4	1 1	0,662 0,658	0,736 0,785	0,701 0,770	0,700 0,738	0,438 0,433	0,542 0,616	0,491 0,593	0,490 0,547	12,74 13,43	5,1708 5,8041	12,74 13,43	5,4874	1,3086
5 5	1 1	0,851 0,788	0,757 0,798	0,877 0,854	0,828 0,813	0,724 0,621	0,573 0,637	0,769 0,729	0,689 0,662	18,84 18,50	7,3135 7,0135	18,84 18,50	7,1635	1,8667

4. COMPARATIVE EVALUATION OF WEAR RATE

Table 1 presents the results of the tribological experimental tests, e.g. the mean values of the wear imprint depth $h_u\left(10^{-4}mm\right)$, and the mean values of the weared material volume $V_u\left(10^{-6}cm^3\right)$.

By dividing h_u and V_u to the duration of experimental test, we could obtain the values of the wear rate in the form of the depth $h_{mu}\left(10^{-4}\,mm/h\right)$ and of the volume $V_{mu}\left(10^{-6}\,cm^3/h\right)$.

Based on the methodology exposed above, the results are processed obtaining the variation curves of the wear with normal loading and relative speed, presented in fig. 4a and 4b, for two of the tested couples.

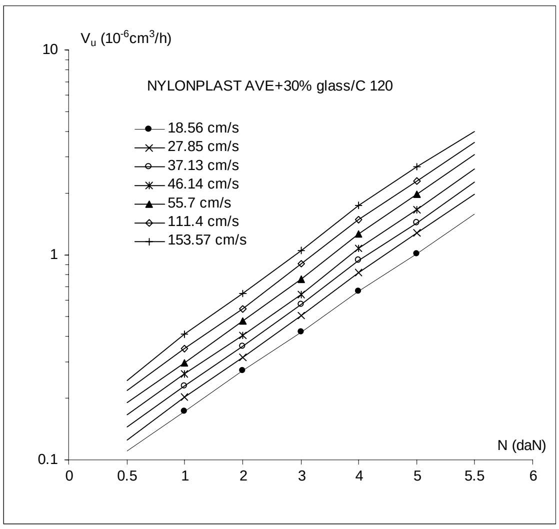

Fig. 4a

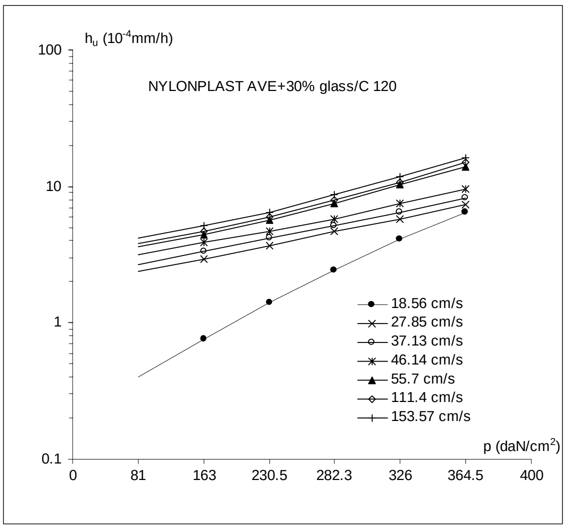

Fig. 4b

These curves could characterize only the frictional couple tested (one combination of materials). Furthemore the comparative evaluation of different couples could be made only qualitatively.

Thus, using relations (9) and (10) we could obtain the variation curves of the "comparative wear coefficients" (as volume and depth), $K\left(cm^3/cm\right)$ and $K^*\left(mm/cm\right)$.

Tab.2. The variation curve of comparative wear coefficient equations

Friction couple	Load (N)	The variation curve of comparative wear coefficient equations	
		K	K*
PA Nylonplast AVE +30% glass / C120	10	$y = 0{,}8030\,e^{-0{,}0110x}$	
	20	$y = 0{,}8739\,e^{-0{,}0090x}$	$y = 5{,}4312\,e^{-0{,}0153x}$
	30	$y = 1{,}1380\,e^{-0{,}0090x}$	$y = 6{,}4915\,e^{-0{,}0173x}$
	40	$y = 1{,}5870\,e^{-0{,}0090x}$	$y = 8{,}8046\,e^{-0{,}0220x}$
PA Nylonplast AVE +30% glass / Rp3	10	$y = 0{,}4240\,e^{-0{,}0190x}$	
	20	$y = 0{,}6640\,e^{-0{,}0130x}$	$y = 5{,}2346\,e^{-0{,}0253x}$
	30	$y = 1{,}0200\,e^{-0{,}0100x}$	$y = 8{,}4032\,e^{-0{,}0249x}$
	40	$y = 1{,}3950\,e^{-0{,}0090x}$	$y = 12{,}6080\,e^{-0{,}0253x}$
PA Noryl +20% glass / C120	10	$y = 1{,}5024\,e^{-0{,}0112x}$	$y = 3{,}8934\,e^{-0{,}0097x}$
PA Noryl +20% glass / Rp3	10	$y = 1{,}7070\,e^{-0{,}0120x}$	$y = 4{,}4259\,e^{-0{,}0098x}$
PC Lexan +20% glass / C120	10	$y = 0{,}4455\,e^{-0{,}0250x}$	$y = 6{,}3660\,e^{-0{,}0218x}$
	20	$y = 0{,}9988\,e^{-0{,}0247x}$	$y = 7{,}1108\,e^{-0{,}0230x}$
	30	$y = 1{,}4396\,e^{-0{,}0211x}$	$y = 6{,}8809\,e^{-0{,}0165x}$
	40	$y = 2{,}2425\,e^{-0{,}0244x}$	$y = 7{,}2365\,e^{-0{,}0144x}$
	50	$y = 3{,}0600\,e^{-0{,}0266x}$	$y = 7{,}0065\,e^{-0{,}0104x}$

These mastercurves are plotted in figures 5a and 5b for all tested couples and for different values of normal loading.

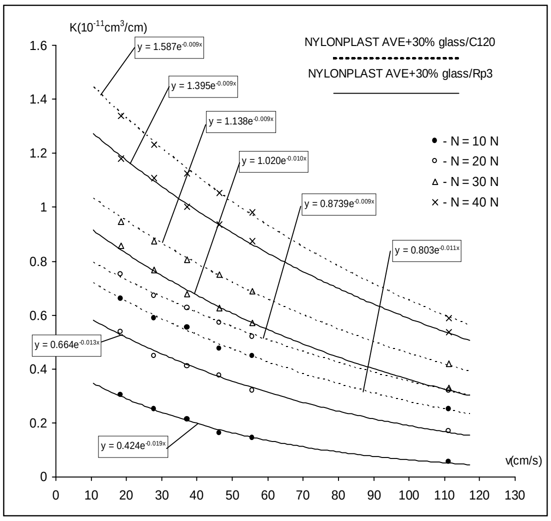

Fig.5a

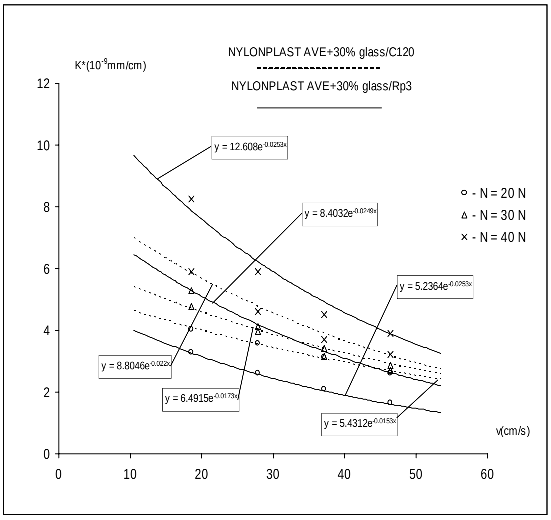

Fig.5b

5. CONCLUSIONS

The analysis of diagrams plotted in figures 2, 3 and 4 allows for establishing the variation equations for the comparative volumetric wear coefficient K and for the comparative depth wear coefficient K^*, for steel in linear contact with friction with glass reinforced thermoplastics.

In table 2 are listed the equations for the comparative wear coefficients (the volumetric and the depth ones). We could notice that the variation is not a linear one, these coefficients evolving expenonentially. We could notice also that the decrease of the K^* coefficient with the increase of relative speed is faster than the decrease of the K coefficient.

We consider that this effect is due to the fact that the thermoplastic material deforms under load which means that for Timken type couples the increase of the wear imprint width is more effective than that of the depth of the wear imprint. From the diagrams plotted here one could notice that the values of wear coefficients for the metalic component of the couple glass reinforced thermoplastic/steel are in the domain $\left(10^{-11} \div 10^{-12}\right) cm^3/cm$ and $10^{-9}\, mm/cm$ respectively. The comparative wearing coefficients and the mastercurves of these vs. relative speed have a special importance from the practical point of view. Based on these we could establish an optimal couple of materials from the design phase.

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