BRICATION TECHNOLOGY, TRANS. ASME, Series F. Vol. 96, No. 3, 1974, pp 391–397.

25   Zaretsky, E. V., Anderson, W. J., and Parker, R. J., "The Effect of Contact Angle on Rolling-Contact Fatigue and Bearing Load Capability," *Trans ASLE*, Vol. 5, 1962, pp 210–219.

26   McCall, J. C., Renewal Theory-Predicting Product Failure and Replacement, *Machine Design*, Vol. 48, No. 7, Mar. 25, 1976, pp 149 154.

Analysis of Tapered Roller Bearings Considering High Speed and Combined Loading[1]

C. A. Moyer.[2] Dr. Liu has presented an interesting analysis of a single tapered roller bearing attempting to show the effects of misalignment and speed on such an isolated, single bearing. It appears because of some of the beginning assumptions Dr. Liu uses, however, that the calculations made and the results presented may not provide realistic insight into the actual performance of such a bearing. Several assumptions are in question.

First, the author states that the raceway and roller is assumed in line contact although a significant crown radius of 2680mm (105.5 inches) is given for the roller body. The load considered is sufficiently low that the elliptical contact would be just truncated for these conditions. Therefore, point contact deformation should be considered at both the inner and outer raceway contact.

Second, the general statement is made that the total load at roller and outer raceway contact is independent of speed. This can only be true if sufficient thrust load is being applied to the bearing so that centrifugal force effects are overcome. Also, whether Figs. 9 and 10 are correct in the author's paper depend on the order of application of the thrust load and centrifugal force.

In considering load and speed changes, does the author consider that the total axial deformations remain constant, that is, does the distance in the 'Z' direction in the author's Figure 1 between the cone backface to cup backface change or remain the same as speed increases centrifugal force and external applied loads stay constant? Such problems in interpreting the single bearing internal conditions illustrate the difficulty in relating the single bearing analysis to real life situations.

Third, the author implies that a high speed bearing can operate with sufficient misalignment that the rollers will actually lift off the raceway allowing *no* contact over about 30 40 percent of the raceway (Fig. 8). In a realistic application, either the bearing would need to be redesigned or the application modified in order to maximize expected performance.

The author mentions the sliding at the inner ring flange or cone rib and calculates only the sliding performance. In terms of rib roller end performance, this may be misleading since both the rib surface and roller end surface are moving and the mean surface velocity $[\frac{1}{2}(u_1 + u_2)]$ is about $2\frac{1}{2}$ times the sliding velocity $[u_1 - u_2]$ for this bearing. For proper evaluation, it is better to include both, perhaps in terms of the slide/roll ratio.

[2] Assistant Chief Engineer, Physical Laboratories, The Timken Co., Canton, Ohio.

Author's Closure

The author welcomes the discussion by Mr. Moyer, who has certainly brought up some valuable and interesting points on the subject paper.

The roller and raceway are designated to be in line contact because the so-called slicing technique is used to define the load-deformation relationship. The slicing technique considers a contact to be composed of a number of slices of constant width, each of which is taken as a line contact. According to reference [1],[3] it can accurately predict the contact characteristics for either line or point contact, provided that the contact length to width ratio is large and/or the stress level is high. These requirements, of course, cannot be strictly fulfilled in the numerical example of the paper, where, when misalignment is present, a roller raceway contact may neither be classified as line contact nor as point contact. It is probable that the numerical accuracy can be improved if a larger number of slices was chosen for each contact at the expense of more computer time.

The author cannot follow how the roller centrifugal force can be overcome by sufficient thrust load since the centrifugal force is needed in the roller equilibrium conditions. Nor can he understand the effect of the order of application of the thrust load and centrifugal force on the reliability of Figs. 9 and 10 in the paper, since the paper considers only the dynamic equilibrium of the bearing. The reasons that the total contact load at the roller and outer raceway contact is independent of speed are: (1) the outer ring is assumed to be rigid, and (2) the sum of the horizontal components of the outer raceway contact loads must balance the applied thrust load. However, as pointed out by Mr. Parker in his discussion on the paper, when a pair of tapered roller bearings are axially set up with a given clearance, an induced thrust load due to speed will come into being in addition to the applied thrust load. This induced thrust load will then increase the total load at the outer contact.

The bearing inner ring displacement in the Z-direction will change with load and/or speed. In the analysis of the paper, all the inner ring displacements will change as the bearing operating conditions change, unless some restrictions be applied to them. If the displacement in a certain direction is specified, then the number of bearing equilibrium equations will be reduced by one and the bearing load distributions will also be different. For instance, in order to avoid that 30–40% of the roller be out of contact with the outer raceway at $\psi = 0$ degrees as shown in Fig. 8 of the paper, one must be sure that the misalignment angle, γ_y, be kept very small, probably less than 0.1 degrees.

It is true that the slide/roll ratio at the flange contact is meaningful as it has a direct relationship with the EHD friction coefficient [2]. However, the heat generation and smearing occurrence are associated with, among other things, sliding. In the analysis, it was assumed that pure rolling exists between the roller and the raceways. Therefore, the exposition of the flange sliding will also serve to disclose the kinematical difference between the roller flange contact and the roller raceway contacts.

Additional References

1   Crecelius, W. J., Jr., "Evaluation of a Slicing Technique Used to Calculate the Characteristics of Concentrated Contacts," MS thesis, Penn. State University, 1975.

2   McCool, J. I., and Chiu, Y. P., et al., "Influence of EHD lubrication on the Life and Operation of Turbine Engine Ball Bearings," Air Force Contract No. F33615-72-C-1467, July 1974.

[3] Numbers in brackets designate Additional References at end of closure.