THE INFLUENCE OF DIFFERENT PARAMETERS ON COUPLINGS WORKING FOR SMALL VELOCITIES SLIDING MOTION

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Abstract: *The working of friction couplings in case of small sliding velocities and in case of a mixed or limit lubrication comes along with an unevenness of motion characterized by abruptness or by fits and starts. This phenomenon is known as stick-slip. The occurrence of this phenomenon in numerous technical applications has always aroused an increased scientific interest because of its unwanted implications on the machining precision in machine-tools and not only. The present paper proposes an analysis on the influence of different parameters that may come between the couplings working where stick-slip occurs.*

Keywords*: sliding motion, stick-slip, friction force, static friction coefficient, kinetic friction coefficient, lubrication, loading, temperature*

1. INTRODUCTION

In case of small sliding velocities (0,18…180 mm/min), in case of an insufficient lubrication, the movement of a friction coupling may come along with abruptness or fits and starts. This phenomenon is frequent in machine-tools, especially in lathes, hydraulic cylinders or slipper clutches and not only.

From phenomenological point of view, this type of movement named *stick-slip* implies a period of sliding with fits and starts and another one with auto vibrations [1]. The self-vibrator phenomenon depends on the sliding velocity, system rigidity and lubricant viscosity.

The most accepted situation for stick-slip is that the static friction exceeds the kinetic one. Out of this, the idea of two types of coefficients – a static one (corresponding to the sticking period) and a kinetic one (corresponding to the sliding period).

The pair of friction coefficient values, static and kinetic, noticed by Coulomb and highlighted by Thomas in 1930, arose a pro and against debate. It is generally accepted that the size and evolution of friction coefficient in case of stick-slip movement depends on many factors such as: material, coupling surface processing, lubrication, loading (pressure, velocity).

Starting from the research presented in the literature, the present paper proposes an analysis upon the influence of different factors on the working of the couplings where *stick-slip* occurs. The computations are applied for a coupling made of cast iron/relamid, materials used the construction of the lathe slide bars. Some experimental data were obtained using the stick-slip stand from the laboratory of Machine Components – Polytechnic University of Bucharest were these materials were analyzed. The mobile proof sample is made of cast iron and the fixed one is made of relamid (composite material). The sliding time was measured for different loads such as: $F_{n1} = 0{,}5N$, where $t_1 = 5s$, $F_{n2} = 1N$ where $t_2 = 9s$, $F_{n3} = 2N$ where $t_3 = 11s$, $F_{n4} = 5N$ where $t_4 = 14s$. The rigidity of the elastic system segment was $k = 1{,}24 \cdot 10^7\ N/m$ and may be changed to $k = 8 \cdot 10^8\ N/m$.

2. FACTORS THAT INFLUENCE THE STICK-SLIP MOTION

a. Lubrication influence

In case of dry friction, the motion equation for the moving body is:

$$m\ddot{x} + kx = \mu_{ak} F_n \tag{1}$$

where:
m - sliding device mass;
k – system rigidity;
x - linear position of the mobile sample;
$\ddot{x}$ - linear acceleration of the mobile element ($\ddot{x} = d^2x/dt^2$)
μ_{ak} - kinetic friction coefficient whose relation is experimentally deduced [2]

$$\mu_{ak} = \frac{\tau_o}{p} + \frac{2}{\pi} \tan\theta \cdot v^a \tag{2}$$

τ_o - critical share stress of the mobile proof sample material;
p - normal pressure;
a - constant parameters dependent to the material;
θ - attack angle of the mobile proof sample $\theta = 6^o$;
v - mobile proof sample displacement velocity.

The evolution of the kinetic friction coefficient computed function on the loading conditions is presented in Fig.1

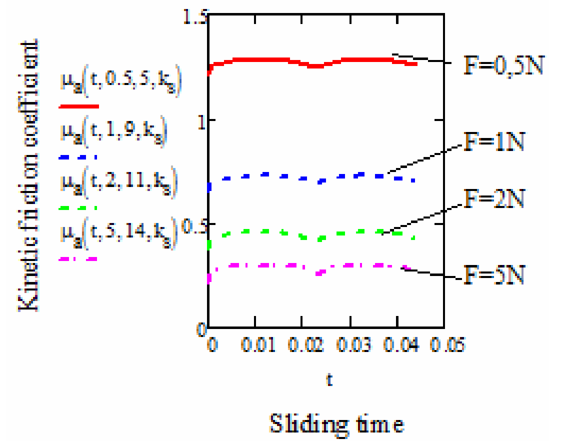

Fig. 1. The evolution of the kinetic friction coefficient function on the loading conditions and sliding time

For the limit conditions $t=0$ the displacement and linear velocity are:

$$x=\frac{\mu_{as}\cdot F_n}{k} \tag{3}$$

$$\dot{x}=\frac{dx}{dt}=v \tag{4}$$

where: μ_{as} - static friction coefficient;

The adherence time for static friction was the period between the moment when the normal load was applied and the moment when the sliding started. This can be expressed by the relation:

$$\mu_{as}=k_s\cdot p_a^b\cdot t_s^c \tag{5}$$

where: k_s, b, c - constant parameters dependent to the material;

t_s - parking time;

p_a - contact relative pressure

$$p_a=p/H_o=(F_n/\pi\cdot r^2)/H_o \tag{6}$$

p - normal pressure;

r - contact radius of the mobile proof sample material;

H_o - hardness of harder material.

where:

$$A=\sqrt{A_o^2+B_o^2}$$

$$\varphi=ctg(-B_o/A_o)$$

$$\omega_o^2=k/m$$

$$A_o=(\mu_{as}-\mu_{ak})F_n/v$$

$$B_o=1/\omega_o$$

Figure 2 presents the evolution of the static coefficient function on loading.

So, the relative velocity during sliding is:

$$v_1=\frac{x}{v}=-\omega_o\cdot A\cdot sin(\omega_o t+\varphi)t \tag{7}$$

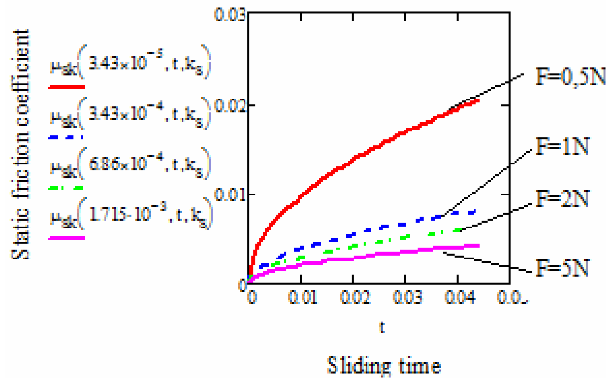

Fig. 2. Evolution of the static coefficient function on loading and parking time

Figure 3 presents the evolution of the relative velocity modulus during sliding in case of dry friction (the coupling surfaces were degreased), for different loadings.

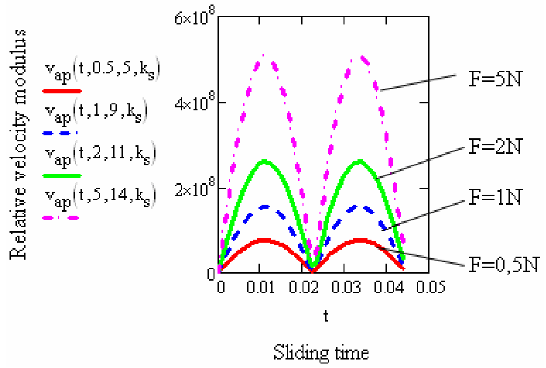

Fig. 3. Relative velocity modulus during sliding in case of dry friction

It is known that the friction size of a coupling is influenced by lubrication.

This is highlighted when using a Newtonian lubricant with dynamic viscosity $\eta=0{,}1Pa\,s/m^2$ with a thickness $h_u\cong 3\cdot 10^{-16}\,m$.

The motion equation when the friction coupling is lubricated, the friction is mixed and it is:

$$m\ddot{x}+\gamma\dot{x}+kx=\mu_{ak}F_n \tag{8}$$

where: γ damping parameter of the lubricant $\gamma=\eta\pi d^2/4h_u\cong 1{,}2Nms$

The relative velocity during sliding will be:

$$v_1=-\frac{A}{v}e^{-\gamma t}\cdot\omega\cdot cos(\omega t+\varphi-\alpha) \tag{9}$$

where: $A = \sqrt{A_o^2 + \left(\frac{\delta_o}{\omega} A_o + B_o\right)^2}$

$B = v / \omega$

$\omega = \sqrt{\omega_o^2 - \delta_o^2}$

δ_o - specific damping ($\delta_o = 1{,}2$)

$$\varphi = ctg\left[\left(\frac{\delta_o^2}{\omega_o} A_o + B\right) / A_o\right]$$

$$\alpha = ctg\frac{\omega}{\delta_o}$$

Figure 4 presents by comparison the evolution of the relative velocity modulus during sliding in case of dry and mixed friction for different loadings.

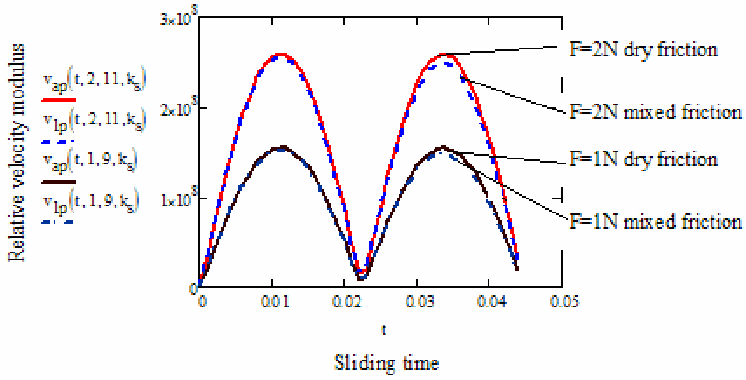

Fig. 4. Evolution of the relative velocity modulus during sliding in case of dry and mixed friction

b. Temperature during sliding influence

It is a well-known fact that when two bodies slide, the most part of the mechanical work turns into heat. This generates the increase of the temperature of the surfaces in contact [3]. The increase of temperature is unfavorable when stick-slip occurs.

In order to analyze the temperature influence during a coupling sliding when stick-slip occurs, the theories of Tian and Kennedy [4] and Lim and Asby [5] for a heat source are considered. According to these theories the medium temperature on the contact surface is:

$$T_m = \beta \cdot r \cdot \frac{q_1}{\lambda_1} \tag{10}$$

where: β characterizes the proof samples dimensions ($\beta = l / r$);

l fixed proof sample thickness;

r mobile proof sample radius;

q_1 heat flow entering the mobile proof sample ($q_1 = \alpha q$);

α heat partition factor;

q total heat flow;

For the heat partition factor it is accepted the idea that this factor depends on the sliding time, Peclet number [6,7]. The Peclet number (Pe) depends on the material diffusivity (a_1), speed (v_1) and moving source size (r):

$$Pe = \frac{v_1 \cdot r}{2 \cdot a_1} \tag{11}$$

The theoretical variation of this parameter during stick-slip sliding is presented in Figure 5.

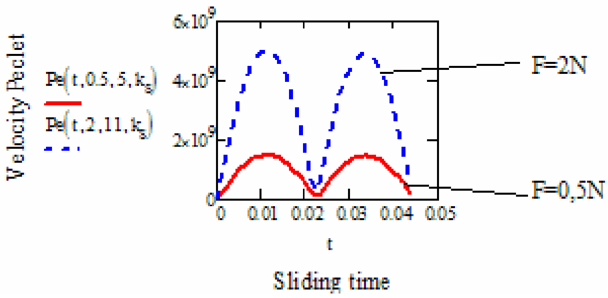

Figure 5. Dimensionless velocity variation during stick-slip sliding

The partition factor of heat depends on the Peclet number and is

$$\alpha = \begin{cases} \dfrac{\lambda_1}{\lambda_1 + \lambda_2} & Pe \le 0{,}1 \\[2ex] \dfrac{1}{1 + 0{,}795 \cdot \dfrac{\lambda_2}{\lambda_1} \cdot \left(\dfrac{a_1}{a_2}\right)^{1/2} \cdot Pe^{-1/2}} & Pe \ge 5 \end{cases} \tag{12}$$

For values between $0{,}1 < Pe < 5$ the partition factor of heat is:

$\alpha = 1{,}02 \cdot \alpha_{01} - 0{,}02 \cdot \alpha_5 + 0{,}204(\alpha_5 - \alpha_{01})Pe$

where:

α_{01} and α_5 are computed for $Pe = 0{,}1$ şi $Pe = 5$

λ_1, λ_2 heat conductivity of proof samples

a_1, a_2 diffusivity of proof sample materials.

If considered that the temperature increases within the entire volume of the surfaces in contact and, considering all those mentioned at partition factor and Peclet number, the dimensionless volumetric temperature relation may be obtained:

$$T_{ma} = 2\alpha\beta \cdot p \cdot \mu_{ak} \cdot Pe \qquad (13)$$

Figure 6 presents the evolution of volumetric temperature during sliding depending on the Peclet number.
It is shown that during sliding the temperature changes

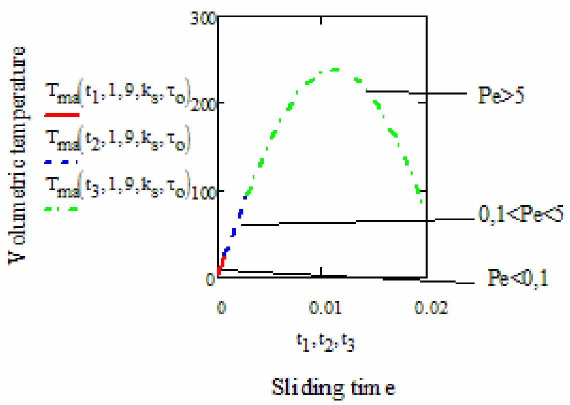

Fig. 6. Evolution of dimensionless volumetric temperature during sliding

c. Loading influence

The Bowden-Tabor [7] adhesion model explains the friction force as the necessary force to shear that occurs at the level of the contact asperities. This model was later changed by Tabor [8] who took into consideration both the shear and normal tensions. The adhesion of the friction couplings contact will occur when the real contact area (A_r) between the two couplings will be equal to the nominal contact area (A_n).

$$\left(\frac{F_n}{A_r}\right)^2 + \alpha_t\left(\mu_{ak}\frac{F_n}{A_r}\right)^2 = H^2 \qquad (14)$$

where: $\alpha_t \cong 12$ is a constant;
H - material hardness.
When adhesion is complete, the contact critical pressure may be set as:

$$p_{ac} = \frac{1}{(1+\alpha_t \cdot \mu_{ak}^2)^{1/2}} \cdot \frac{H}{H_o} \qquad (15)$$

where: H_o material hardness at environmental temperature.
This contact critical pressure may be considered as a maximum loading capacity of the coupling. Figure 7 presents the variation of the maximum loading capacity based on the theoretical hypothesis for a normal force $F_n = 0{,}5N$ and a normal force $F_n = 5N$. One may notice a decline of the coupling loading capacity during stick-slip and, at the same time, an encoding when the loading increases.

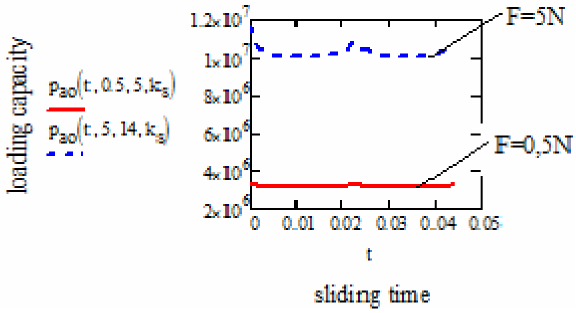

Fig. 7. Variation of maximum loading capacity

3. CONCLUSIONS

From the above mentioned, one may notice that in case of jerky motion there is not a singular factor to change the phenomenon but a series of factors that influence it differently. By extending the analysis for some other material couplings, lubrication and type of lubricant, as well as by changing the system rigidity, the followings were noticed:
- the coupling lubrication – the more abundant it is, the smaller is the friction within the coupling and the stick-slip phenomenon is absent, however, the fluid viscosity reduction leads to the phenomenon increase;
- the higher the roughness of surfaces in contact the more intense phenomenon so, a smoother processing is needed;
- the coupling loading is also important because its increase leads to an increase of friction coefficient and subsequently, the stick-slip phenomenon intensity increases;
- the system rigidity may damp the stick-slip;
- the coupling material chosen proved to have an important influence and the plastic-plastic ones have been frequently chose.

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