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Research article

Accounting for the Temperature Dependence of Hardness in Numerical Simulation of Wear in Pivot-Jewel Bearings

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A B S T R A C T

Pivot-jewel bearings are important components in rotor systems and flywheel energy storage systems; the relative rotation speed can reach about 10^3 rps under nominal operating conditions, while the lifespan can amount to years. Wear calculations of the contact surfaces are necessary in such scenarios to assess the life cycle of friction joints. Accounting for the variation in the surface hardness due to friction heating can considerably affect the amount of wear in the contact surface.

1. INTRODUCTION

Miniature bearings are essential components in many devices (clocks, gyroscopes, accelerometers, electric meters, etc.), often determining their output parameters and reliability. Jewel bearings make up a special group of friction joints, used in the most critical precision devices. The main advantages of such bearings are their high wear resistance and stability of friction characteristics [1].

Aside from measuring instruments [2,3], pivot-jewel bearings find application in diverse rotor systems and flywheel energy storage [4,5]. The pivot rotation speed relative to the bearing support is 10^2–10^3 rps under nominal operating conditions of such devices, while their lifespan can reach several decades. The wear of the contact surfaces can greatly affect the overall performance of the device in such cases.

Numerical simulation is an effective method for assessing the performance of a contact pair operating under surface wear; this includes solving the wear contact problem, which consists in calculating the evolution of wear and contact pressure during wear of the friction pair. Solution of the wear contact problem can be considered as the solution of the contact problem at each instant in time for bodies whose shape changes due to wear. The law of wear acts as an evolution equation here, relating the wear rate to the contact characteristics such as contact pressure, sliding speed, etc.

The wear contact problem is based on the theory of contact interaction of deformable bodies and the field of tribology concerned with theoretical and experimental studies of wear patterns [6]. Wear is usually understood as the process of gradual removal of material from the surface of a solid and (or) an increase in its residual strain under the mechanical action of another body or medium on it. Wear manifests as the gradual change in body size, quantified by w; linear, mass and volumetric wear are distinguished [7,8].

The following classification of wear types is the most widespread [9,10]:

- abrasive wear, occurring in contact between two bodies with significantly different hardnesses or in the presence of hard particles in the intermediary layer;
- adhesive wear, occurring in contact between bodies whose hardness is the same or on the same order of magnitude;
- corrosive wear, associated with chemical modification of the surface with subsequent removal of the surface layer;
- fatigue wear, occurring when the surface is repeatedly loaded during sliding or rolling, so that each separate loading cycle does not produce noticeable changes in the surface.

2. ACCOUNTING FOR THE TEMPERATURE DEPENDENCE OF HARDNESS

2.1 Mathematical models

There are over three hundred models describing the wear process during contact interaction [11]. In this paper, we consider adhesive wear of surfaces, using the Archard model to describe it [12–14]:

$$\dot{w} = \frac{K}{H} p^n v^m \tag{1}$$

The dimensionless parameter K included in Archard's law (1) is called the wear coefficient and is a measure for the magnitude of wear in the material [15]. The wear coefficient can vary in a wide range from 10^{-14} to 10^{-2} for different materials under different friction conditions and depending on the presence of lubricants in the contact [16]. Table 1 gives the values for some polymer, ceramic and metal materials operating in the absence of lubricants [13,17-19]:

Table 1. Values of dimensionless wear coefficients.

Material	Wear coefficient
PTFE polyamide	$1.22{\times}10^{-6}$
Silicon nitride on silicon nitride	$2.16{\times}10^{-5}$
Ruby	$7.60{\times}10^{-7}$
Mild steel on mild steel	$7.00{\times}10^{-3}$
Hardened tool steel	$1.30{\times}10^{-4}$
Ferritic stainless steel	$1.70{\times}10^{-5}$

The parameter H in Eq. (1) denotes the hardness of the material, which is one of the mechanical properties often used in tribology [20] as an indicator of resistance to localized plastic deformation.

The hardness of steels [21,22], copper [23,24] and aluminum [25,26,27] alloys, ceramic materials [28,29] decreases with increasing temperature; a smooth curve can approximate this dependence with a high level of fidelity. Analyzing the above literature, we find that a smooth 5th order step function can be used as such a curve.

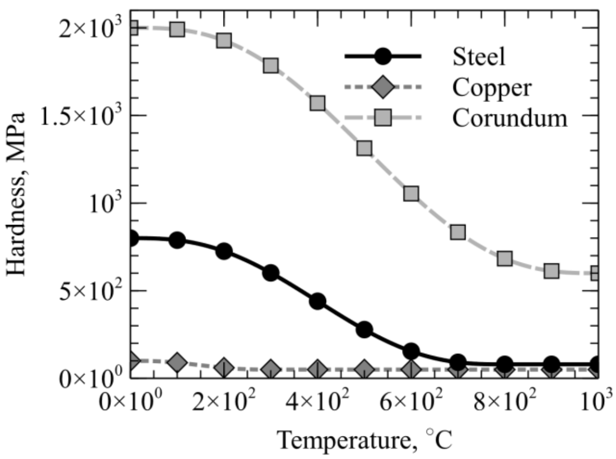

Fig. 1. Temperature dependences of hardness.

It is effective to run numerical simulation of the wear process by the finite element method for a bearing pair at high revolution during a given period in a steady-state formulation. Similar calculation methods have been proposed by numerous researchers [18, 30–33]. The difference of our study is that we account for the thermal state of the contact surface without directly calculating the temperature of the contacting bodies.

The magnitude of the heat flux generated on the contact surface is calculated assuming that the contact pair is isolated, and the work of the friction forces therefore equals the thermal energy according to the first law of thermodynamics.

The magnitude of the heat flux generated on the surface of the contact element *(e)* is determined from the relationship [34]:

$$q^{(e)} = \mu p^{(e)} v^{(e)} \quad (2)$$

where μ is the friction coefficient between the contacting surfaces.

The total heat flux is calculated as:

$$q = \frac{1}{A} \sum_{(e)} q^{(e)} A^{(e)} \quad (3)$$

where $A^{(e)}$ is the area of the contact element *(e)*, A is the total contact area, divided between two contacting surfaces in the ratio [35]:

$$q = q_{(1)} + q_{(2)} = \alpha q + (1 - \alpha) q \quad (4)$$

The coefficient α in expression (4) is calculated from the values of the thermal characteristics of the contacting materials:

$$\alpha = \left(1 + \sqrt{\frac{\kappa_{(2)} c_{(2)} \rho_{(2)}}{\kappa_{(1)} c_{(1)} \rho_{(1)}}}\right)^{-1} \quad (5)$$

where κ, c, ρ are the thermal conductivity, specific heat capacity and density of the material, respectively, the subscripts *(1)* and *(2)* correspond to the number of the contact surface.

The temperature of the contact surface is determined from the solution of a one-dimensional transient heat equation with second-kind boundary conditions imposed on one boundary and first-kind boundary conditions imposed on the other, as shown in Fig. 2:

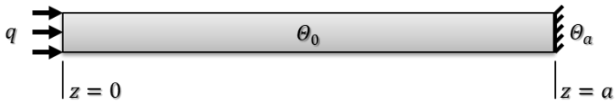

Fig. 2. Scheme for 1D transient thermal problem.

The solution of the heat equation with such boundary conditions on the contact surface (at z = 0) is formulated as:

$$\Theta(t) = \frac{4}{\pi^2 \kappa} \sum_{m=0}^{\infty} C_m e^{-D_m t} + \Theta_a + \frac{aq}{\kappa} \quad (6)$$

where

$$C_m = \frac{(-1)^m \pi \kappa (\Theta_a - \Theta_0)(2m+1) - 2aq}{(2m+1)^2} \quad (7)$$

$$D_m = \frac{\pi^2 \kappa (2m+1)^2}{4a^2 c \rho} \quad (8)$$

After determining the temperature of the contact surface (6) and knowing the approximation parameters of the temperature dependence of hardness, we can account for the variation in surface hardness solving the wear contact problem.

Numerical solution of the wear process in the contact pair was carried out in this study in an axisymmetric steady-state formulation using the ANSYS Mechanical software package.

2.2 Simulation results

The influence from taking into account the temperature dependence of hardness of the contact surface was estimated by solving several verification problems shown in Fig. 3:

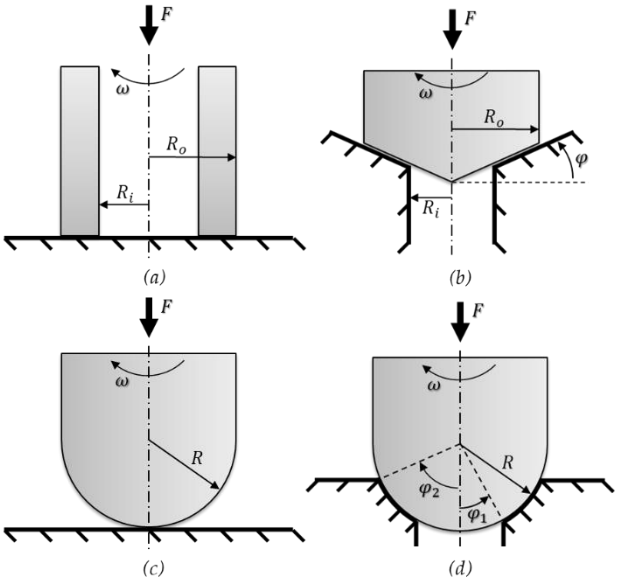

Fig. 3. Verification problems.

The volumetric wear of the material in a rotating body serves as a comparable value. Analytical expressions for estimating wear in these verification problems without accounting for the temperature dependence of hardness are taken from the literature [36,37] and are given in Eq.9-Eq.12:

a: $$W = \frac{1}{2}\frac{K}{H}F(R_i + R_o)\omega t \quad (9)$$

b: $$W = \frac{1}{2}\frac{K}{H}\frac{F}{\cos\varphi}(R_i + R_o)\omega t \quad (10)$$

c: $$W = \frac{3}{8}\left(\frac{3R}{2\pi}\right)^{\frac{1}{3}}\left(\frac{K}{H}F\omega t\right)^{\frac{4}{3}} \quad (11)$$

d: $$W = \frac{K}{H}FR\frac{\cos 2\varphi_1 - \cos 2\varphi_2}{\sin 2\varphi_2 + 2\varphi_2 - \sin 2\varphi_1 - 2\varphi_1}\omega t \quad (12)$$

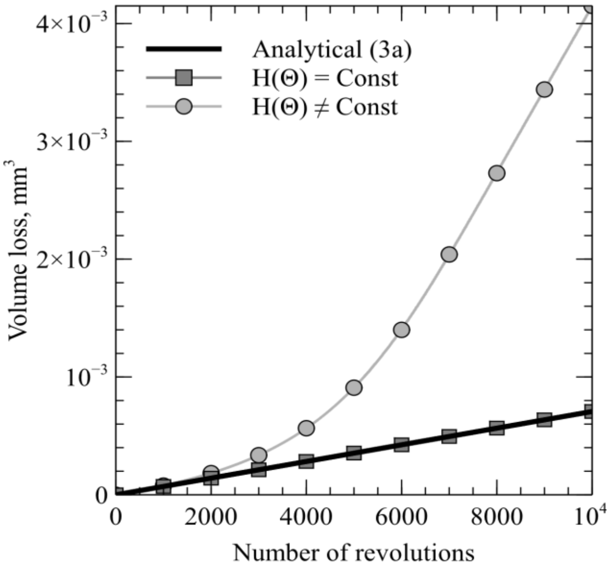

Fig. 4. Influence of temperature dependence of hardness on wear of flat surface.

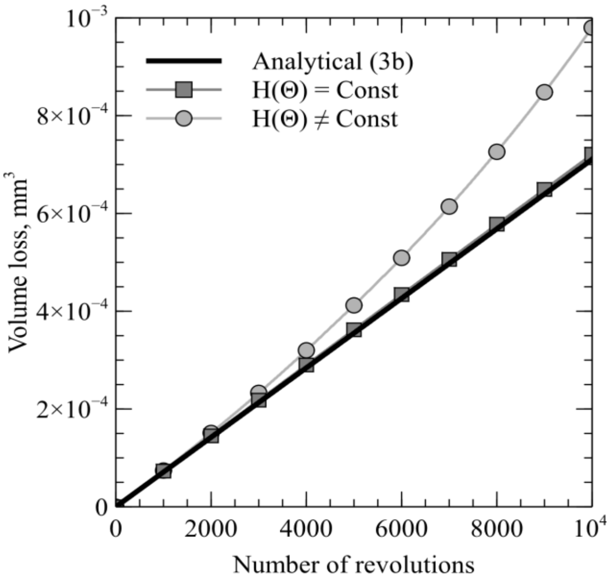

Fig. 5. Influence of temperature dependence of hardness on wear of conical surface.

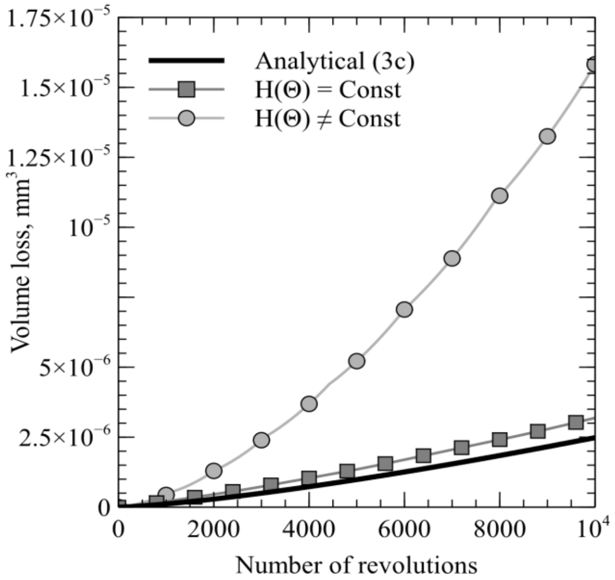

Fig. 6. Influence of temperature dependence of hardness on wear of spherical surface on plane.

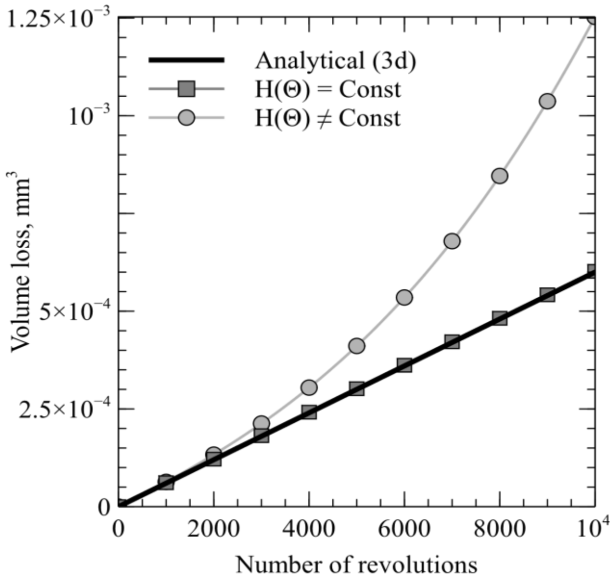

Fig. 7. Influence of temperature dependence of hardness on wear of spherical surface.

Fig. 4 – Fig. 7 shows the curves reflecting the variation in volumetric wear in the contact pairs from Fig. 3, compared with the analytical estimates of volumetric wear values; additionally, the figure illustrates the influence from taking into account the changes in the material hardness induced by heating of the contact surface.

3. CONCLUSION

Taking into account the variation in the hardness of the material due to frictional heating can significantly affect the life cycle assessment of friction joints in wear simulations of jewel bearings.

The proposed method for determining the surface temperature is based on analytical calculation of shear velocities, heat flux and temperature; it is applicable for steady-state calculations.
The boundary conditions of the transient differential equation of thermal conductivity, used to calculate the temperature, allow to account for heat removal from the contact surface.

Thus, this method can be used to account for frictional heating in wear simulations by the finite element method in a steady-state formulation (in particular, an axisymmetric one).

A large number of parameters can complicate the fine-tuning of the model and validation of the results.

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