An analytical interpretation of the high temperature linear contact between composite materials reinforced with glass fibers and steel

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**Abstract:** In this paper we have tried to present the influence of the metal surface wear and of the contact temperature on the evolution of the sliding speed, of the normal load and of the friction coefficient. We have performed numerous experimental trials that have highlighted the dependency between load and wear in relation to the friction coefficient. A dry linear friction couple was used with a large range of loads and speeds, simulating real-life working conditions: temperature, sliding speed, contact pressure. We have made a connection between the theoretical case and the experimental results arising from the use of the "wear imprint method" for the volume and depth of wear.

**Keywords:** plastic material transfer, steel surface wear, plastics with glass fibers, linear contact, dry friction hardness of steel.

1. Introduction

Thermoplastic materials with glass fiber are characterized by high plasticity under high pressure and temperature, and they can be machined so that they return to their initial shape after being cast. These thermoplastic materials are biphasic, consisting of a polymer mass and reinforcement materials such as glass fibers, in order to obtain good mechanical characteristics as a result of the fact the stresses are transmitted through the glass fibers. At the same time the polymer does not allow a big deformation, limiting the significant deformation of the composite material. Also, it is good to use alkali-free glass fibers for a slow degradation in time and oxides can be added to maintain the high values of the elasticity module.

In specialist papers, values are given for the friction coefficient for different couples, plastic material/plastic material, plastic material/metal. The values are given for couples with lubricant or with dry
contact. The couples are formed from reinforced composite materials or composite materials without reinforcement.

Shen and Dumbleton [1] have studied comparatively the tribological behaviour of the polyoxymethylene and of the high density polyethylene (UHMWPE). They suggest a relationship for the calculation of the wear as follows:

\[ h = kpx \]

Where: \( h \) – linear wear; \( k \) - wear factor; \( p \) - nominal pressure; \( x \) - sliding distance.

Bartenev, Lavrentiev [2] establish that for the metal/plastic contact, the increase of the friction force is proportional to the sliding speed. This dependency is shown through the increase of the friction force together with the normal load. This was also demonstrated by Vinogradov

Jacobi [3] presents friction coefficient values between 0.04 and 0.5 for glass fiber reinforced polyamides. Barlow [4] shows that, for lubricated surfaces, an increase of the friction coefficient occurs with the increase of the relative sliding speed at the level of the friction surface.

Bilik [5] has shown that, for the couple polyamides/ steel, the friction coefficient is not constant. The coefficient depends on several factors: temperature, the ruggedness of the surface, the sliding speed and the contact pressure. Lancaster and Evans [6] have studied the tribological behaviour of glass fiber reinforced composite polymers, hydrodynamically lubricated, and notice the decrease of \( npd^2 / N \) for beak type couplings.

Clerico [7] has discovered that for the polyamide/ metal couple the value of the friction coefficient is higher for short periods of operation than for longer periods of operation, due to the viscoelastic behaviour of the composite material.

Lancaster [8] has studied different natural glass fiber reinforced polymers. He has shown that the friction coefficient decreases with the decrease in the ruggedness of the metal surface. Jost [9] shows that for polyamide/ metal couples the most encountered type of wear is adhesion wear.

Hrusciov and Babicev [10,11] have shown that, if we increase the polymer content for the plastic material reinforced with glass fiber/ steel couplings an increase of the microcutting component of the friction force occurs.

The Johnson-Kendall-Roberts (JKR) model [12] shows that for composite materials a strong abrasive component occurs upon contact, and several models of the contact have been drawn up together with the description of the adhesion contact.

Bely [13], Bartenev and Laventiev [14] have discovered that if the glass fibers inside the polymer matrix do not have the same orientation, the friction coefficient increases. Bowden and Tabor [15] have shown the importance of distributing forces on the Hertzian contact surface with an elliptical pressure distribution. In this situation, the central area will be the most affected area.

Watanabe et al [16] shows the influence of the friction coefficient depending on the temperature and the transfer of the plastic material unto steel.

Myshkin [17] mentions that, at the surface of the composite material, the links are viscous-elastic and that the abrasive friction component is higher that the adhesive friction component.

Therefore we can infer that the friction process for composite materials is complex, because numerous parameters influence the friction process, with or without lubricants. In specialist literature there is a heterogeneous approach, because different test installations, different experimental conditions and different materials are used.
2. Materials and methods

2.1 Analytical Method

The functioning under load of the Timken type friction couple (with linear contact) subjected to a load and in the presence of a relative sliding movement of the bushing made of glass fiber reinforced plastic on the surface of the steel sample highlights the occurrence of a wear imprint on the plane metal surface, the imprint being graphically represented in (Figure 1)

![Figure 1. The imprint scheme for Timken friction couples](image)

Knowing Coulomb’s law that the friction force $F_f$ is direct proportional to the normal force $N$:

$$F_f = \mu N$$  \hspace{1cm} (1)

More studies have shown that $\mu$, the friction coefficient is not only dependent on the normal load applied. Relations for variations of the friction force, depending on the load applied:

$$F_f = aN + bN^n$$  \hspace{1cm} (2)

$$F_f = a + bN$$  \hspace{1cm} (3)

$$F_f = a + bN^n$$  \hspace{1cm} (4)

$$F_f = kN^n$$  \hspace{1cm} (5)
Where \( n \) is subunitary.

We express the friction coefficient for the plastic materials:

\[
\mu = \frac{\tau_f}{p_c}
\]  

(6)

Where \( \tau_f \) represents the shear strength of the softer material

\( p_c \) represents the flow pressure of the same material.

Because \( p_c = \frac{HB}{3} \), we have:

\[
\mu = 3\frac{\tau_f}{HB}
\]  

(7)

The bush is rigid and accounting for the generally low non-uniformity of the imprint, considered as being formed by a series of cylindrical sectors having the length \( q \). The area of the lateral surface of the cylindrical sector is a circle segment:

\[
S_i = 0.5r^2\left(\pi\varphi_i^0/180^\circ - \sin\varphi_i\right)
\]  

(8)

Where \( S_i \) – lateral surface of the crystal sector;

\( \varphi_i \) - the angle;

\( r \) - the circle radius.

The radius \( r \) could not be identified with the cylindrical bush radius for the plastic / metal couples.

It is possible due to the elastic deformation of the bush under loading conditions and we illustrated this by the sketch in (Figure 2)

**Figure 2.** The elastic deformation of the cylindrical liner on the contact area for Timken friction couples (a – theoretical b - practical)

Using \( r_1 \) for the non-deformable liner radius and \( r_2 \) for the radius – in the contact area – of the deformed liner, we see in (Figure 2b):

\[
r_2 > r_1
\]  

(9)
Increasing the bush radius in the contact area leads to the decrease of the depth of the wear imprint from $h_1$ - which would appear if the elastic deformation of the bush would be neglected, to the value $h$ with the quantity $h_2$:

$$h = h_1 - h$$

(10)

Using $l$ for the width of the wear imprint, from $\Delta ABC$ it is:

$$(2r - h_1)h_1 = l^2/4$$

(11)

Because the value of the depth $h_1$ is very small, the term is negligible, we write:

$$h_1 = l^2/8r_1$$

(12)

Similarly, in $\Delta FGH$, we have:

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

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

$$h_2 = l^2/8r_2$$

(13)

Introducing (12) and (13) in (11) we have the following results:

$$h = l^2/8r_1 - l^2/8r_2 = l^2/8(r_2 - r_1)/r_1r_2$$

(14)

Where $r$ is the equivalent curvature radius:

$$1/r = 1/r_1 - 1/r_2 = (r_2 - r_1)/r_1r_2$$

(15)

From (14) we have:

$$(r_2 - r_1)/r_1r_2 = l^2/8h_2$$

(16)

Considering the frictional couple is loaded in the elastic domain with an elliptic distribution of stresses, the Hertz formula is:

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

(17)

Where:
- $\nu$ - 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:

$$1/E = 0.5[(1 - \nu_1^2)/E_1 + (1 - \nu_2^2)/E_2]$$

$$E = 2E_1E_2/0.91(E_1 + E_2)$$

(18)

(19)

From (17) we can express the width of the wear imprint:

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

(20)
Introducing in (20) the equivalent elasticity modulus and the equivalent radius expressions, the numerical value of Poisson ratio is:

\[ h_2 = \frac{0.527 N (E_1 + E_2)}{LE_1 E_2} \quad (21) \]

Considering (12) and (21), we have for the depth of the wear imprint:

\[ h = (l^2 / 8r_1) - \frac{0.527 N (E_1 + E_2)}{LE_1 E_2} \quad (22) \]

The wear imprint is the sum of cylindrical sectors, expanding in series the relation (21), neglecting the high-order terms and reducing the similar terms, the area of the lateral surface of a sector is:

\[ S_i = \frac{r^2 \phi}{12} \quad (23) \]

Replacing in the relation above the angle \( \phi \) with the ratio \( l/r \) and accounting for (15) and (16):

\[ S_i = l^3 (r_2 - r_1) / 12 r_1 r_2 = 2 l h_2 / 3 \quad (24) \]

Replacing the value of \( h_2 \) obtained (22) in (24), we obtain the expression for the area of lateral transversal surface of a cylindrical sector:

\[ S_i = 0.35 l (E_1 + E_2) N l / E_1 E_2 L \quad (25) \]

The volume of worn metal material will be:

\[ V_u = \sum_{i=1}^{n} (S_i q_i) = 0.35 l (E_1 + E_2) N l / E_1 E_2 \quad (26) \]

Where \( l_m \) is the mean width of the wear imprint.

With this value we can obtain the volume of worn metal material \( V_u \) and the mean value of the depth of removed layer \( h_{nu} \).

We have some correlation functions between wear scar volume \( V_u \) and friction coefficient \( \mu \). The wear scar volume and the wear scar depth are according to the normal load \( N \) in polynomial forms:

\[ T = \sum_{i=0}^{2} a_i N^i, \]

\[ V_u = \sum_{i=0}^{2} b_i N^i, \quad (27) \]

\[ h_{nu} = \sum_{i=0}^{2} c_i N^i, \]

And functions between friction coefficient \( \mu \) and normal load \( N \) are:

\[ \mu = A \ln N + B \quad (28) \]

Where \( a_i, b_i, c_i \) and \( A, B \) are determined from regression functions.

From relations (13) we obtain
\[ V'_c = \frac{b_5}{a_2} T + \left( h_1 - \frac{b_5}{a_2} a_1 \right) N + c_0 - \frac{b_5}{a_2} a_0, \quad (29) \]

or, after substituting \( N \) from (28) and replacing in (29) we have

\[ V'_c = \frac{b_6}{a_2} T + \left( h_1 - \frac{b_6}{a_2} a_1 \right) \exp[(\mu - B)/A] + c_0 - \frac{b_6}{a_2} a_0. \quad (30) \]

Similarly, the wear scar depth is

\[ h_2 = \frac{c_1}{a_1} T + \left( c_0 - \frac{c_1}{a_1} a_1 \right) N + c_0 - \frac{c_1}{a_1} a_0, \quad (31) \]

\[ h_3 = \frac{c_2}{a_1} T + \left( c_0 - \frac{c_2}{a_1} a_1 \right) \exp[(\mu - B)/A] + c_0 - \frac{c_2}{a_1} a_0. \quad (32) \]

Also, contact temperature as a function of \( \mu \) such:

\[ T = \sum_{i=0}^{2} a_i \exp[(\mu - B)/A] \quad (33) \]

Considering that wear is adhesive, in case of linear polymer/steel contact, we consider the volume of wear material according to J.F. Archard’s relationship (18) [18]. Taking into account (8) and (9):

\[ V'_a = kNvt = \frac{b_5}{a_2} T + \left( h_1 - \frac{b_5}{a_2} a_1 \right) N + c_0 - \frac{b_5}{a_2} a_0 \quad (34) \]

and from (18) and (26):

\[ V'_a = kNvt = \frac{b_6}{a_2} T + \left( h_1 - \frac{b_6}{a_2} a_1 \right) \exp[(\mu - B)/A] + c_0 - \frac{b_6}{a_2} a_0. \quad (35) \]

Writing the Archard relationship for the wear depth (20), from (25) and (27):

\[ h_2 = k' \rho \rho V_t = \frac{c_1}{a_2} T + \left( c_0 - \frac{c_1}{a_2} a_1 \right) N + c_0 - \frac{c_1}{a_2} a_0 \quad (36) \]

From (20) and (21):

\[ h_3 = k' \rho \rho V_t = \frac{c_2}{a_2} T + \left( c_0 - \frac{c_2}{a_2} a_1 \right) \exp[(\mu - B)/A] + c_0 - \frac{c_2}{a_2} a_0. \quad (37) \]

### 3. Results and Discussion

The relationships (34), (35) and (36), (37) show the connection between the output parameters \( V'_a, T, \mu \) and the input parameters of the wear process \( N, v, t \), respectively between \( h_2, T, \mu \) and \( p, v, t \). These relationships show the complexity of the wear process for metal surfaces during the linear dry friction.
contact with short glass fiber reinforced polymers [19]. By changing an input parameter, all the other parameters will be modified. This modification changes all output parameters of the system due to the numerous factors which contribute to the phenomenon themselves.

In conclusion, the phenomena that occur have a high level of complexity and we can illustrate this by means of a drawing (Fig. 3) which helps us see the complexity of the phenomenon.

![Figure 3](image.png)

**Figure 3.** The complexity of the evolution of the friction – wear process for the linear dry contact polymer + SGF/steel.

4. Conclusions

The studies carried out concerning the friction and wear of metal surfaces belonging to friction couples, in linear contact with short glass fiber reinforced thermoplastic material / steel, were aimed at highlighting the influence of the input parameters of the tribosystem (normal load – contact pressure, relative sliding speed, type of friction, the characteristics of the materials coming into contact), on the output parameters of the tribosystem (the volume of the worn metal material, the depth of the worn material, the contact temperature and the friction coefficient).

According to tribological knowledge published at international level up to this point, intuitively, there is a clear influence of the friction coefficient on the wear of the metal surfaces, however no theoretical connection between the two values has been achieved.

By presenting and comparing the variation functions of the friction coefficient, of the contact temperature, of the volume of worn metal material and of the depth of wear depending on normal load, we have sought and we considered that we have succeeded to show that between the input parameters of the tribosystem with linear contact (normal load, relative sliding speed, physical and mechanical characteristics of the materials coming into contact, the state of the surface) and the output parameters (friction coefficient, contact temperature and wear) there is an interdependency which changes constantly during the friction process.

Presenting the evolution of the tribosystem with dry linear contact, SGF-reinforced polymer / steel is very useful for tribological research. This actually shows the closed loop between the input and output parameters, taking into consideration the modification of the initial parameters (and especially the modification of the state of the metal surface after wear).
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