Internal loading distribution in statically loaded ball bearings

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Ball and roller bearings, generically called *rolling bearings*, are commonly used machine elements. They are employed to permit rotary motions of, or about, shafts in simple commercial devices such as bicycles, roller skates, and electric motors. They are also used in complex engineering mechanisms such as aircraft gas turbines, rolling mils, dental drills, gyroscopes, and power transmissions.

The standardized forms of ball or roller bearings permit rotary motion between two machine elements and always include a complement of ball or rollers that maintain the shaft and a usually stationary supporting structure, frequently called a *housing*, in a radially or axially spaced-apart relationship. Usually, a bearing may be obtained as a unit, which includes two steel rings each of which has a hardened raceway on which hardened balls or rollers roll. The balls or rollers, also called *rolling elements*, are usually held in an angularly spaced relationship by a *cage*, also called a *separator* or *retainer*.

There are many different kinds of rolling bearings. This work deals with internal loading distribution in statically loaded *single-row angular-contact ball bearings*, which are designed to support combined radial and thrust loads or heavy thrust loads depending on the contact angle magnitude. Several researchers have studied the subject, as for example, [1] to [4]. The methods developed by them to calculate distribution of load among the balls and rollers of rolling bearings can be used in most bearing applications because rotational speeds are usually slow to moderate. Under these speed conditions, the effects of rolling element centrifugal forces and gyroscopic moments are negligible. At high speeds of rotation these body forces become significant, tending to alter contact angles and clearance. Thus, they can affect the static load distribution to a great extension.

Harris [5] described methods for internal loading distribution in statically loaded bearings addressing pure radial; pure thrust (centric and eccentric loads); combined radial and thrust load, which uses radial and thrust integrals introduced by Sjovāll [2]; and for ball bearings under combined radial, thrust, and moment load, initially due to Jones [3].

We can observe that there are many works describing the parameters variation models under static loads but few show such variations in practice, even under simple static loadings. The author believes that the lack of practical examples is mainly due to the inherent difficulties of the numerical procedures that, in general, deal with the resolution of several non-linear algebraic equations that must to be solved simultaneously.

In an attempt to cover this gap studies are being developed in parallel [6, 7]. Particularly in this work is described a new, precise method for internal load distribution computation in statically loaded, single-row, angular-contact ball bearings subjected to a known external combined radial and thrust load. The novelty of the method is in the choice of the set of the nonlinear equations, which must be solved simultaneously. The author didn't found in the literature the resolution of this problem using the same set of equations, which are

$$\delta_a - \delta_r \tan\beta_j \cos\psi_j - A\frac{\sin\left(\beta_j - \beta_f\right)}{\cos\beta_j} = 0, \qquad j = 1,\ldots, Z, \tag{1}$$

$$F_a - \sum_{j=1}^{Z} K_{nj} \sin\beta_j \left(A\left(\frac{\cos\beta_f}{\cos\beta_j} - 1 \right) + \frac{\delta_r \cos\psi_j}{\cos\beta_j} \right)^{3/2} = 0, \tag{2}$$

$$F_r - \sum_{j=1}^{Z} K_{nj} \cos\psi_j \cos\beta_j \left(A\left(\frac{\cos\beta_f}{\cos\beta_j} - 1 \right) + \frac{\delta_r \cos\psi_j}{\cos\beta_j} \right)^{3/2} = 0, \tag{3}$$

where δ_a and δ_r are the relative axial and radial displacements between the inner and outer ring raceways; β_f and β_j are the free-contact angle and the j-th ball contact angle (located at ψ_j angular position), respectively; A is the distance between raceway groove curvature centers for the unloaded bearing; K_{nj} are load-deflection factors; F_a and F_r are external thrust and radial loads, and Z is the number of balls.

Equations (1) to (3) are $Z + 2$ simultaneous nonlinear equations with unknowns δ_a, δ_r, and β_j, $j = 1,\ldots, Z$. Since K_{nj} are functions of final contact angle, β_j, the equations must be solved iteratively to yield an exact solution for δ_a, δ_r and β_j.

The Figure 1 shows the normal ball applied load as a function of the thrust load, F_a.

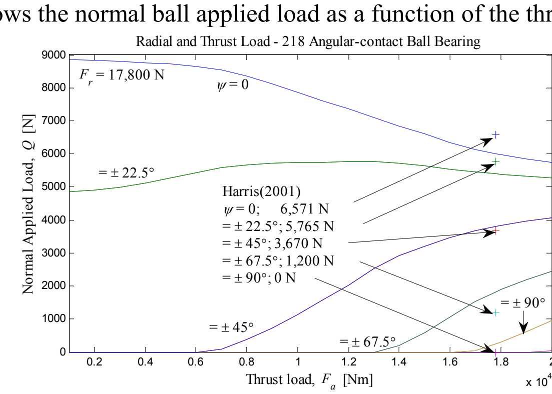

Figure 1: Normal ball applied load as a function of the thrust load, F_a, for a 17,800 N radial load.

References

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