A METHOD FOR DETERMINING THE ROLLING FRICTION TORQUE IN A THRUST BEARING. PART I

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Abstract: The paper presents a method and device intended to determine the rolling friction torque in a thrust ball bearing. The methodology is based on the fact that, in the case of a thrust ball bearing the cage maintains the balls axially equidistant and thus, the rolling components of the friction moment from the cage-ball contact points generate a vector system equivalent to zero. Based on kinematical aspects it is shown that both spinning and rolling components of the friction torque are simultaneously present in a running thrust ball bearing.

Keywords: *rolling friction moment, thrust ball-bearing, kinematical analysis*

1. Introduction

Two solid bodies can interact either directly (by direct contact) or indirectly (through fields). Concerning mechanical engineering applications, the first type is the more widespread although lately applications from the second category are more and more common (magnetic bearings, electrical shafts etc.).

Regarding the direct contact, there are two distinct cases: the boundary surfaces of the two bodies are identical and when a body is fixed and the other one is mobile, the motion will be identical the one of the mobile body obtained when the roles are switched. This contact type corresponds to lower pairs where the contact between bodies is made on large regions between surfaces of the same type.

The classic cases to be mentioned are: the spherical pair, Fig. 1, the planar pair, Fig. 2 and the cylindrical pair, Fig. 3, with particular cases: prismatic pair and revolute pair. When the contact is made between surfaces in a manner that the motion of a body, when the other is kept immobile, differs from the motion of the other body when the parts reverse, the pair is named higher pair or irreversible, as presented in Fig. 4.

The frictionless contact between a plane and a cylinder is considered to exemplify the above.

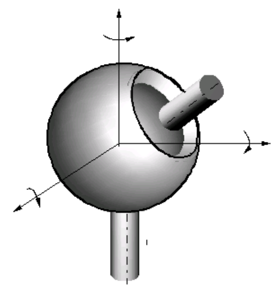

Figure 1. Spherical pair

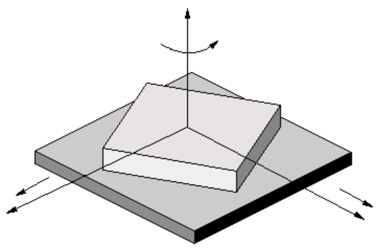

Figure 2. Planar pair

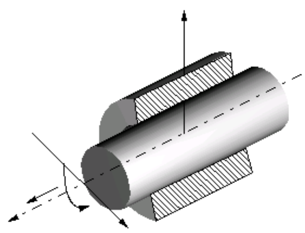

Figure 3. Cylindrical pair

The motion can be studied in any plane normal to the generatrix of the contact, as in Fig. 4. The nearest frontal plane is chosen as the study plane. In this plane, a circle and a straight line are contacting in point C. If the straight line is fixed, the C point attached to the mobile circle describes a cycloidal arc. When the circle is maintained immobile, the C point attached to the straight line, describes involute arcs [1].

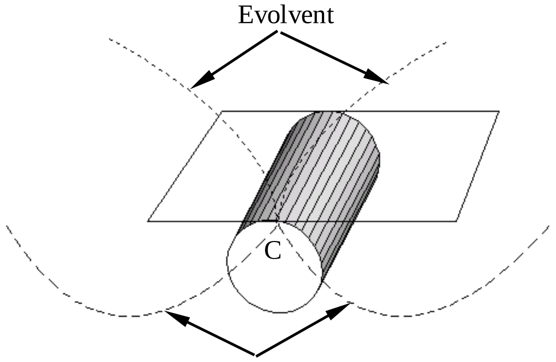

Figure 4 Irreversibility of cylinder-plane contact

From the examples above, it can be concluded that for the higher pair the relative motion between the two contacting elements is more complex than for the lower pair case. The most encompassing situation refers to the case of point contact between the two bodies.

2. The torsor of friction in a concentrated contact

The point contact between two bodies, denoted C is considered. The normal $\boldsymbol{n}$ and the tangent plane $\boldsymbol{t}$ are well defined in this point. The relative motion between the two bodies is characterized by the sliding velocity $\boldsymbol{v}$ from the tangent plane and the angular relative velocity ω. The component parallel to the normal ω_s is the angular spinning velocity:

$$\omega_s = \boldsymbol{n}\boldsymbol{n}^T \omega \tag{1}$$

and the component from the tangent plane is the angular rolling velocity, expressed by the relation:

$$\omega_r = (\boldsymbol{I}_3 - \boldsymbol{n}\boldsymbol{n}^T)\omega \tag{2}$$

where $\boldsymbol{I}_3$ is the unit matrix of third order. To each of the three components of the relative motion defined above, a force or a torque will oppose. Therefore, to the sliding velocity, the friction force $\boldsymbol{T}$ parallel and of opposite sense to the velocity $\boldsymbol{v}$ will resist, to the spinning angular velocity ω_s the spinning friction torque $\boldsymbol{M}_s$ parallel to the normal $\boldsymbol{n}$ will oppose, while to the rolling angular velocity ω_r a rolling friction moment from the tangent plane will resist. On the normal direction, the interaction between the two bodies is described by the normal reaction $\boldsymbol{N}$.

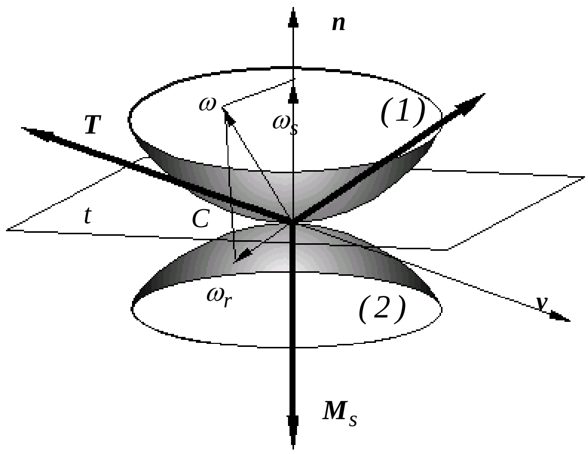

Figure 5. The relative motions and the torsor of friction in a point contact

Regarding the values of the components of the torsor of friction: for the friction force in the case when between the points C_1 and C_2 which superpose in contact there is no relative motion, explicitly:

$$\boldsymbol{v} = \boldsymbol{v}_{C_1C_2} = \boldsymbol{0} \tag{3}$$

It is said that pure rolling takes place between the two bodies. In this case, the magnitude of the friction force is undetermined and represents one of the unknowns of the problem. When the relation (3) is not satisfied, sliding motion occurs between the points $\boldsymbol{C}_1$ and $\boldsymbol{C}_2$ and the value of the friction force $\boldsymbol{T}$ is expressed as function of the magnitude of the normal reaction according to Amontons-Coulomb law:

$$T = \mu N \quad (4)$$

where μ is the dynamic coefficient of sliding friction. To be remarked that fro the rolling friction case the following inequality:

$$T < \mu N \quad (5)$$

must be permanently verified.

Concerning the magnitudes of the two friction moments, both the spinning and the rolling friction torques are expressed as depending on the magnitude of the normal force:

$$M_s = M_s(\boldsymbol{N}) \quad (6)$$

and

$$M_r = M_r(N) \quad (7)$$

In technical literature one can found expressions for the dependencies (6) and (7). The simplest forms [2] accept a linear dependency between the magnitudes of the friction torques and the magnitude of the normal force, specifically

$$M_s = s_s N \quad (8)$$

and

$$M_r = s_r N \quad (9)$$

where s_s and s_r are coefficients with dimension of length and are named coefficient of spinning friction and coefficient of rolling friction, respectively.

Using the theory of elasticity one can conclude that the dependencies (6) and (7) are power law functions.

3. Kinematics of relative motion for the thrust ball bearing

The axial section of a thrust ball bearing is presented in Fig. 6. The relative motion of the ring is a relative rotation about its axis performed with the angular velocity ω. In order to study the kinematics of the relative motion of the ball bearing the next artifice is made: the relative motion of any of the two rings is considered as the resultant of two relative motions with respect to the plane of symmetry of the bearing normal to the axis of rotation. With respect to this plane each of the two rings performs a rotation motion of angular velocity $-\omega/2$ and $\omega/2$ respectively. Additionally, the motion of the ball can be regarded as a rotation about a fixed axis contained in an horizontal plane. Thus, all the elements of the bearing execute rotation motion about fixed axes.

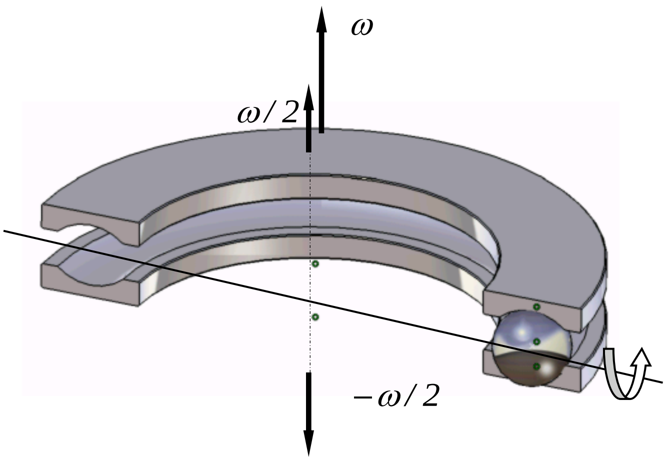

Figure 6 Axial section through a thrust ball bearing

In Fig. 7 is represented an axial section through the thrust ball bearing evidencing the angular velocities together to a side view required for highlighting the motion of the contact points between the ball and the race. The rings were denoted with 1 and 2 and the ball with 0. The fact that the ball has fixed rotation axis explains that the points C_1 and C_2 have the same velocity.

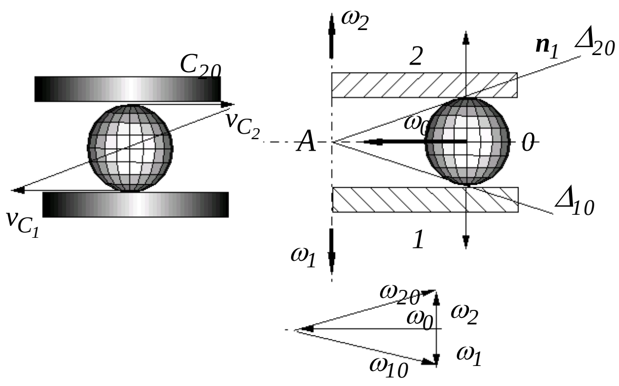

Figure 7. Distribution of velocities in a thrust ball bearing

The intersection point between the axis of rotation of the ball and the axis of rotation of the rings is immobile. In the case when pure rolling exists between the balls and races it results that the straight lines passing through the points A and the contact point, C_1 and C_2 respectively, define the *instantaneous axes of relative motion between the balls and the races*. The relative velocities ω_{10} and ω_{20} are parallel to the lines Δ_{10} and Δ_{20}, as it can also be noticed from the polygon of angular velocities.

For the thrust ball bearing case, the normal in the contact point will always be parallel to the axis of the bearing. The relative angular velocity ball-race presents components both along the normal and in the tangent plane and this fact conducts to the conclusion that *always in a thrust bearing there will be simultaneously present both rolling friction torque and spinning friction torque.*

From experimental point of view, this situation is complex when the estimation of the spinning friction moment is aimed. The quickest method consists in considering separately a single ball and a bearing ring and determining the spinning friction [3], [4].

The shortcoming of the method resides in the fact that the ball-race spinning motion must be obtained using a kinematical chain where the frictions have to be insignificant compared to the ones from the considered contact.

In [5], [6] it is proved on experimental approach that when a lubricant is present in a ball-spherical cavity contact, the ball passes from a spatial motion regime to a regime of rotation motion about a fixed axis. The attempt to apply the same technique for the case of dry contact showed that the ball always maintains the spatial motion regime, fact theoretically demonstrated.

There is a complete different situation in the case of ball-toroidal groove contact. In these circumstances, a body obtained from a ball with attached fly-wheel, first has a spatial motion followed next by a rotation motion around a fixed vertical axis.

Based on this remark, the second part of the paper presents a proposed method for determining the moment of rolling friction in a thrust bearing, starting from the statement that, when the balls are placed angular equidistantly on the race, the rolling friction moments have the resultant torque equal to zero.

4. Conclusions

In the first part of the paper, the categories of contacts that can be met in technical applications are revealed.

For the concentrated contacts, there are next identified the possible relative motions with respect to an intrinsic coordinate system - defined by the common normal and the tangent plane in the contact point, and afterwards the components of the torsor of friction are found.

For the kinematical study, an artifice was applied to allow for transforming the ball-bearing into a system of bodies with immobile axes of rotation.

The kinematical analysis conducts to the conclusion that in a thrust ball bearing there always will be present in the ball-race contact points, moth spinning friction moment and rolling friction moment. This fact conducts to difficulties when the experimental finding of the friction moments in a thrust ball-bearing is intended.

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