DETERMINING THE HEAT REGIME IN THE WORKING OF A COUPLING WITH SLIDING MOTION

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Abstract: *Following the sliding motion between two surfaces a heat release occures which leads to a changing of the mechanical properties of the surfaces in contact. Amongst the factors that influence the heat release one may name the loading, the velocity, the surface topography, the surface material, the lubrication and last but not the least, the environment.*

The present paper proposes a computation model to determine the temperature in the working of a coupling to which the mechanical characteristics of the materials are not affected by changes which, at their turn, lead to undesired effects.

Key words: *friction coupling, temperature, heat flux, contact surface, working parameters*

1. INTRODUCTION

The scientific literature abundes of opinions on the thermic aspects that occur during the working of a friction .coupling. The temperature distribution in a coupling depends on the interconditioning of the working parameters (loading, speed etc), on the surfaces topography, on the properties of the materials in contact, on the lubrication etc.

It is considered that for small loadings, the contact between surfaces occurs on a finite number of asperities which define the nominal area or the contact area and the heat transfer is solid. For big loadings, the real contact area is closed to the nominal one and the heat transfer in asperities is a unidimensional one. When a lubricant film is inserted between the surfaces, the heat released following the sliding motion (friction) also get through the film, the lubricant being essential in diminishing the thermic effects.

In order to explain the distribution of the temperature and heat resulted by the friction of a sliding coupling, the most unfavourable case is considered, when between surfaces there is no lubricant film to help in fast release of heat during working.

2. WEAR MODEL BY MICRO-JONCTIONS MELTING AND SHARING

Following the friction process, the temperature resulted by the friction of the asperities in contact is much higher than the medium temperature of the coupling surface [1,2,3,4]. It is considred that in the contact points of the surface asperities the temperature is so high that it generates the melting of the material.

The temperature value generated during the coupling working is connected with the loading parameters – contact pressure, sliding speed. For high pressure and speed (10-1000m/s) a fluid metal film that leads to a “lubrication by melting” results [2,3,4,5]. Though the friction coefficient dicreases because of this melting layer, the wear increases because the high temperature continues to melt more and more solid material.

For a complex analysis of the friction coupling wear process, wear maps may be defined to illustrate the interconditioning way of the working parameters so that to obtain a maximum durability or an imposed working precision for the friction coupling.

Awear model by micro-jonctions melting and shearing is proposed by Lim and Ashby [6,7,8,9]. They got to the conclusion that the wear by melting takes places when the power dissipated by friction on the surface unit exceeds a certain value considered critical.

For a geometry of the pin-disk type surfaces with a circular movement, it is considered that the heat release occurs in the sliding area and its dissipation occurs only through the friction coupling elements.

It is estimated that a part (α) of the heat released (q) is dissipated in the pin and another part ($1-\alpha$) is dissipated in the disk. Evaluation of the heat partition coefficient (α) is done by considering that the medium temperature on the mobile surface is equal to the temperature on the fixed surface [10,11,12].

This coefficient depends not only on the thermic diffusion of the material and the piece size but also on the rate of travel of the coupling elements. This speed is determined by the help of the Peclet invariant [11,12,13,14] which is for the concret case of the conctact (pin-disk):

$$Pe = \frac{vL}{4a_2}$$

where: a_2 - thermic diffusion of the pin material;

L - contact surface of the pin

Function of the Pe invariant, the partition coefficient is:

$$\alpha = \frac{\lambda_1}{\lambda_1 + \lambda_2} \text{ for } Pe \le 0{,}1$$

$$\alpha = \frac{1}{1 + 0{,}795 \frac{\lambda_1}{\lambda_2}\left(\frac{a_1}{a_2}\right)^{1/2} Pe^{-1/2}} \text{ for } Pe \ge 5$$

where: λ_1, λ_2 – thermic conductivities of the disk material, respectively of the pin

a_1, a_2 – thermic diffusion of the disk/pin material

For values of the Peclet invariant between $0{,}1 < Pe < 5$, a linear interpolation is recommended.

Considering, according to the mechanical molecular theory, that the friction is achieved at the asperities contact level, on the real contact area, the friction coefficient is:

$$\mu = \frac{\bar{\tau}}{\bar{p}} + \mu_m$$

where: $\bar{\tau}$ - specific molecular resistance of material;

$\bar{p}$ - loading parameter;

μ_m - mechanical part of the friction coefficient

For the mechanical part of the friction coefficient (μ_m), the dependence of the Pe invariant is accepted as [8]:

$$\mu_m = \frac{2}{\pi} tg\theta \left(\frac{Pe \cdot 4 \cdot a_1}{L}\right)^a$$

where: θ - attack angle of the pin asperities;

a - a constant that depends on the material type.

After getting the thermic equillibrium in the elements of the two coupling, liniarizing the first law of the heat flow, the medium temperature of the contact surfaces may be established:

$$T_m = T_o + \alpha\mu F v \frac{l_m}{A_n \lambda_2}$$

where: T_o - temperature of the clamping system (about 20^o);

μ - friction coefficient between the coupling materials;

F - coupling loading;

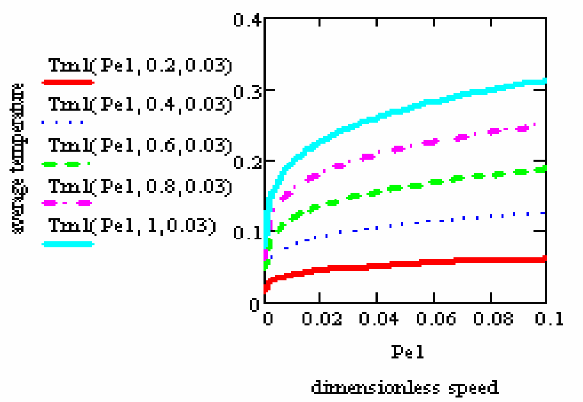
v - displacement speed;

l_m - distance of diffusion of the heat to the pin clamping system;

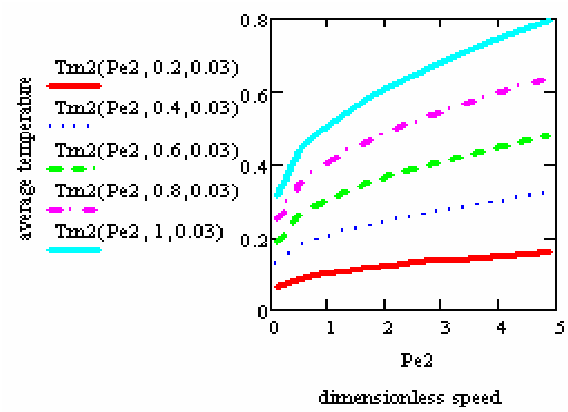
A_n - nominal contact area;

λ_2 - thermic conductivity of the pin material

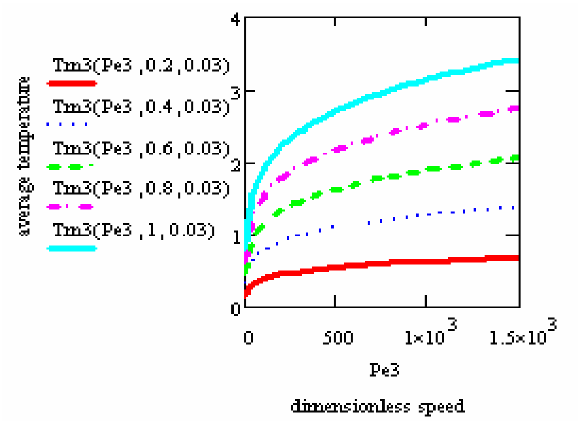
For a pin-disk type friction coupling whose material and geometry features are known, one may establish the evolution of the medium temperature for different loadings ($\bar{p}$), function on the Peclet dimensionless speed, for the three intervals ($Pe1, Pe2, Pe3$) (figure 1).



$Pe < 0{,}1$



$0{,}1 < Pe < 5$



$Pe > 5$

Figure.1 Variation of medium temperature with loading and Peclet variant

It is noted that the higher loading and speed of the coupling, the higher medium temperature on the contact surfaces.

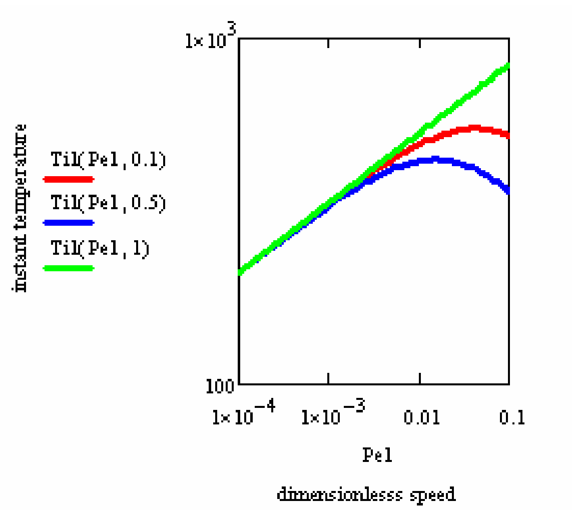
With the view to evaluate the instant temperature, one says that the contact area increses with the external load as a result of the increase of the micro-areas. For small loadings, the total number of micro-areas in contact tends to one, and for large loadings the micro-areas get connected so that the nominal area tends to be equal to the real area ($A_n = A_r$). The solutions accepted by Tian and Kennedy [6,10,11,12] at the level of the surface micro-contacts appraise the dimensionless instant temperature as:

$$T_i = 2{,}72\frac{r_a}{L}\frac{A_n}{A_r}\frac{\left(\bar{\tau}+\mu_m\right)Pe}{\sqrt{0{,}6575+\left(2\frac{r_a}{L}\right)Pe\left[1+\frac{A_n}{A_r}\bar{p}(1-\bar{p})\right]}}$$

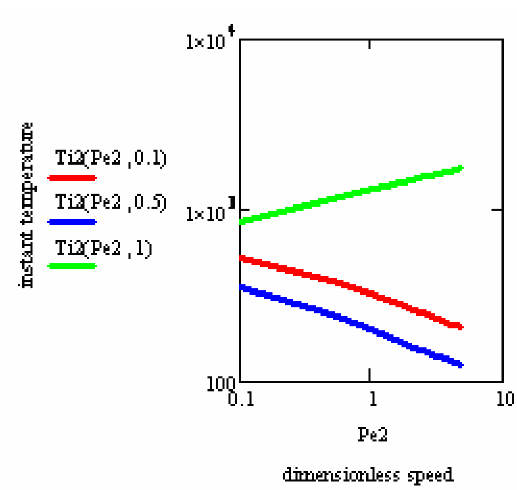
where r_a is the micro-contact radium.

Figure 2 presents the theoretical evolution of this instant temperature with the speed parameter Pe, for a pin-disk type friction coupling whose dimensional and material features are known. In case of small and very small speeds, for a large loading, the increase of temperatures at the asperities peak is high. For small loadings, the increase of instant temperature is much slowed.

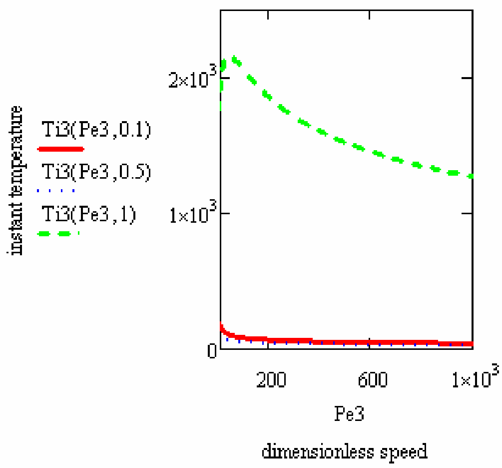
If the instant temperature is limited by factors connected to the wear intensity by friction, a map of the dimensionless loading function of the dimensionless speed for different instant temperatures may be determined (figure 3).



$Pe < 0{,}1$



$0{,}1 < Pe < 5$



$Pe > 5$

Figure 2. Instant temperature evolution with speed parameter

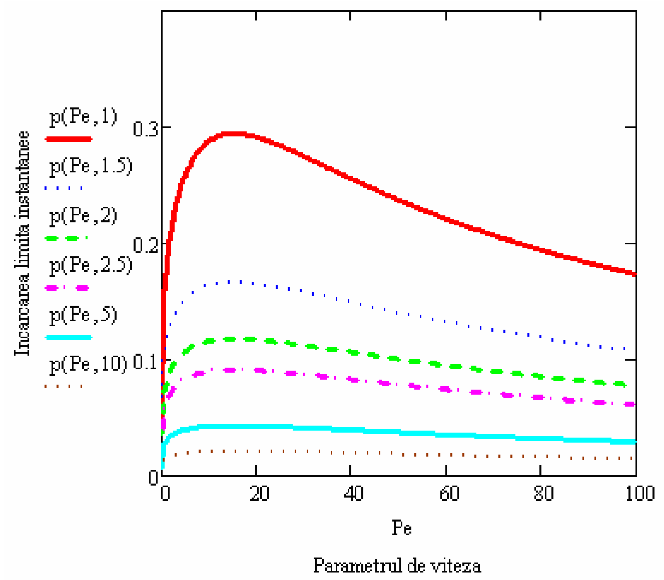
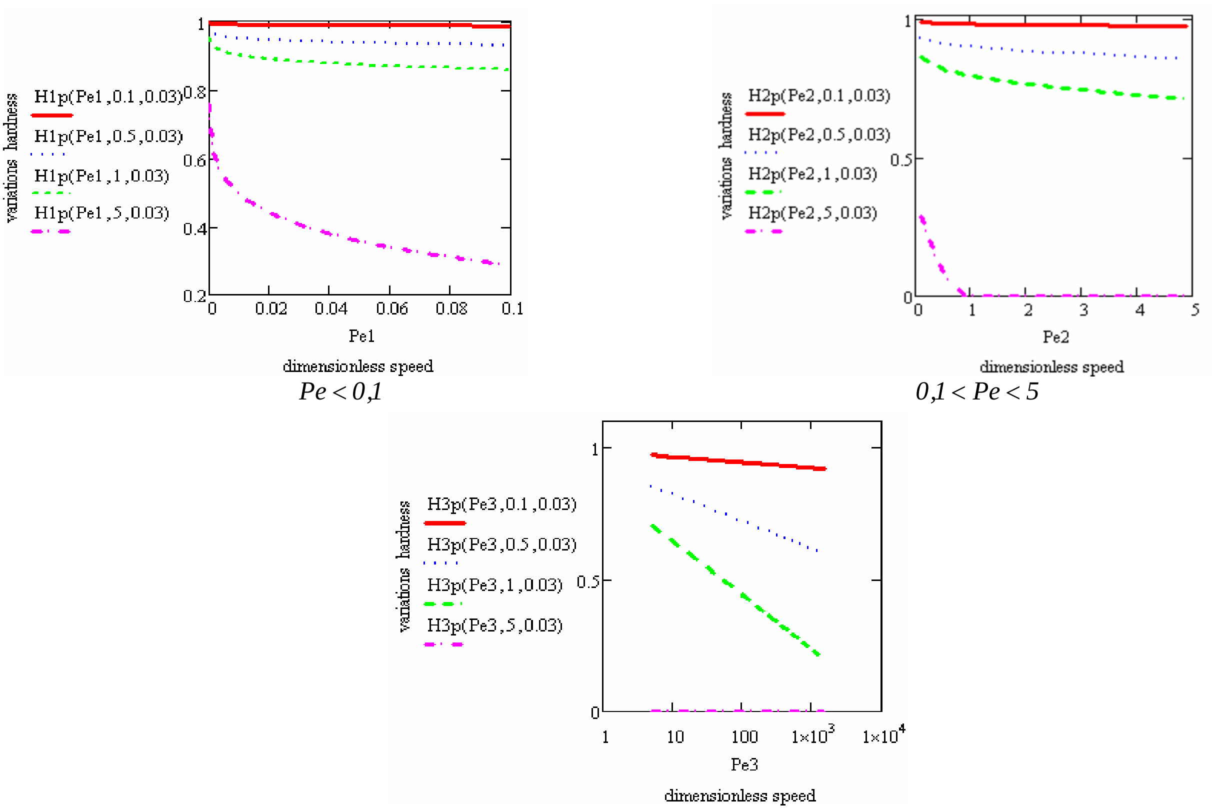


Figure 3. Loading evolution for different tempetarures

The increase of temperature in the contact ares of the asperities also influences the hardness of the surfaces in contact. Thus, at the asperities level, a change in their micro-hardness occurs. This results in local plastic deformations occurance which, at their turn, negatively influence the good working of the coupling. If this law is applied at the micro-contacts level, for an instant temperature, the map of micro-hardness variation may be obtained (figure 4).



Pe < 0,1 *0,1 < Pe < 5*

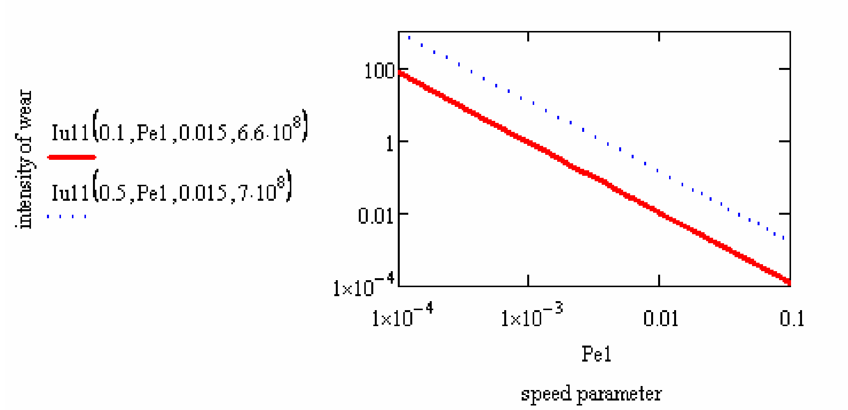
Pe > 5

Figure 4. Hardness variation for instant temperature

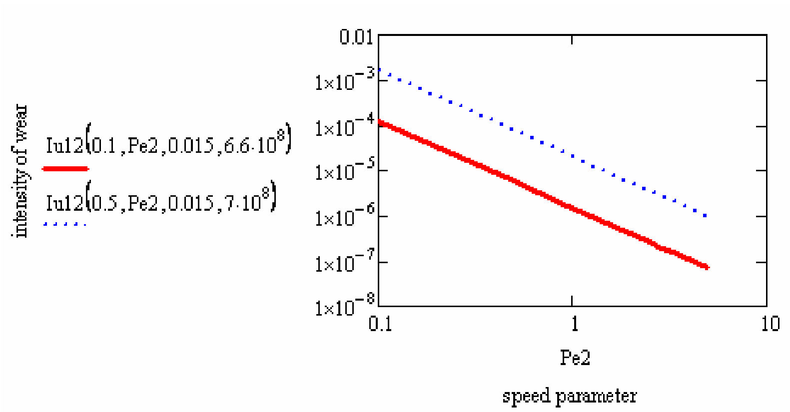
All these changes combined with the fatigue by friction are conclusive in wear particles occurrence [6, E, 7]. For the practical example, at the micro-contacts level, the variation of the flow voltage with the local contact temperature is considered [3,8,13,14]. So, the linear intensity of wear for each element may be determined, this one being an indicator of wear process if the following are known: the length of friction for a single cycle of each element of the coupling (L_{01}), the number of friction cycles (N_c) and the thickness of the worn layer (Δh_{12}):

$$I_{h1,2} = \frac{\Delta h_{1,2}}{N_c \cdot L_{01,2}}$$

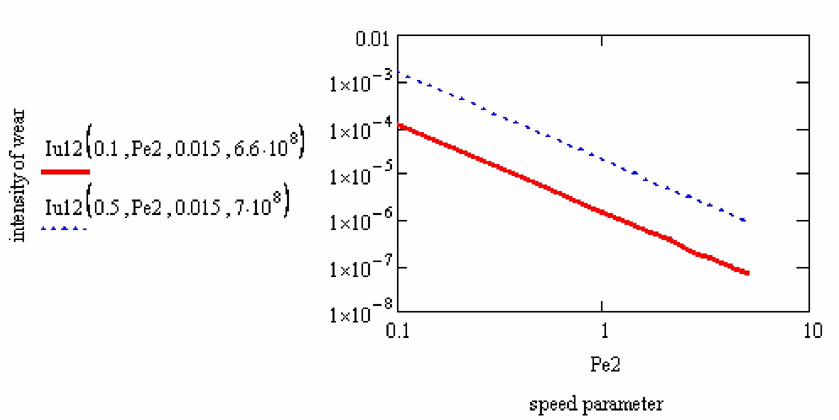
Figure 4 presents the evolution of the wear intensity for a pin-disk type friction coupling whose dimensional and material features are known, according to the mathematical model proposed. One may note the dicrease of the wear intensity with the loading parameters and temperature.



Pe < 0,1



0,1 < Pe < 5



Pe > 5

Figure 5. Evolution of the wear intensity

If the contact surface remains constant during the wear, the wear coefficient may be determined:

$$k = \frac{\Delta V_{12}}{F \cdot L_{f1,2}} = \frac{A_n \cdot \Delta h_{12}}{F \cdot L_{f1,2}} = \frac{A_n \cdot \Delta h_{12}}{p_n \cdot L_{f1,2}} = \frac{I_{h12}}{\overline{p}}$$

where: ΔV - worn material volume;
F - normal force transmitted;
$L_{f1,2}$ - friction length of the couplings;
p_n - normal pressure

3. CONCLUSIONS

In designing any friction couplings one may establish from the beginning the optimal working conditions by interpreting the experimental analytical results with the help of the wear maps. These ones provide information with regard to the velocity parameter and the exploiting conditions. It may be considered that the working of the coupling is optimal if the velocity parameter and the exploiting conditions are situated under these high curves. These wear maps are considered working tools for tribologycal applications.

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