CALCULATION PRINCIPLES REGARDING THE WEAR OF SHAFT-SLIDING BEARING COUPLING

PETRE IVONA, TACHE CARMEN

VALAHIA University of Târgovişte

Abstract: The present paper deals with the calculation of wear for a shaft-sliding bearing coupling, trying to point out the factors that influence the wear of the coupling.

Keywords: wear, temperature, contact pressure, kinematics coupling

1. INTRODUCTION

As a consequence of the influence of different random factors that operate during the running of a kinematics coupling (specific pressures, sliding speed, surface micro geometry, temperature, etc.), the wear of the part is uneven. This wear of the kinematics coupling influences directly the running of the assembly that has it among his components. The analysis of the potential distribution of the wear on coupling's contact surfaces and the pointing out of the factors which determine it constitute the premises for the calculation and the evaluation of the coupling's wear.

Wear calculation for kinematics couplings can be dealt with taking into consideration two aspects:

- the calculation according to the pressures on the friction surfaces, where there are compared the values of the pressures (medium and maximum) that operate on the friction surface and the accepted values given in the technical documentation;

- the calculation of wear size and the determination of the shape of the worn surface, which allows input information about the wear of surfaces in each point, the distribution of pressures on the friction surfaces and the variation of reciprocal position of coupling's surfaces as a consequence of the wear.

We shall not insist on the first aspect of the calculation, due to the fact that we consider that the second one is of much more interest.

2. WEAR CALCULATION FOR SHAFT-SLIDING BEARING COUPLING

The wear of the shaft-sliding bearing kinematics coupling (Fig. 1) can be studied under the conditions of dry friction or limit lubrication.

a. The shaft is not worn out
b. The wear of the shaft is higher than the one of the bearing
c. The wear of the bearing is higher than the one of the shaft

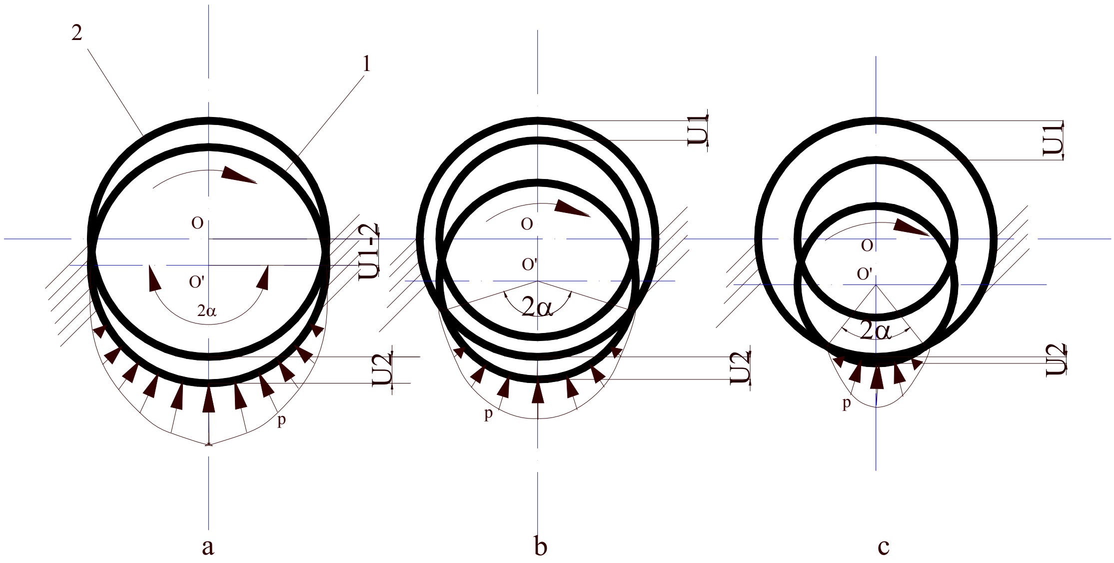

Fig. 1 Wear of shaft-sliding bearing coupling

Starting from the fully recognized hypothesis that a continuous film of lube oil (in the case of dry friction or limit lubrication) can not be formed, it is presumed that the wear is of adhesive type and the detachment of a wear particle is a cumulative phenomena of contact fatigue by means of elastic deformation [1,2], plastic deformation or elastic-plastic deformation.

We consider the linear, dimensionless intensity as the indicator of the wear process:

$$I_u = \frac{dU_x}{dL_f} \tag{1}$$

where: dU_x - linear, elementary wear in a certain point;
dL_f – friction elementary length that "passed" over that point.

As far as these values are concerned, we want to make the following observations:

The elementary friction length is detailed according to the kinematics conditions specific for the coupling:
- for the (shaft) neck

$$dL_{fa} = 2Rd\alpha \tag{2}$$

- for the bearing backing

$$L_{fa} = 2\pi R \tag{3}$$

where α – contact angle between the neck and the backing, which depends on the loading, on the properties of the materials of the coupling and on the geometry of the elements.
- elementary wear

$$dU = kpv\, dt = kpvt\, d\alpha / 2 \tag{4}$$

From the tribological point of view, the wear intensity depends on the material's characteristics (elasticity mode E, breaking tension or flow tension in one cycle), on micro geometry's characteristics and loading conditions (contact pressure p, speed, and friction coefficient μ) [2]:

$$I_u = k\left(\frac{p}{E}\right)^a \left(\frac{\mu E}{\sigma_o}\right)^b \tag{5}$$

where a, b, k – constant dependant on the characteristics of the micro geometry and on the material. Given these conditions, the elementary wear shall be:

$$dU_x = k\left(\frac{p}{E}\right)^a \left(\frac{\mu E}{\sigma_o}\right)^b dL_f \tag{6}$$

It is well known that in the sliding process, the mechanical work of friction generates a certain amount of heat, which dissipates into the elements of the coupling. In the process of flux transmission and in the presence of relative motion, the phenomena take place at the level of real contact area, so that we can talk about an instantaneous temperature at the level of contact micro-areas ("flash") and an average temperature of the nominal surface [3, 4, 5].

Part of this heat flow is distributed into the shaft and the rest into the backing bearing. The evaluation of the heat partition coefficient in the shaft and backing bearing is based on Jaeger's hypothesis [3, 5, 7, 8], namely that the average temperature on the mobile surface (shaft) equals the temperature on the fix surface (bearing). It is also considered that the dissipation of the heat flow is done according to Peclet's number (Pe – dimensionless speed), which depends on the diffusivity of the material (neck), coupling's geometry and sliding speed. So, we can take into consideration three heat flow variation ranges, for which Pe≤0.1; 0.1<Pe<0.5 and Pe≥5.

3. THE INFLUENCE OF THE HEAT FLOW ON THE WEAR INTENSITY

In the specialized literature [4, 5, 6, 7] the melting temperature for the irregularities has the form:

$$T = 4\alpha\mu\beta p Pe \tag{7}$$

where:
α – heat partition coefficient
μ – molecular friction coefficient
β – dimensional coefficient

For the three variation ranges of the partition coefficient in relation 6, the variation of the pressure on the surface will be:

$$p = \frac{T(1+\lambda_2/\lambda_1) \quad 4\beta P_e \tau_o}{4\beta P_e \mu_m} \qquad \text{for} \quad P_e < 0{,}1 \tag{8}$$

$$p = \frac{T\left[1+0{,}795(\lambda_2/\lambda_1)(a_1/a_2)^{1/2} P_e^{\;1/2} \quad 4\beta P_e \tau_o\right]}{4\beta P_e \mu_m} \qquad \text{for} \quad P_e > 5 \tag{9}$$

$$p = \frac{T \quad 4\beta P_e \tau_o\left[(1{,}02\alpha_{01} \quad 0{,}02\alpha_5) + 0{,}204(\alpha_5 \quad \alpha_{01})P_e\right]}{4\beta P_e \mu_m\left[(1{,}02\alpha_{01} \quad 0{,}02\alpha_5) + 0{,}204(\alpha_5 \quad \alpha_{01})P_e\right]} \quad \text{pentru} \;\; 0{,}1 < P_e < 5 \tag{10}$$

where:
$\alpha_{01,}$ α_5 – partition coefficients calculated for Pe=0.1, Pe=5
λ_1 λ_2 – thermal conductivity for the materials of the coupling
$\beta = l_m/L$
l_m – distance of heat diffusion in the coupling

L – square side equivalent to heat generation and dissipation
τ_0 – shearing stress for absorbed and/or chemisorbed layers of the lube oil on the solid surface

Fig. 2 presents the evolution of pressure for half of the contact surface (2x/2), depending on the dimensionless speed, and temperature at the contact surface for a coupling having known dimensional characteristics.

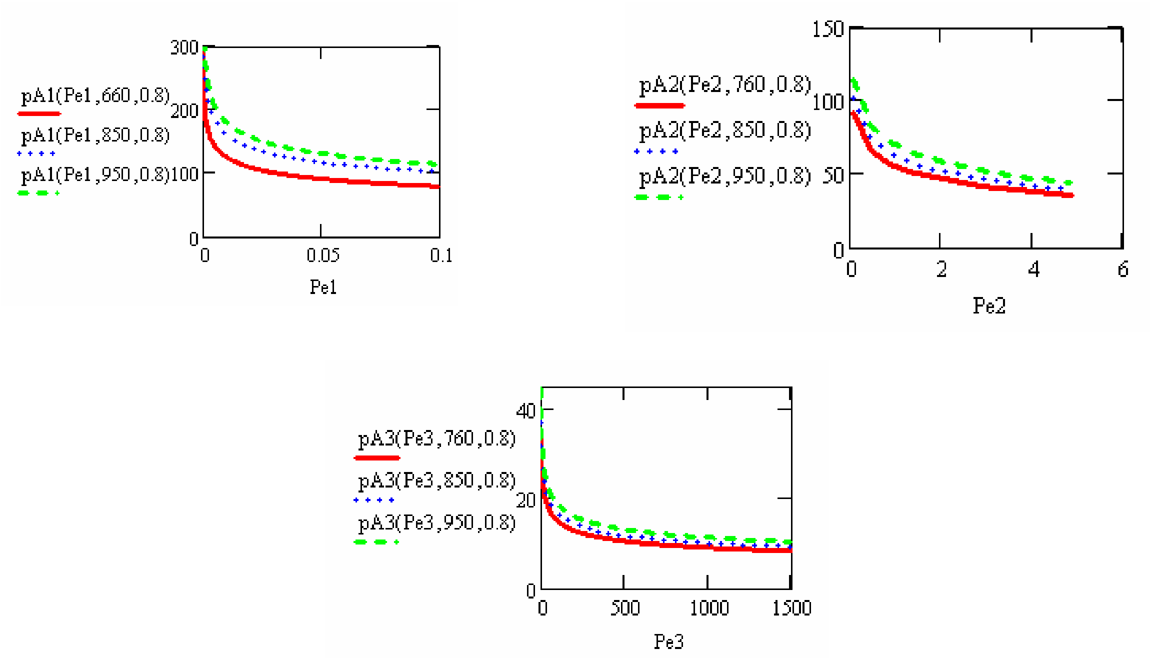

Fig. 2 Pressure evolution for contact surface depending on the dimensionless speed and temperature

The effects of the increase in temperature and deformation on the wear intensity is evaluated by means of a reduction in hardness and micro hardness and, implicitly, by the appearance of local plastic deformations and, in this way, an increase in coupling's elements clearance.
Dispersion of hardness value Law and deformations of the type Lim and Ashby [3, 4, 5, 6, 7, 8] are accepted, which for the majority of the metals can be written as follows:

$$\frac{H}{H_o} = 1 \; \frac{T_m \quad T_o}{20T_t} \; \ln\frac{10^6}{\beta\, Pe} \tag{11}$$

where:
H – modified hardness
H_0 – hardness for temperature T_0= 20°C
T_m – average temperature at the contact level of irregularities
T_t – melting temperature of the material

Fig. 3 presents the hardness evolution with pressure, speed and temperature for the same dimensional characteristics of the coupling.

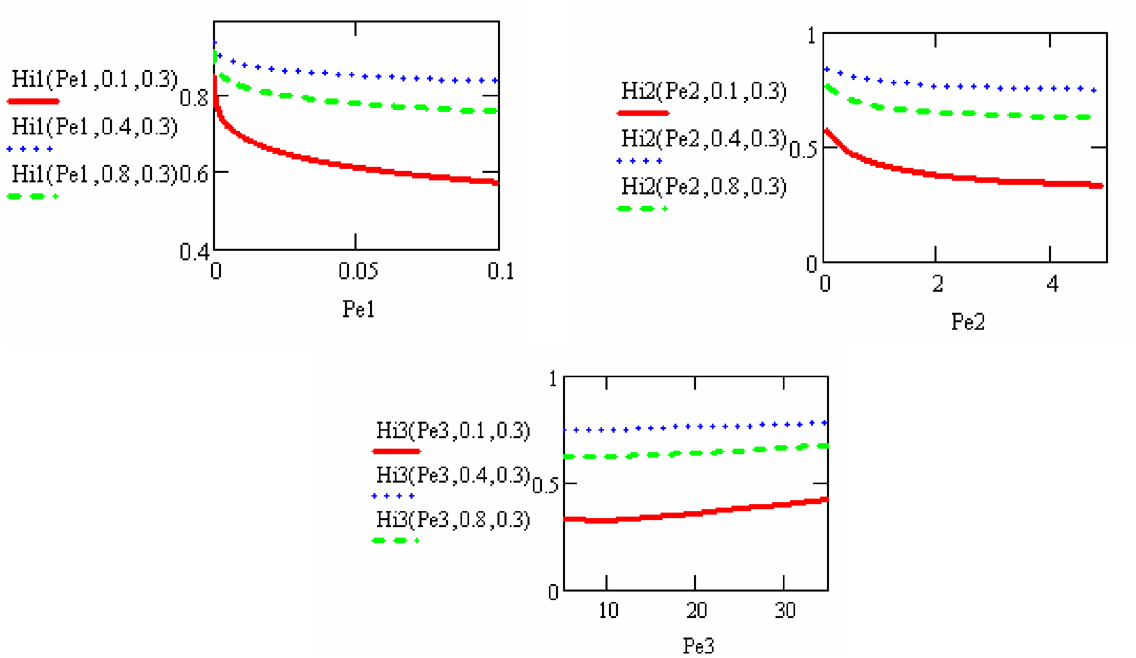

Fig. 3 Hardness evolution with pressure, speed and temperature

The variation of the wear intensity for a coupling having known characteristics is presented in Fig. 4.

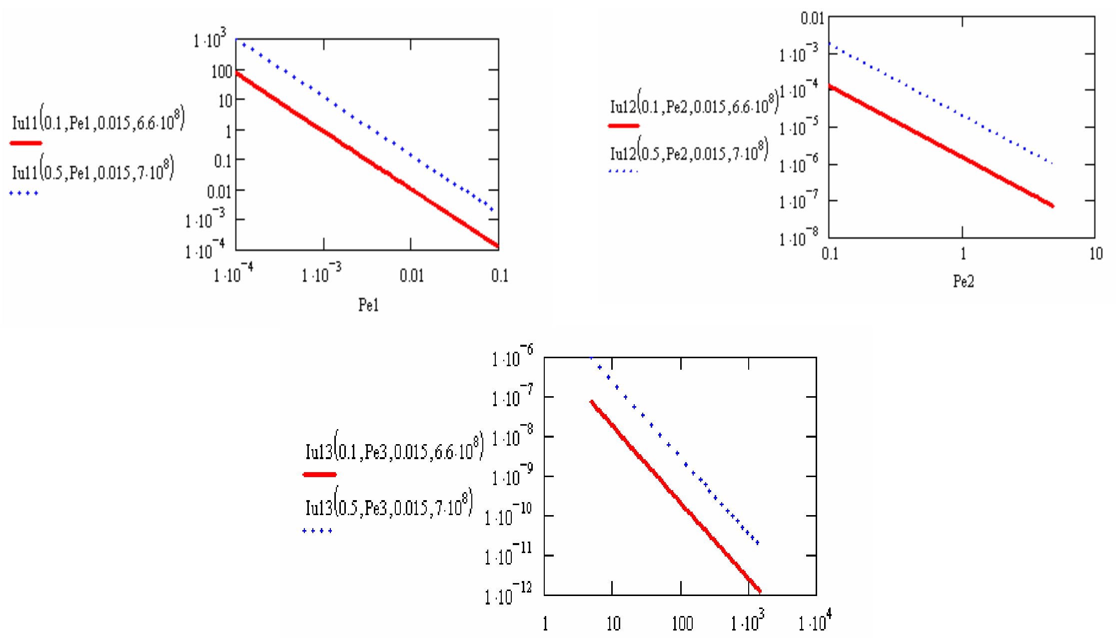

Fig. 4 The variation of wear intensity

Fig. 5 presents the evolution of the wear for a shaft-sliding bearing type of coupling, depending on the pressure, speed and temperature.

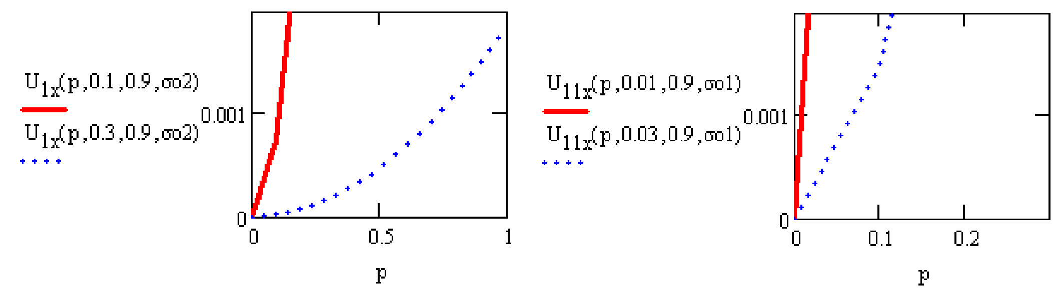

Fig. 5 Wear evolution for shaft (a) and bearing (b)

4. CONCLUSIONS

The paper deals with a more complex calculation of the wear (for a shaft-sliding bearing coupling, considering the different factors that influence the wear size.
Out of the wear analysis of this type of coupling, one can notice that the size of the wear decreases once the speed of sliding increases and increases as the functioning temperature increases also, things which are confirmed also by the specialized literature.

REFERENCES

Kraghelski, I.V., *Calculation of Wear Rate*, Journal of the Conf. on lubric. And Wear, London, page 302-307, 0ct 1957
Pavelescu, D., *Conceptii noi, calcul şi aplicatii in trecarea si uzarea solidelor deformabile*, Editura Academiei Române, Bucureşti, 1971
Gecim, B. and Winer, W.O., *Transient Temperatures in the vicinity of an asperity contact*, Trans ASME, Jour. Of. Tribology, Vol 107, page 333-342, 1985;
Greenwood, J.A., *Flash Temperatures for Bodies Moving at Equal High speeds in opposite directions*, Trans ASME, Jour. Of. Tribology, Vol 118, page.255-257, 1996
Kennedy, jr. F.E., *Thermal and thermomechanical effects in dry sliding*, Wear no.100, page 453-476, 1984
Tian, X. and Kennedy, F.E., *Maximum and average flash temperatures in sliding contacts*, *Journal of Tribology*, Vol 116, page174-176, 1994
Petre, I., *Durabilitatea şi precizia ghidajelor cu alunecare*, Editura Macarie 2000
Lim, S.C. and Ashby, M.F., *Wear mechanism maps*, Acta metal, Vol 35, no.1, page.1-24, 1987.