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Influence of microgeometry in the point contact zone of rest friction on fatigue life for friction bearing units

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Abstract

A comprehensive methodology has been developed to assess the maximum contact stresses and deformations in the point contact zone and the maximum tangential stresses, including their position in the subsurface zone in depth and in the rolling direction when the microgeometry of the rest contact.

Key words: Point contact zone, fatigue life, friction bearing units (bearing assemblies), maximum contact stresses and deformations, maximum tangential stresses, microgeometry.

Introduction

One of the most important factors limiting the durability of bearing units is fatigue damage, namely pitting, which occurs during repeated cyclic loading. Since metal elements are damaged, fatigue is usually associated with the problem of metal stability and microgeometry of the contact zone. While fatigue life has been significantly increased by controlling the type and size of non-metallic inclusions, heat treatment and the introduction of alloying additives to the base metal, little or no thorough research has been paid to the influence of microgeometry in the contact zone, especially for bearing assemblies.

In any case, a comprehensive calculation methodology is needed that would allow to take into account the influence of microgeometry on the fatigue life of bearing units.

The purpose of the work

To develop a comprehensive methodology for assessing the influence of microgeometry on the maximum contact stresses and deformations in point contact, the maximum tangential stresses and their penetration position in the subsurface zone along the depth and direction of rolling under resting friction conditions.

1. Technique for researching the properties of coatings for the influence of microgeometry.

The method of calculating the maximum stresses, deformations, position and value of the maximum subsurface tangential stress is necessary to assess the fatigue life of friction bearing units [1].

Hydrodynamic lubrication is characterized by surfaces that fit well together, that is, surfaces that have a high degree of geometric similarity, and the load is transferred over a relatively larger plane. In addition, the actual plane for such surfaces remains virtually constant with increasing load.

But many bearings do not have a very good surface fit. The full load falls on a relatively small plane. As a rule, the actual contact area increases significantly with increasing load, but still remains small compared to surfaces that fit well together. The loads per unit area for adjacent bearings are relatively small, about 1 MPa and rarely 7 MPa. But the load per unit area in contacts with non-adjacent surfaces, like to the contacts of ball bearings, usually exceeds 700 MPa even with a moderate load on the bearing. Such high pressures lead to elastic deformation of the materials, resulting in elliptical contacts that can support these loads. Therefore, appropriate simplified calculations of stresses and deformations in the contacts of non-adjacent surfaces are required.

To model a real friction unit, it is necessary to know the field of static contact of interacting bodies. Determination of the properties of static contact of interacting parts in non-conformal (with local contact) nodes is reduced to determining the type and type of contact field, maximum contact and tangential stresses and strains in the field and within the contact field (Fig. 1).

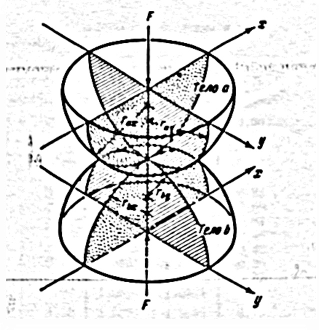


Fig. 1 - Geometry with point contact friction

In general, the geometry of undeformed contacting bodies can be represented by pressing two ellipsoids. Two bodies with different radii of curvature in the two principal planes (x and y), passing through the contact between the bodies, touch at one point at zero load. This state is called point contact, in which the radii of curvature are denoted by r (see Fig. 1). It is assumed that convex bodies have positive curvature and concave bodies have negative curvature. Thus, if the center of curvature lies inside the body, the radius of curvature is positive, otherwise it is negative.

It is important to note that if the choice of x and y coordinates satisfies the condition:

$$\frac{1}{r_{ax}}+\frac{1}{r_{bx}} \geq \frac{1}{r_{ay}}+\frac{1}{r_{by}}, \tag{1}$$

then the x-coordinate determines the direction of the minor semi-axis and the y-coordinate determines the major semi-axis of the contact ellipse that occurs when the load is applied. The direction of motion is always given along the x-axis. The sum of curvatures (reduced radius of curvature), which is necessary in the analysis of contact stresses and strains, is determined by the following formula:

$$\frac{1}{R}=\frac{1}{R_x}+\frac{1}{R_y}, \tag{2}$$

where:

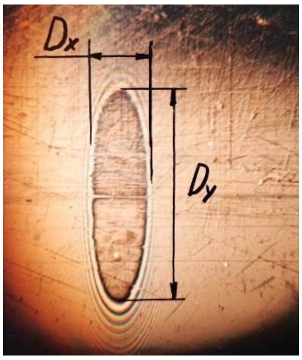
$$\frac{1}{R_x}=\frac{1}{r_{ax}}+\frac{1}{r_{bx}}, \tag{3}$$

$$\frac{1}{R_y}=\frac{1}{r_{ay}}+\frac{1}{r_{by}}. \tag{4}$$

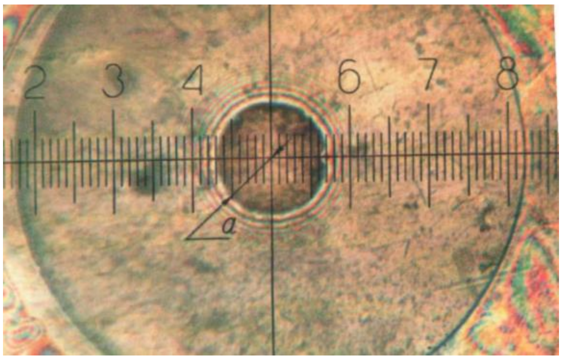
The ratio of radii of curves α is determined by the following formula:

$$\alpha=\frac{R_y}{R_x}. \tag{5}$$

The shape of the plane of such contacts is called point contacts. Point contacts can be elliptical (Fig. 2, *a*), when the ratio of radii of curvature $\alpha \neq 1$, or circular (Fig. 2, *b*), when $\alpha = 1$, since $r_{ax} = r_{ay}$ and $r_{bx} = r_{by}$, then according to expressions (3) and (4), it turns out that the radii of curvature $R_x = R_y = r/2$. If the radii of curvature r_{ay} and r_{by} are infinite, the initial linear contact is transformed into a rectangular contact under load.



a – elliptical contact



б – circular contact

Fig. 2 *a, b*. Shape of friction point contact at change of microgeometry, made by optical interferometry

The ellipticity parameter k is defined as the ratio of the diameter of the elliptical contact in the y direction (transverse direction) to the diameter in the x direction (direction of movement):

$$k \equiv \frac{D_y}{D_x}. \tag{6}$$

If condition (6) is satisfied and $\alpha \geq 1$, then the contact ellipse will be oriented with a large diameter across the direction of movement (see Fig. 2, *a*), i.e., $k \geq 1$, which is characteristic of the contact form formed in ball bearings with an outer ring and tubular roller bearings. Circular contact (see Fig. 2, *b*), where $\alpha = 1$, $k = 1$, is characteristic of ball bearings with self-aligning outer ring. In the elliptical contact, in which $\alpha < 1$, $k < 1$, the contact ellipse, on the contrary, will be oriented with a small diameter across the direction of movement and is characteristic of some gears and locomotive wheel contact on the rail (this option was not considered in this work).

If two elastic bodies are brought into contact under load, a plane appears, the shape and size of which depends on the applied load, the elastic properties of the materials and the microgeometry of the contact.

Under conditions of elastohydrodynamic (EHD) lubrication, two surfaces are separated by a lubricating layer, the thickness of which has the shape shown in Fig. 3.

With the usual parabolic approximation for the shape of an undeformed film, the thickness of the lubricating layer under deformation will be as follows:

$$h(x;y) = h_o + \frac{x^2+y^2}{2\cdot R} + d(x;y) - d(0;0), \tag{7}$$

where x; y - Cartesian coordinates.

The total normal deformation $d(x_1; y_1)$ of two surfaces is defined by the following equation:

$$d(x_1;y_1) = \frac{1}{\pi\cdot E'}\iint_A \frac{p(x;y)\cdot dx\cdot dy}{((x-x_1)^2+(y-y_1)^2)^{\frac{1}{2}}}, \tag{8}$$

where the reduced modulus of elasticity E′ is equal:

$$E' = \frac{2}{\left(\frac{1-\nu_1^2}{E_1}+\frac{1-\nu_2^2}{E_2}\right)}, \tag{9}$$

where E_1 and E_2 are the elastic modulus of the 1st and 2nd bodies in contact with each other; ν_1; ν_2 are the Poisson's ratios of the 1st and 2nd bodies, respectively; x_1; y_1 are the nodal points along the x and y axes.

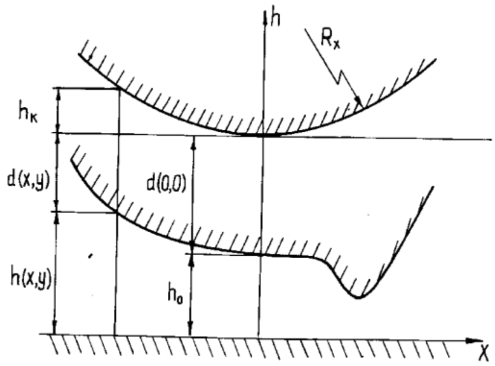


Fig. 3. Microgeometry of circular friction contact taking into account elastic deformations:
h_0 - central thickness of the lubricating layer; d(0;0) - the value of elastic deformation in the central contact area; h(x; y), d(x; y) - current value of thickness and deformation; $h_k = (x^2+y^2)/2R$ - current value of thickness at h_0 = 0; R_x - equivalent radius of curvature in the X plane.

For an approximate calculation of deformations and stresses in point contact, a simplified calculation of stresses and deformations can be used according to the method [2], which allows solving the classical Hertz problem without the use of complex mathematical calculations on a computer using simplified formulas.

The classical Hertzian solution for deformations requires the calculation of the ellipticity parameters k and the calculation of elliptic integrals of the first ψ and second ε kind. For point friction contact, the parameters ψ and ε as functions of α are simplified by means of approximating curves. These parameters make it possible to determine the deformation δ in the center of contact with a small loss of accuracy, but without the use of complex mathematical calculations when using diagrams, as well as the maximum contact stress σ_{max} in the center of contact depending on the ratio of the radii of the curves α.

The maximum contact stress in the center of the point contact σ_{max} is calculated by the following formulas:

- for circular contact:

$$\sigma_{max} = \frac{3 \cdot F}{2 \cdot \pi \cdot a^2}, \quad (10)$$

where F is the applied load, N;

$a = \left(\frac{6 \cdot \varepsilon \cdot F \cdot R}{\pi \cdot E'}\right)^{\frac{1}{3}}$ – circumferential contact radius, m.

- for elliptical contact:

$$\sigma_{max} = \frac{6 \cdot F}{\pi \cdot D_y \cdot D_x}, \quad (11)$$

where F is the applied load, N;

$D_y = 2 \cdot ((6 \cdot k^2 \cdot \varepsilon \cdot F \cdot R)/\pi \cdot E')$ – diameter of the major axis of the elliptical contact, m (see Fig. 2, *a*);

$D_x = 2 \cdot ((6 \cdot \varepsilon \cdot F \cdot R)/\pi \cdot k \cdot E')$ – diameter of the small axis of the elliptical contact, m (see Fig. 2, *a*).

The maximum deformation in the central contact zone δ is calculated by the following formulas:

- for circular contact:

$$\delta = \psi \cdot \left[\left(\frac{4{,}5}{\varepsilon \cdot R}\right) \cdot \left(\frac{F}{\pi \cdot E'}\right)^2\right]^{\frac{1}{3}}, \quad (12)$$

- for elliptical contact:

$$\delta = \psi \cdot \left[\left(\frac{4{,}5}{\varepsilon \cdot R}\right) \cdot \left(\frac{F}{\pi \cdot k \cdot E'}\right)^2\right]^{\frac{1}{3}}. \quad (13)$$

One of the causes of wear is material fatigue caused by cyclic strong and elastic deformations on the surface. Fatigue cracks are formed at a certain depth in the plane parallel to the rolling direction. Therefore, special attention is paid to the amplitude of the tangential stress in the part of the plane where it reaches a maximum.

The value of the maximum tangential stress of the point contact max is determined by the formula:

$$\tau_{max} = \sigma_{max} \cdot \frac{\sqrt{2 \cdot t - 1}}{2 \cdot t \cdot (t+1)}, \quad (14)$$

where $t = 1 + 0{,}16 \cdot csch\,(\alpha^{2\pi}/2)$ – a reduced auxiliary parameter.

It should be noted that max represents the maximum half-amplitude of the subsurface orthogonal tangential stress.

Taking into account that the stresses are referred to a rectangular coordinate system with the origin at the center of contact, the z - axis coinciding with the internal normal of the body under consideration, the x - axis along the rolling direction and the y - axis perpendicular to it, we find the position of the maximum point (depth) max in the xz- plane:

- for the circular contact:

$$|Z_0| = \frac{a}{(t+1) \cdot \sqrt{2t-1}}; \quad (15)$$

$$|X_0| = \frac{t}{t+1} \cdot \sqrt{\frac{2t+1}{2t-1}} \cdot a, \quad (16)$$

- for elliptical contact:

$$|Z_0| = \frac{D_x}{2 \cdot (t+1) \cdot \sqrt{2t-1}}; \quad (17)$$

$$|X_0| = \frac{t}{t+1} \cdot \sqrt{\frac{2t+1}{2t-1}} \cdot \frac{D_x}{2}. \quad (18)$$

Results of calculations for the influence of microgeometry

Below are the input parameters of microgeometry, materials and values of elliptic integrals of the 1st and 2nd kinds for simplified calculation of stresses and strains of friction point contacts (respectively for circular and elliptic contacts) in the range α ≤ 100 (Table 1) for two selected bearings.

Table1

The input parameters of microgeometry, materials and values of elliptic integrals of the 1st and 2nd kinds

Input and initial design parameters	Circular contact, $\alpha = 1$	Elliptical contact, $\alpha = 6$
1. The reduced radius of curvature, R_x, mm	6,35	15,24
2. Ellipticity parameter, k	1	5
3. Ellipticity parameter of the 1st kind, $\psi = \frac{\pi}{2} + \left(\frac{\pi}{2} - 1\right) \cdot ln\alpha$	1,5708	2,5935
4. Ellipticity parameter of the 2nd kind, $\varepsilon = 1 + ((\frac{\pi}{2} - 1)/\alpha)$	1,5708	1,0951
5. Auxiliary parameter t	1,307	1,070
6. Modulus of elasticity E_1, Pa	$2{,}07 \cdot 10^{11}$	$2{,}07 \cdot 10^{11}$
7. Modulus of elasticity E_2, Pa	$0{,}757 \cdot 10^{11}$	$0{,}757 \cdot 10^{11}$
8. Poisson's ratio υ_1	0,30	0,30
9. Poisson's ratio υ_2	0,25	0,25

The results of a simplified calculation of the maximum contact stresses, deformations, maximum tangential subsurface stresses and its position in the xz - plane with an increase in the applied load for two bearing units with different microgeometry of contact are presented graphically in Fig. 4 - 8.

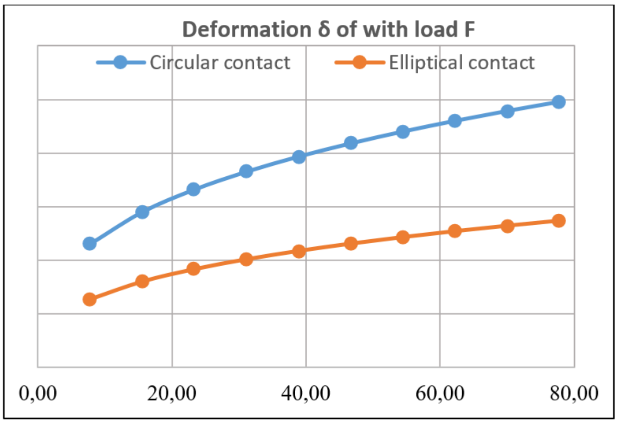


Fig. 4. Influence of microgeometry in the point contact zone on the deformation δ of contacting friction surfaces of bearing units with increasing load F

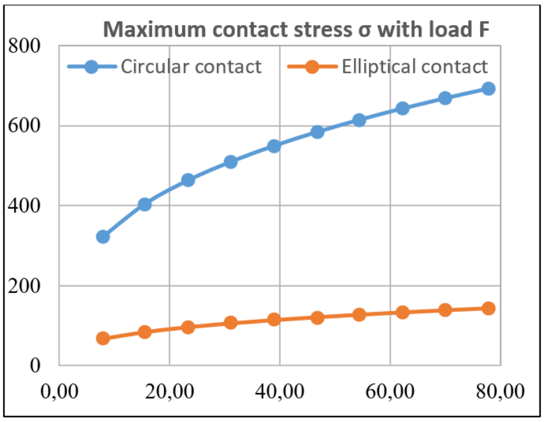


Fig. 5. Influence of microgeometry in the point contact zone on the change in contact stress σ for friction bearing units with increasing load F

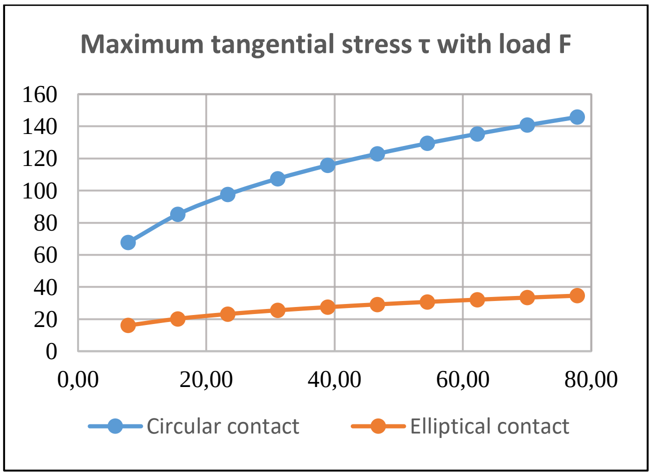


Fig. 6. Effect of microgeometry in the point contact zone on the change in tangential stress τ for friction bearing units with increasing load F

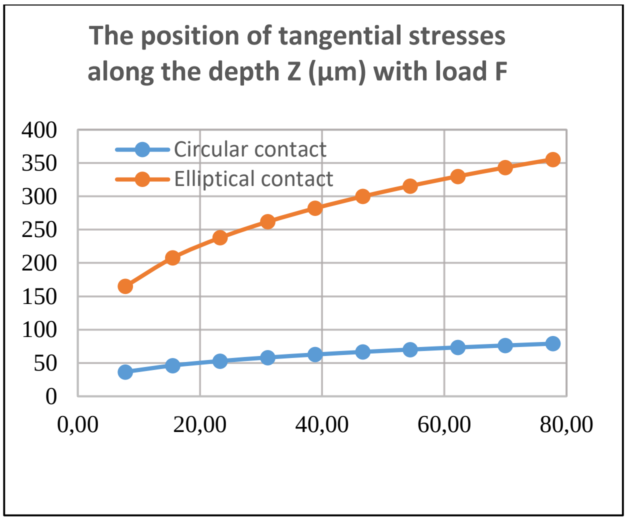


Fig. 7. Influence of microgeometry on the position of penetration of tangential stresses τ in the subsurface zone of point contact along the depth z - (µm) for friction bearing units with increasing load F

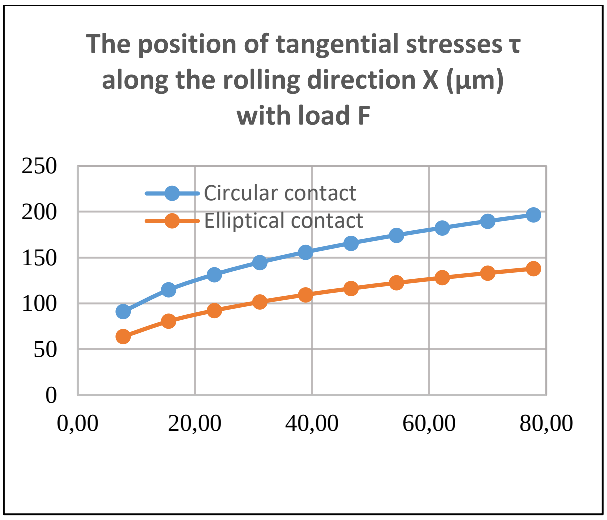


Fig. 8. Influence of microgeometry on the position of penetration of tangential stresses τ in the subsurface zone of point contact along the rolling direction x - (µm) for friction bearing units with increasing load F

Conclusions

An alternative method of calculation to the classical Hertz solution for local stresses and strains of two elastic contacting bodies is presented, i.e., the need to solve transcendental equations to establish the influence of microgeometry in the contact zone is eliminated.

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Міланенко О.А. Вплив мікрогеометрії в зоні точкового контакту на втомну довговічність в умовах тертя спокою для підшипникових вузлів тертя.

Представлена альтернативна методика розрахунку класичному рішенню Герца для локальних напружень і деформацій двох пружних контактуючих тіл, тобто, усувається необхідність вирішувати трансцендентні рівняння для встановлення впливу мікрогеометрії в зоні контакту.

За допомогою спрощених формул можна безпосередньо розрахувати максимальні контактні напруження, деформації, підповерхневі дотичні напруження, а також їх положення по глибині та за напрямом кочення та встановити вплив мікрогеометрії.

Максимальні контактні й дотичні напруження, а також максимальні деформації прогнозовано вище за величиною для упорних (осьових) кулькових підшипниках з коловим контактом тертя в порівнянні з радіальними підшипниками, які мають еліптичну форму контакту.

Для радіальних підшипників кочення з характерною еліптичною формою контакту положення проникнення дотичних напружень по глибині (вісь z) значно перевищує дане положення для упорних (осьових) підшипників з коловим контактом при рівних умовах дослідження, що опосередковано вказує на меншу втомну довговічність радіальних підшипників у зв’язку з нерівномірним розподіленням тиску в зоні локального контакту.

Ключові слова: Точковий контакт тертя, втомна довговічність, підшипники кочення, максимальні контактні напруження, максимальні дотичні напруження, мікрогеометрія, тертя спокою.