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SIMULATION BASED DESIGN OF A HIGH-SPEED TAPERED ROLLER BEARING

TRACK OR CATEGORY

Rolling Element Bearings

AUTHORS AND INSTITUTIONS

Alexander Hassis, Christian Brecher, Marcel Fey
Chair of Machine Tools at WZL at RWTH Aachen University

INTRODUCTION

Tapered roller bearings (TRB) show a far higher stiffness and load rating than high-speed angular contact ball bearings, so called spindle bearings (SPB). Thus, TRBs can be a favorable alternative to SPBs in main spindles for applications with low to medium spindle speeds at high spindle loads as they occur in high performance cutting (HPC) or heavy roughing applications. The use of TRBs in main spindles would allow new spindle designs like high load spindles or more compact spindles. However, the very limited speed rating of TRBs makes them unsuitable for most of the above mentioned applications. The design of an all new high-speed TRB requires taking into account not only the desired properties of the bearing but also limitations given by the available methods to manufacture it and the restrictions resulting from its targeted application. The design approach described here is divided in three major steps: First, general requirements, restrictions and given conditions are defined. Second, an optimization problem for an interior point method optimization algorithm is developed and solved to obtain the bearing's macro geometry. Third, a newly developed bearing calculation software is used to define the micro geometry.

GENERAL REQUIREMENTS FOR THE NEW DESIGN

By focusing on main spindles as a possible application for the newly developed high-speed TRB one major source of objective properties and limitations is given. Many other design limitations are given by the manufacturing processes available for the bearing.

SPBs with large steel balls and a contact angle of 25° degrees feature the highest axial stiffness (for a given preload class) and the lowest speed rating of all common SPBs. The new TRB is designed to be suitable to replace such an SPB. Thus, the overall outer dimensions of a 70 mm x 110 mm x 20 mm (inner, outer diameter, width) SPB are chosen as the main dimensional limitations. To give an advantage over the SPB, one main demand is an axial stiffness twice that of the SPB to allow the replacement of two SPBs with one TRB. The main objective of the optimization is a high speed rating to give an advantage over standard TRBs. The difficulties to address this demand in a manner suitable for the formulation of an optimization problem are discussed in the subsequent paragraphs.

In previous research efforts at WZL a fist prototype of a high-speed TRB was developed and successfully tested [1]. Although the design process was not based on calculations but on rules of thumb, expert knowledge and an overall design similar to that of a standard 32014 X TRB some properties of the new bearings were derived from the first prototype since they prove to be beneficial: The bearing has silicon nitride (Si_3N_4) rollers to reduce the mass induced forces and improve the friction coefficient in the roller-ring contacts. The rings are made from X 30 CrMoN 15 1 (AMS 5898) since it is the standard ring material of the chosen bearing supplier. One of the most important differences to a standard TRB is the location of the rib. Here, the rib is located on the outer ring. Whereas this has some disadvantages regarding load distribution in the bearing and slip conditions in the roller-rib contact it is very favorable for the heat dissipation from the roller-rib contact. Another main advantage is the possibility to direct lubricant through the rib directly in the roller-rib contact. Since the roller-rib contact is expected to be crucial for the high-speed performance of a TRB, this change of the rib location is seen as key feature of the new bearing.

The rings of the bearing are made by hard turning without any additional surface finishing. Due to the existing manufacturing processes at the supplier's, the ring raceways and rib have to be unprofiled thus being sections of ideal cones. The rollers are made from cylindrical semi finished parts by grinding with an added surface finishing by vibratory grinding.

FORMULATION AND SOLUTION OF AN OPTIMIZATION PROBLEM FOR THE MACRO GEOMETRY

To define an optimization problem, one or more objective functions, objective variables, design variables and restrictions have to be defined. To determine the macro geometry of the TRB the design variables chosen are the contact angle α, the pitch diameter d_m, the mean roller diameter D_m, the included roller angle γ, the roller length l_R and the number of rollers Z. Besides the restrictions given by the overall outer geometry and the limitations due to the manufacturing process an additional restriction can be derived from the demand that the roller cone and the two raceway cones intersect at one point on the bearing axis. Two more restrictions are derived from the above mentioned minimum stiffness requirement and the demand for an upper limit of the maximum contact pressure in the roller raceway contacts.

However, there is no single variable being suitable to represent the objective of the optimization which is the speed rating. The maximum permissible speed of a bearing depends on many interdependent factors like the bearing's temperature in a given application, the tribological conditions in the rolling element-ring contacts or mechanical limitations. Since none of these speed limits is suitable for a generalized calculation in a simple, algorithm based optimization, a different approach is chosen: It is assumed that friction in the roller-rib contact is the most crucial factor limiting the permissible bearing rotational speed. Besides the thermal and tribological conditions that are not easily accessible for calculations, the roller-rib friction is governed by the contact normal force and the relative speeds there. Since only the macro geometry is considered in this design step, the micro geometry is assumed to be the same for every design. Thus, higher contact normal forces will result in higher friction as will higher relative speeds. Higher friction however is assumed to result in higher temperatures and hence in lower permissible speeds. Thus, the contact normal force and the relative speed are chosen to be a measure for the roller-rib friction and hence as the objective variables to be minimized.

To define an optimization problem accessible for established solving methods, equations have to be defined to represent the objective variables and the restrictions. Whereas the geometrical restrictions give simple equations, the bearing stiffness can generally not be calculated explicitly. It is therefore assumed that the TRB and the SPB show a comparable qualitative load-displacement relationship. Thus, the demand for a minimum stiffness at the maximum allowable load is represented by a maximum allowable inner ring displacement at this load. To calculate this displacement, the roller-raceway contacts at both rings are assumed to be perfect cylindrical contacts. With the load-deformation relationship for line contacts according to Tripp [2] the roller-inner ring raceway and roller-outer ring raceway deformations are calculated. Adding these deformations under consideration of the respective contact angles gives an approximate value for the overall inner ring displacement at a given load. Perfect line contact between two contacting cylinders is also assumed to calculate the maximum contact pressure in the roller-raceway contacts according to Hertz [3] giving two more equations to represent restrictions.

An equation to approximate the roller-rib contact normal force can be derived from geometrical considerations taking into account the centrifugal forces acting on the rollers. The relative speed in the roller-rib contact point depends on the actual geometric conditions (i. e. the micro geometry). Generally, it increases linearly with the distance of the contact point from the transition point between the raceway and the rib where it is zero. Thus, for any possible contact point location on the rib which is assumed to be the same for every macro geometry, the relative speed there is lower when the increase of the relative speed over the rib height (i. e. the slope of the speed-over-contact-point-location-function) is lower. This allows for a simple equation representing the relative speed independently from the absolute speed or the micro geometry.

Since optimization problems with two or more objective functions are more difficult to solve than those with only one objective function, the two objective functions representing the load and speed in the roller-rib contact are combined in one objective function. Both functions are multiplied with a weighing factor, one function is additionally multiplied with a scaling factor and the scaled and weighed functions are added. The weighing factors allow for giving more weight to that objective function that is considered to be more important for the result. The scaling factor is necessary since the sum of the two objective functions is a numerical equation where one of the summands (i. e. one objective function) will overweigh the other if their numeric results do not have the same order of magnitude. This is the case here where the order of magnitude of the speed slope is about one or two orders lower than that of the force.

The composed objective function and the set of restrictions are solved simultaneously using the MATLAB® integrated interior point method in combination with a global search method. While the scaling factor is kept constant, the weighing factor is varied to show the influence of a larger weight of the load or the relative speed on the results.

DEFINING THE MICRO GEOMETRY

Defining the micro geometry is more complex and not accessible for a relatively simple optimization algorithm as the definition of the macro geometry. The micro geometry involves the profile of the conical roller surface, the profile of the roller large end surface and the angle of the outer ring rib. Since the roller-raceway contacts are considered non-crucial regarding the speed rating, their profile mainly affects the bearing stiffness and the maximum hertzian pressure. Generally, a logarithmic profile is favorable for rolling line contacts with regard to load distribution [4]. Given the manufacturing processes available, the roller profile for the line contact has to consist of a non-profiled center part followed by shallow circular arc shaped profiled areas and rounded edges at both ends. The profile and the transition from the non-profiled into the profiled zone have to be designed properly to reduce contact pressure peaks. However, those peaks can not be avoided completely since the manufacturing only allows for one large radius encompassing both profiled zones. Thus, a continuous transition from the non-profiled zone into the profiled zone can not be realized. It is assumed that the roller-raceway profiling has only negligible effects on the conditions in the roller-rib contact and vice versa. Thus, both profiles are designed independently. To design the roller-raceway profile, the non-profiled length is varied to set the stiffness. In the second step the profiling radius following the center part is chosen to keep the contact pressure low.

The surface of the large roller end is favorably profiled to form a section of a spherical surface [5] giving only one design parameter, the sphere's radius. The angle of the outer ring rib has to be chosen according to the large end face radius since it defines the location of the roller-rib contact point. Again, there is no simple objective variable to assess the high-speed suitability of the roller-rib contact. Pure sliding in the roller-rib contact occurs in a TRB with a non-rotating outer ring and an outer ring rib. This contact is therefore treated as a plain bearing and the product of normal contact pressure and relative speed (equaling the sum speed in this case) in the contact is taken as the main criteria to asses this contact. The large end face radius is varied in a wide range and for each discrete value the rib angle is analytical determined to give a nominal contact point location at the center of the rib height. For all parameter variations the bearing properties at different loads and speeds are calculated by using a newly developed TRB calculation program considering high-speed effects [6]. Subsequently the above mentioned objective variables are compared and a set of values for the design parameters is chosen.

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KEYWORDS

Rolling Bearings: Tapered Roller Bearings, Components: Machine Tools