METHOD FOR CALCULATING THE CONTACT BETWEEN ROLLER END FACE AND RING RIB OF ROLLER BEARINX IN MULTI-BODY SIMULATIONS

TRACK OR CATEGORY

7I | Contact Mechanics I

AUTHORS AND INSTITUTIONS

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INTRODUCTION

In roller bearings, there is a large number of different contacts. Depending on the load situation, each of these contacts influences the frictional power loss, temperature development and thus the operating capacity of the bearing in different ways. Under high axial loads, the rollers are supported on the ribs, as it is the case for tapered or cylindrical roller bearings. Therefore, a precise calculation of this contact between the front face of the rolling element and the rib is important to design robust bearings and to reduce frictional losses. The state of the art for calculating point contacts in current rolling bearing calculation programs are the theories according to HERTZ [1] or REUSNER [2]. With these theories, only a few geometric pairings can be described sufficiently precisely. However, complex geometries of the contact partners, which are typically used in modern bearings to reduce friction between the rolling element face and the rib, can only be described in simplified form. This leads to different discrepancies in the calculation of deflections, contact surfaces and pressures compared to high-quality numerical calculations e.g. finite element analysis (FEM). The shape of the contact zone can vary widely, which affects the pressure distribution and the velocity field of the sliding movement. Since the frictional power is calculated from the solid and the fluid friction, it cannot be described with sufficient accuracy, too. A more precise calculation method for the point contact can be used to design more efficient bearings.

HOW THE CONTACT CALCULATION METHOD OPERATES

The aim is to reduce the frictional power loss of roller bearings, especially under axial load. The use of simulation tools is essential to efficiently perform precise optimization. To achieve this, two basic steps – simulation and testing – are necessary. Within this contribution, a part of the simulation and a plausibility check against HERTZian theory is presented.

Calculation of the contact point

Since the presented Pressure-in-Point contact (PinP) method is used in multi-body simulation tools, a coupling of the contact calculation with the global multi-body calculation is necessary. In a first step, the position of the individual bodies in relation to each other must be determined. The position of the bodies' coordinate system and the contact point is described in the global coordinate system, as described e.g. in [3]. Starting from the global coordinate system, the position of the contact point is transformed into each body coordinate system. As result we get the position and the normal vector of the surface in the contact point.

Generating local geometry

Analytical equations in a Cartesian coordinate system are used to describe the geometries of the two contact partners, the rolling element face and the rib. Most surfaces, which appear in roller bearings, can be represented by the three equations of cone, sphere and torus shown in equation (1), see [4].

$$\begin{aligned} &\text{cone:} && x^2 + y^2 - m * z^2 = 0 \\ &\text{sphere:} && x^2 + y^2 + z^2 = r^2 \\ &\text{torus:} && (x^2 + y^2 + z^2 + R^2 - r^2)^2 = 4R^2 * (x^2 + y^2) \end{aligned} \quad (1)$$

A contact plane is placed on and aligned to the geometry in the contact point using the previously determined spatial and normal vectors. Using the ray equation (2) of the ray tracing method, similar to [5], the contact plane is projected onto the geometry so that the geometry is described by a distance function to this plane.

$$\vec{s}(t) = \vec{q} + t * \vec{n} \tag{2}$$

This procedure is applied to both contact bodies. The contact plane touches the geometry, i.e. the distance in the contact point is zero. The two geometries are superimposed in the contact planes. The difference between the two distance functions form a combination of the two geometries, the substitutive geometry. In a further step, the substitutive geometry is superposed with a penetration value *d* to the contact plane, so that it is cut through the plane. The cutting surface thus gives the contact zone. The size of the contact zone depends on the penetration value. To describe the material properties, the contact plane is discretized by spring elements. These elements represented by a modulus of subgrade reaction, which describes the material reaction in a similar way to WINKLER [6]. For this the reduced YOUNG's modulus from equation (3) can be used, as described by POPOV [7].

$$\frac{1}{E^*} = \frac{1-\nu_1}{E_1} - \frac{1-\nu_2}{E_2} \tag{3}$$

Calculation of the contact force

Subsequently the contact force is calculated from the displacement of the springs. However, simplifying assumptions must be made: isotropic homogeneous material, purely elastic behavior, dry normal contact, ideally smooth. Under these conditions, the discrete contact force similar to POPOV [8] can be calculated with equation (4). A spring element of the discretization occupies a length Δ*l* for each direction. This is deformed by the displacement profile $u_z(x,y)$, which is the superposed substitutional geometry. With the material properties, described by the reduced Young's modulus E^*, a discrete reaction force $\Delta F_{z\,x/y}(x,y)$ is determined for each element.

$$\Delta F_{z_{x/y}}(x,y) = E^* * u_z(x,y) * \Delta l_{x/y} = E^* * [g(x,y) - d] * \Delta l_{x/y} \tag{4}$$

With (4) the discrete contact force $\Delta F_{z_{x,y}}(x,y) = \Delta F_{z_x}(x,y) * X(x) + \Delta F_{z_y}(x,y) * Y(y)$ on the discrete surface element $\Delta A = \Delta l_x * \Delta l_y$ is calculated. $X(x)$, $Y(y)$ are area parameters for both directions. In order to determine the contact force F_N, one must integrate over each line within the contact zone Ω. The condition $\Delta F_{z\,x,y}(x,y) = 0$ applies, if $u_z(x,y) \geq 0$. Thus, the contact force is calculated to:

$$F_N = \int_{\Omega} \Delta F_{z_{x,y}}(x,y)\,\mathrm{dxdy} = E^* \int_{\Omega} [g(x,y) - d] * [X(x) + Y(y)]\,\mathrm{dxdy} \tag{5}$$

The contact force F_N is needed to solve the equilibrium of forces. Therefore, the determination of the penetration is an iterative process. Due to the continuous adjustment of the penetration, the contact zone and the contact force change until equilibrium.

Calculation of the pressure distribution

Similar to POPOV [8], the discrete pressure at each element can be determined as in equation (6). The discrete contact force $\Delta F_{z\,x,y}(x,y)$ acts on the discrete surface Δ*A* and leads to a discrete pressure $\Delta p(x,y)$. Thus the pressure is independent from the surface area Δ*A* and only dependent on material properties *E** and the resulting penetration d_{res} from the equilibrium of forces of the substitutive geometry $g(x,y)$.

$$\Delta p(x,y) = \frac{\Delta F_{z_{x,y}}(x,y)}{\Delta A} = E^* * [g(x,y) - d_{res}] \tag{6}$$

This reflects the pressure distribution of the entire contact zone, which is necessary for the friction calculation. At the end, the pressure *p* is determined from the contact force F_N.

COMPARISON TO HERTZ'S THEORY

The plausibility of the results must be examined so that this method can be used for the friction calculation in rolling bearing simulations. For this check, first the method according to HERTZ is used as a reference, since it is already established and accepted. The contact situations – sphere onto plane and sphere onto torus, as shown in Figure 1 – are examined and compared to HERTZ. The used parameters are listed in Table *1*. Different curvatures of the sphere are considered and the contact force is increased. The maximum pressure, the size of the contact zone and the penetration are evaluated.

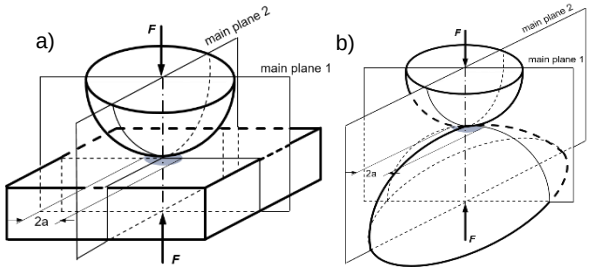

Figure 1: Investigated contact situations
a) sphere onto plane b) sphere onto torus

Table 1: Used parameters for the comparison with HERTZ's theory; left: sphere on plane; right: sphere torus

	unit	sphere	plane	sphere	torus	
radius	mm	10 – 1 000	∞	10 – 100	20	30
YOUNG's modulus	MPa	210 000	210 000	210 000	210 000	
POISSON's ratio		0.3	0.3	0.3	0.3	
contact force	N	100 – 500		100 – 500		

Figure 2 shows the results of the two contact situations relative to HERTZ. It can be seen that the maximum pressure of the PinP method corresponds very well to HERTZ. The same can be seen in the size of the contact zone and the penetration. The values of the PinP method range between ± 5 %. Out of this, the PinP method provides plausible results compared to HERTZ's theory.

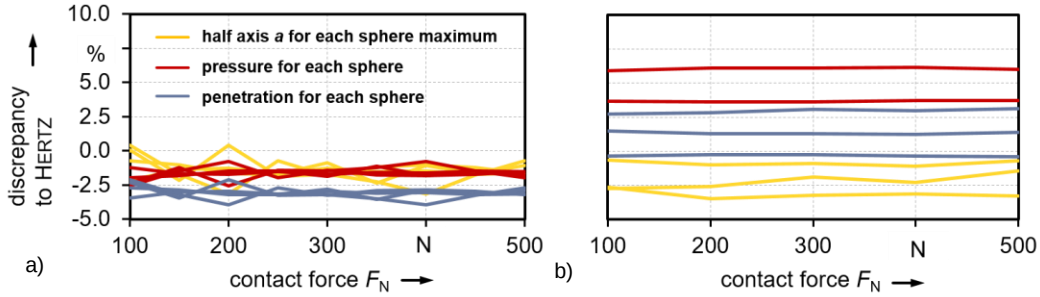

Figure 2: relative comparison of PinP to HERTZ of the area, pressure and penetration
a) sphere onto plane b) sphere onto torus

SUMMARY AND OUTLOOK

To summarize, the PinP method provides similarly accurate results as the theory of HERTZ. But in addition, the PinP method can be used to describe more complex geometries that generate curved ellipses as contact zones. These can only be described simplified by HERTZ. Instead, contact situations of two contact zones are not. This is the great benefit of the PinP method, which is suitable for such constellations. In a further step, this has to be examined more precisely. For this purpose, higher numerical methods, such as FEM, will be used for a comparison.

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KEYWORDS

Contacts: Contact Mechanics
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