MEASUREMENT OF TORQUE ON TAPERED ROLLER BEARINGS

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SUMMARY
There are several parameters, which need to be either measured or calculated in order to quantify the performance of a rolling element bearing. Established tests or techniques, which are used, and these have been developed over the years to form the current standards. ISO standards are available for determining the static and dynamic loading capabilities. However, even though friction in taper roller bearings has been studied [1, 2, 3], and formulae derived for the calculation of friction levels, there are few instances where the friction is measured on bearings subjected to both axial and radial loads. Comparison of the level of friction within taper roller bearings has been previously undertaken. Usually the bearing is run under load whilst measuring the temperature of the bearings at a point on the outer raceway. Since the friction within the bearing represents energy loss and heat generation, the bearing, which operates at the lowest level temperature, is said to have the lowest level of friction. This is a relatively simple test but only an approximate comparison between the bearings is possible. A numerical value for the torque required overcoming friction to enable it to be compared to calculated values.

Keywords: Tapered roller bearings, torque, friction.

1 INTRODUCTION
This paper shows designs of bearing test rigs and their shortfalls before showing a new design of rig, which may be used to measure the friction within a 80mm diameter taper roller bearing over a wide range of speeds with either grease or oil lubrication. These results are to then be used as a benchmark to compare against self-tracking bearing of similar size.

2 PREVIOUS DESIGNS OF TEST RIGS
There have been several designs of equipment to test the levels of friction in taper roller bearings. None of these previous designs were suitable with the main problem being the method, which the radial load was applied. This load is usually applied using a slave bearing which affects the running conditions of the rig and also a source of parasitic friction. Figure 1 shows a rig in which the radial load is applied via a slave bearing.

The rig shown above uses a spring to apply the radial load to the shaft. This load is applied via two deep groove ball bearings, which are situated at a point on the shaft, mid distance between the test bearings. The slave bearings lead to several fundamental flaws, which would affect the rig if it were to be used to measure the torque required to overcome friction.

For instance, heat is applied to the shaft from the friction, which is present in the slave bearings, which in turn affects the running conditions of the test bearings. Therefore it is difficult to derive the normal running temperature of the test bearings.

![Figure 1: Standard contemporary rig](image1)

![Figure 2: The centre slave bearing assembly](image2)

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3 PROPOSED RIG

To overcome the problems described above, it was decided to make the slave bearings of the same type as the test bearings, in this case taper roller bearings. These slave bearings would be subjected to the same axial and radial loads and therefore the rig would test four bearings simultaneously. The centre two bearings would be fitted in a back-to-back arrangement and the two outermost bearings would be arranged face-to-face. This is shown in Figure 2.

Axial movement would be required so that one of the centre bearings (3) could move in relation to the other centre bearing (2) and that the whole centre bearing assembly could also move in relation to the end bearings. Finally, one of the end bearings (1) would also need to be able to move in the axial direction. Thermal expansion of the shaft would not affect the axial loading since only one bearing (4), which is situated at the end of the shaft, is located axially. All bearings are held so that they cannot rotate but are always retained both concentric, and square, to each other.

The shaft is a one piece unit with bearings (2) and (3) being pressed against a raised shoulder in the centre of the shaft. Bearings (1) and (4) are fitted to the ends of the shaft, using suitable spacers (7) to ensure they are located in the correct position and are retained with large diameter lock nuts.

Loads are again applied by the use of springs, with the radial load being applied with compression springs (6), as is usual. Tension springs (5) are fitted, equally spaced around the shaft to apply the axial loads to the bearings. The system of using a variable speed drive together with a torque transducer allows a torque curve to be attained for the bearings under set conditions.

Since the only bearings used in this design are the test bearings themselves and the loads are equal throughout, then the maximum speed is not limited by the slave bearings. The drive is direct, through torsionally stiff couplings and the speed is controlled by a three-phase, inverter (12). Both the torque transducer (11) and the motor (13) both have speed capabilities higher than the test bearings. A speed range of 500 - 6000 RPM is available. The minimum speed is limited by the heat produced by the motor but this may be reduced further with the application of additional cooling. It is the frequency and the reduction in torque caused by the three-phase, inverter, which limits the top speed. The use of the inverter enables an increase in the speed gradually until the temperature of the bearings becomes excessive. Direct drive also removes any radial loading due to the tension within vee belts.

4 ASSEMBLY

Assembly is straightforward, with the cups of the centre bearing being fitted first. Then the cones of bearings (1) and (2) are fitted to the shaft, together with the spacers and lock nuts. The shaft is then fed through the centre housing and cones of bearings (3) and (4) are fitted. Cups of the bearings (1) and (4) are pressed into the fixed (8) and floating housings (10) respectively. The shaft, together with the centre bearing assembly is passed through the L.H. support (9) and supported until the floating housing is fitted. The drive and all other sundry equipment may then be installed.

5 THEORETICAL PREDICTIONS

The simplest method for the prediction of torque has been derived by Palmgren. The following formulae are used [4].

\[ M_I = f_j F_\beta d_m \]  \hspace{1cm} (1)

\[ M_v = 10^{-7} \times f_v (V_v n)^2 \frac{d_m}{3} \]  \hspace{1cm} \text{if } V_v n > 2000  \hspace{1cm} (2)

\[ M_v = 160 \times 10^{-9} \times f_v d_m^3 \]  \hspace{1cm} \text{if } V_v n \leq 2000  \hspace{1cm} (3)

\[ M_f = f_f F_v d_m \]  \hspace{1cm} (4)

\[ M = M_I + M_v + M_f \]  \hspace{1cm} (5)

Figure 3: The four bearing test rig
The last equation above shows that the total torque is the sum of the torque due to the load, the viscous torque and roller end flange friction, which are calculated from the other equations. These equations are not specific to taper roller bearings since they use constants which have been derived empirically for most types of rolling element bearing.

A more comprehensive solution, which takes elasto-hydrodynamic lubrication into account has been developed by Aihara [1]. This theory showed that the total torque ($M$), is the sum of the torque due to rolling resistance ($M_R$), and the torque due to sliding ($M_S$). Two terms for the rolling friction need to be calculated, one for the inner and another for the outer contact.

To calculate these frictional forces, the minimum lubrication film thickness must first be calculated since this affects the contact between the roller end and the rib. Using the formula developed by Hamrock and Dowson [5].

$$h_{min} = 3.63 U^{0.68} G^{0.49} W^{-0.073} (1 - e^{-0.684}) R_x \tag{6}$$

where

$U$ is a dimensionless speed parameter

$G$ a material constant

$W$ is a dimensionless load parameter

$R_x$ is the equivalent radius in the rolling direction

$$k = 1.03 \left( \frac{R_e}{R_x} \right)^{0.64} \tag{7}$$

The torque due to sliding reduces as the speed increases due to the elastohydrodynamic film thickness increasing.

Frictional moments at the inner and outer raceways are calculated for each of the rolling elements separately using

$$M_{i,o} = \left[ \left( \frac{1.76 \times 10^3}{1 + 0.29 L^{0.76}} \right) \frac{1}{\alpha_o} \left( \frac{GU}{0.685} R^{0.51} \right) \right]_{i,o} \tag{8}$$

Finally, the torque for the full bearing is found by finding the sum of the torque for the inner and outer contacts for each roller and the torque due to the roller end to rib contact. The equation for this is

$$M = \frac{z}{D_a} (R_e M_i + R_i M_o) + e \mu_o \cos \beta F_o \exp(-1.8 \Lambda_{e,12}) \tag{9}$$

Figure 4 shows the torque predicted in a 80 mm bore taper roller bearing which has been loaded in both the axial and radial direction, lubricated by oil. The graph clearly shows a discrepancy between the two methods, especially at the lower speeds.

Figure 5 shows the same bearing as the ratio of axial, to radial, $(F_a)/(F_r)$ load changes. It is clearly shown that the torque required to overcome friction at low speeds is high. This reduces as the speed increases to a point where there is a sufficiently thick lubrication film to fully separate the contacts. The rate of this initial drop off is greatly affected by the ratio between the axial and radial loads.

Figure 6 shows the predicted running torque plotted against axial load. The new design of rig has provisions so that both the axial and radial loads can be adjusted whilst the test is running. This gives the rig greater flexibility to the tests that it can undertake.
6 CONCLUSION

For the project at the University of Bristol, 80 mm taper roller bearings were used due to availability of this size of bearing from several manufacturers. This study is part of a larger project for which 80 mm roller bearings, of various types, are used. Nevertheless, they are representative of bearings over a wide range of sizes.

A new method of testing was necessary since the previous test rig was incapable of producing a numerical value for torque. Both the radial and axial loads are applied independent of each other and the ratio between these loads may also be varied. The speed may also be easily changed over a range from 500 RPM to the limiting speed of the test bearings. Therefore the new rig can be set to represent industrial situations, for instance a gearbox assembly, where taper roller bearings are often used.

7 REFERENCES