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A study of the surface roughness in elasto-plastic shrink fitted joint ☆

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ABSTRACT

The aim of the present study is to provide a methodology for improving the design of a shrink-fitted assembly by focusing on two objectives: (i) taking into account the surface roughness of components, (ii) introducing a non-classical hardening friction law in which tangential micro-slip displacements are considered. The calculations have been made for elasto-plastic properties of component in axisymmetric conditions. Both the loading and unloading processes have been considered. After finite element simulations, it is observed that overall static behaviour of the assembly is influenced considerably by the surface finish of the mating components. The current numerical results of the shrink fitted assembly received from the coarse mesh could be treated as contact benchmark for further finite element analyses using as closely as possible identical input data.

1. Introduction

Connections in machine joints can be classified as fixed (interference fits, bolted joints, polygon joints) and sliding or moving joints (tool slideways, dovetail slides). Interference fits, in turn, are classified as press fits and shrink fits. In press fits the mating elements are pressed against each other to overcome the size difference between them and assembled without changing the temperature of either part. In thermal assemblies the elements are subjected to thermal expansion (heating of the hub or cooling of the shaft) resulting in, respectively, expansion and contraction of the components and enabling easy assembly. Due to low cost of manufacture, easy assembly and compact integrity, shrink fits and press fits, providing high torque moments and axial forces, are successfully replacing conventional mechanical fastening devices using keys and splines or other rigid couplings.

In the presence of friction, accuracy of the mathematical analysis much depends on the possibility of accounting for irreversible frictional micro-displacements (or micro-slip) and roughness of the contacting bodies. They are important for the friction fretting processes and result in premature wear of machine elements. Forces in both press and shrink joints are transmitted across the joint interfaces. Therefore, we can expect that overall static and dynamic behaviour of the machine tool is influenced largely by the surface finish of the mating components.

The paper presents some remarks on factors affecting the characteristics of shrink-fit assemblies in presence of surface roughness and micro-slip. The resulting factors influencing the contact zone will be explained in terms of the deformation of asperities which are always present, even on the finest machined surfaces. When these surfaces are loaded together, contact is made at the asperity summits. It is generally observed that the better surface finish the strength of the assembly is greater [1].

Only few studies have been done to take roughness into account in designs of interference-fit assemblies. Yang et al. [2] showed that for some surface texture topology, roughness has a noticeable influence on fit strength. In the paper of Boutoutaou et al. [3], using the homogenization technique, they demonstrated that the form of the both surfaces in contact is the key factor in determination of the strength of the assembly. French standard NF E22-621 indicates that loss of the tightening due to roughness should be estimated from the arithmetic roughness of the both contacting surfaces in the following form:

$$L_{\Delta} = 3(R_{a1} + R_{a2}) \tag{1}$$

where R_{a1} and R_{a2} are the arithmetic roughness of the hub and shaft, respectively. This relation is empirical and has no scientific background [3]. Boutoutaou et al. [4] and Zhang et al. [5] proposed an effective approach using finite element simulations to achieve the required quality at lower manufacturing costs of interference-fitted connections. Some researches also suggest that the shrink fit design formulations should take into account stick-slip phenomena and they indicate that current and widely used design formulae are inadequate in prediction of the holding torque [6,7].

☆Dedicated to Prof. Peter Wriggers on the occasion of his 65th birthday.

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Nomenclature

p_N	contact pressure
p_T	tangential contact stress
W	normal contact load
A_0	nominal contact area
k_N	normal contact stiffness
k_T	tangential contact stiffness
R_{a1}	arithmetic roughness of the hub
R_{a2}	arithmetic roughness of the shaft
p	axial pressure acting on the front wall of the hub
$p_{peak}(\zeta, t)$	second-order or joint probability density function
$t=-\kappa/\sigma_\kappa$	normalized curvatures
$\zeta=z/\sigma$	normalized height
$\eta=d/\sigma$	normalized separation
σ	standard deviation of the rough surface (arithmetic roughness)
σ_κ	standard deviation of the peak curvatures
D	fractal dimension of a surface profile
D_s	density of summits
E	effective Young's modulus
μ_m	macroscopic (or static) coefficient of friction
n	slip hardening parameter
β	initial value of coefficient of macroscopic friction of μ_m
$\Vert \mathbf{u}_T^{\mathrm{p}} \Vert = u_{T_{\mathrm{eff}}}^{\mathrm{p}}$	effective plastic tangential displacement

There are various methods employed to deal with surface roughness. One of these methods is based on statistical and fractal modelling. Still another approach consists in experimental evaluation of the load-displacement characteristics of real models of rough surfaces and description of the load-displacement relations using simple mathematical equations [8].

2. Friction modelling

When a tangential force is applied to two rough bodies in contact there is a certain tangential movement between the bodies before the coefficient of friction reaches the value at which gross slip occurs. That phenomenon has been called as micro-slip (or pre-sliding phase). The experiments carried out on metallic materials indicate that in the pre-sliding phase the coefficient of friction increased with the plastic displacement approaching the macroscopic coefficient of friction μ_m [9]. If we introduce three independent axial parameters: macroscopic coefficient of friction μ_m, slip hardening parameters n and the initial coefficients of friction β, the following relationships can be written [9]:

$$\frac{\mu_F}{\mu_m} = 1-(1-\beta)\exp(-n \Vert \mathbf{u}_T^{\mathrm{p}} \Vert) \tag{2}$$

where μ_m is macroscopic (or static) coefficient of friction, β defines initial value of μ_m, n denotes hardening parameter and $\Vert \mathbf{u}_T^{\mathrm{p}} \Vert = u_{T_{\mathrm{eff}}}^{\mathrm{p}}$ is the effective plastic tangential displacement.

A phenomenological description of the frictional phenomena is based here on a similarity of friction and elasto-plastic behaviour. The main idea is not new. It seems that for the first time this approach has been introduced by Seguchi et al., 1974 (*Computational Methods in Nonlinear Mechanics*, University of Texas, pp. 683–692), then developed later, independently, by Fredriksson [9], in the context of the finite element linear approach, Michałowski and Mróz [10] in relation to geomechanics issues and then, ten years later by Wriggers [11] in the case of large deformation contact problem via consistent linearization. We present much more general model than has usually been realized in which the surface roughness of assembly components and a non-classical hardening friction law will be considered. The principal features of this model are: (i) decomposition of the contact displacements into an elastic part (describing the preliminary micro-slip) and a plastic part, (ii) introduction of a slip criterion and a slip potential, (iii) use of a non-associated slip rule for the contact of metallic bodies (non-dilatancy effect), and (iv) inclusion of contact stiffness parameters, respectively. In the following we introduce the isotropic slip criterion or sliding function f which is specified in terms of tangential tractions $\mathbf{t}_T$ and contact pressure t_N. Let us approximate the limit friction condition by a paraboloid slip surface

$$f(t_T, t_N) = \Vert \mathbf{t}_T \Vert - \mu_F t_N = 0, \tag{3}$$

where μ_F is the friction coefficient defined by Eq. (2).

The following additive relation is assumed for the incremental elasto-plastic sliding model

$$\Delta \mathbf{u} = (\Delta \mathbf{u}_T^{\mathrm{e}} + \Delta \mathbf{u}_T^{\mathrm{p}}) + \Delta \mathrm{u}_N \mathbf{n}, \tag{4}$$

with the contact displacements indexed by e and p corresponding to the elastic (reversible) and plastic (irreversible) behaviour, respectively and $\mathbf{n}$ is the normal vector to the contact surface.

3. Contact stiffness and constitutive relations

3.1. Normal contact stiffness

The elastic normal stiffness per unit area is adopted from [12], so we have

$$k_n = \frac{4}{3} E \left(\frac{3-D}{2-D}\right) D_s\, \sigma^{1/2} \int_\eta^\infty \int_0^\infty \sqrt{1/(\sigma_\kappa t)} (\zeta-\eta)^{1/2}\, p_{peak}(\zeta, t)\, \mathrm{d}\zeta\, \mathrm{d}t \tag{5}$$

where $p_{peak}(\zeta, t)$ is the second-order or joint probability density function of the normalized heights ζ and curvatures $t=-\kappa/\sigma_\kappa$, D is the fractal dimension of a surface profile $(1<D<2)$ and D_s is the density of summits. The effective Young's modulus E is given in Eq. (8).

The normalized height ζ and the normalized separation η are defined as follows:

$$\zeta = z/\sigma, \quad \eta = d/\sigma \tag{6}$$

where σ and σ_κ are standard deviation of the rough surface and curvature, respectively.

The values of profilometric parameters of the examined surfaces are given in Table 1. The results of both the calculated and experimental contact stiffness for different kind of machined processes have been recently presented in [12]. The surfaces of the

Table 1
Surface roughness values for different machining processes.

Surface roughness parameters	FSB	CSB	EDM
Arithmetic mean deviation, R_a (μm)	0.832	5.13	8.94
RMS deviation, $\sigma=R_q$ (μm)	1.08	6.67	11.62
Density of summits, D_p ($1/\mu m^2$)	500	230	160
Arithmetic mean peak radius, R (μm)	40	30	19
Fractal parameter, D	1.62	1.58	1.70