Uncertainties in Heat Loss Models of Rolling Bearings of Machine Tools

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Abstract

The mastering of the thermal behavior of machine tools is an important task to increase the machining accuracy. Tools that are progressively used for this task are models that simulate the thermal behavior of the machines. A main part of the models is the representation of heat loss processes. An example of these processes is the friction in rolling bearings. The model approaches available to describe the bearing friction show a high degree of uncertainty. The causes and consequences of these uncertainties are investigated on the example of the representative Palmgren friction model for rolling bearings. Thereby especially bearings on feed axes are considered. The results are quantitative statements regarding individual uncertainties. These can now be used to estimate the overall uncertainty of the model under specific operational conditions and model assumptions. This improves the understanding of heat loss modeling and will be the basis for future model enhancements.

1. Mastering the thermal behaviour of machine tools

Mastering of the thermal behaviour of machine tools is an important task to increase its machining accuracy. This is due to the fact that a major part of the positioning errors that can cause machining accuracy are thermally induced [1].

Fig. 1. Thermal causal chain of machine tools and modelling uncertainties

Causes of these errors are inner heat losses and variations of ambient temperatures. The coherences are shown in the thermal causal chain (Fig. 1). The beginning is the load profile of the machine. This is characterized by time varying axis positions \( x \) and axis velocities \( v_x \) as well as ambient temperatures \( T_a \). The load profile initiates heat flows caused by ambient temperatures and heat losses. These heat flows change the temperature fields of the machine assemblies and lead to their thermo-elastic deformation. Due to the kinematic coupling of the assemblies a displacement between the work piece and the tool occurs and results in production deviations.

There are two measures to decrease the deviations that are different in principle. First there are compensatory measures, the aim of which is to influence the thermal behaviour in a positive way. Examples are the reduction of injected heat losses or of ambient temperature variations. Second, there are corrective measures, whose mode of operation permits the thermal processes including the assembly deformation. Thermal models are used to reproduce these processes. Based on a calculated thermal deflection, the error is corrected by applying offsets to the set values of the feed axes [2].

2. Thermal machine models and uncertainties of friction modeling

Essential for the presented measures are thermal machine models. They are used as an analysis tool for the design of compensatory measures as well as for the calculation of set values needed during controller-based correction. Thereby the
models reproduce the complex accuracy-relevant thermal and thermo-elastic behaviour of the thermal causal chain [2]. A crucial part is the modeling of heat flows causing the temperature changes. This is typically done using empirical model approaches and shows high uncertainty (Fig. 1).

Some of these heat flows are caused by inner friction of guiding elements. The model approaches for friction based heat losses show significant uncertainties. Without metrological adjustments the models can deviate 30 to 200 % from readings as investigations of rolling bearings show [3]. Due to a direct proportional correlation between the heat losses and resulting thermal deformations, the uncertainties of the calculated deformations are also significant [4]. Therefore the resulting statements in compensation and correction measures based on this type of models have also considerable uncertainties.

The aim of this paper is to enhance the understanding of uncertainty effects and its influences as a base for sufficient modeling of the thermal behavior as well as adequate interpretation of simulation results.

3. Uncertainty studies of rolling bearing models

The friction of guiding elements is studied on the example of pre-loaded and grease lubricated angular contact bearings. These bearing construction characteristics are common on machine tools. The focus is on fixed bearings of ball screw drives because these are one of typical main heat sources.

The friction modelling is based on the approach established by Palmgren [5]. It is one of the most used and investigated models for this type of bearings. Therefore a wide range of information on its application exists. The parameters are easily available in manufacturer information and in machine design data. Due to the low number of parameters there is little effort for parameterization. The accuracy of the friction prediction is not far behind and sometimes better than with other, more detailed approaches [3]. So altogether, the Palmgren approach provides a very efficient way to model and calculate friction.

4. Correlations of the Palmgren friction model

The heat loss $Q_f$ generated by a bearing is calculated as the product of frictional torque $M_f$ and rotational speed $\omega$, whereby the frictional torque of rolling bearings can, according to Palmgren [5], be described by the sum of the two components $M_0$ and $M_1$ (units converted to SI-system):

$$Q_f = M_f \cdot \omega = (M_0 + M_1) \cdot \omega$$

$$M_0 = \begin{cases} 16 f_0 d_m^2, & \omega \cdot \nu < 2 \cdot 10^4 \\ 4501 f_0 d_m^2 (\nu \cdot \omega)^2, & \omega \cdot \nu \geq 2 \cdot 10^4 \end{cases}$$

$$M_1 = f_1 (P_d) g_1 P_d d_m.$$  

(1)

(2)

(3)

The component $M_0$ is a function of speed and has the following parameters: $f_0$ - characterizing the bearing design and lubrication mode; $d_m$ - average diameter of the bearing and $\nu$ - viscosity of the lubricant or in case of grease the viscosity of the base oil. Parameters of the load component $M_1$ are: $f_1$ - coefficient depending on bearing type and equivalent load ($P_d$), $g_1$ - actual load as a function of axial and radial forces and $d_m$ - average bearing diameter.

The model is of empirical type, since it is based on correlative relations to measurement data, obtained from numerous experiments. The magnitude of uncertainty of the original Palmgen correlation is shown in Fig. 2. The typical variation of measuring points and the fitted curve of the approach is demonstrated with the example of axial spherical roller bearings. The deviations of the measuring points range from -40 to +60 % of the fitted curve. Because of missing information about the conditions of the studies, a deeper investigation of the causes of uncertainty is not possible in this particular case.

5. Model uncertainties considering operational conditions

The analysis of uncertainties outlined here addresses the friction behaviour under operational conditions on machine tool bearings. This behaviour includes the long-term effects of running-in and wear as well as the effects during machine operation because of varying outer loads and temperatures.

5.1. Influence of long-term effects

At first the effects of the long-term behaviour are investigated. There are no torque measurements known over the lifetime of grease lubricated roller bearings, measurements on preloaded profile rail guideways [7] are used instead (Fig. 3), since the friction processes are similar to roller bearings because of the balls as rolling elements. However, additional effects of ball recirculation only permit a partial transferability of the findings.

The measurements in Fig. 3 show a significant change of the frictional force in a running-in and a wear period compared to nearly no change during a relatively long period of constant friction. The magnitude of friction reduction in the running-in period can only be roughly estimated because of warming up effects in the beginning of the measurements. The wear period at the end shows a big impact. It accounts to
28% of the theoretical life time and increases the friction by up to 1/3 of the value of the constant period.

Fig. 3. Long term behaviour of a profile guide rail (measurements [7])

Information about the running-in behaviour of loaded angular contact bearings lubricated by grease gives [8]. Herein the measured values show a significant decrease of frictional torque to about half of the initial value. Unfortunately there is no information given regarding temperature and temperature related influences on friction.

The Palmgren model represents the effects of running-in and wear in a simplified manner. In [9] a statement is given, that the model is only valid after running-in. If the bearing is freshly greased, the bearing factor $f_0$ can be 2...5 times higher.

5.2. Influence of outer operating Load

During operation of the machine tool there are constantly changing loads on the feed axes. This is due to altering of work piece mass, acceleration and process forces. The resulting forces on the fixed bearings of ball screw drives are mainly in axial direction. The bearings are typically selected so that the operational load uses the full range permitted.

Fig. 4: Load diagram of a preloaded angular contact bearing

Since the Palmgren model considers the actual load, the load would have to be adapted as it changes over operation time. But in case of preloaded bearings, based on certain assumptions, it is possible to neglect this and consider only the initial preload. This simplification is usually applied because the measurement or the estimation of the actual operational loads is very expensive.

The key assumption for neglecting the actual load is that, despite of the changing outer operating loads, the sum of the two inner forces on the rows of rolling elements is nearly not changing. Therefore, if the sum of the forces is not changing, neither does the resulting sum of friction.

This assumption applies only approximately, because of the non-linear elastic behaviour of the rolling contact. Fig. 4 depicts this on the example of a preloaded bearing of the type Schaeffler ZKLN30100. The calculations are based on data from [10]. The deviations of the load sum regarding to the bearing without operational load are -16/+15 %, which is in turn the uncertainty for neglecting the operational load.

Fig. 5. Angular contact bearing for feed axes

5.3. Influence of preload

Another influence with relevant uncertainty is the preload. In case of the currently considered bearing (Fig. 5) the preload is realized by a mounting force. The mounting force pushes the inner bearing rings up to their contact. The resulting preload is only indirectly related to this. Rather, it depends on the bearing stiffness and a geometrically defined deflection. The stiffness is mainly determined by the rolling contact. Whereas the amount of deformation is specified by a manufactured gap between the inner rings that exists prior to assembly.

The nominal size and the manufacturing tolerance of this gap determine the actual deformation, and therefore the preload. Both influencing factors can be determined for the bearing type ZKLN30100. The calculated gap is 23 µm. The lower limit of aggregate manufacturing tolerance of the gap can be estimated with ±2.5 µm. It can therefore be concluded, that the preload is varying in the range of ±11 % around its nominal value.

5.4. Temperature influence

The time-varying character of the machine’s load profile leads to changing heat loss in and around the bearings. This in turn changes the temperatures of the bearings and its surroundings. The resulting rise in bearing temperature may extend up to $\Delta T = 40$ K above ambient temperature of the feed axes. The range of the ambient temperature in turn is approximately 0 to 40°C, depending on the operating location of the machine.

The changing temperature affects the viscosity $\nu$ of the lubricant, which is an input value of the Palmgren model. The influences on the viscosity can be observed especially at low temperatures. There is a strong non-linear dependence. These effects can be estimated on the example of Fig. 6. It shows the
calculated frictional torque of a bearing as a function of temperature. The calculation uses the Palmgren model and the temperature dependent viscosity of a typical base oil.

![Figure 6. Frictional torque as a function of temperature](image)

Neglecting this dependency on certain operational conditions can lead to significant uncertainties. In these cases the changing viscosity of the base oil has to be taken into account e.g. with the approach according to Vogel [11].

The model approaches for the temperature dependent viscosity require viscosity data of the lubricant. This is provided by the manufacturer. For the information of the viscosity class a tolerance of ±10 % is permitted according to ISO-3448. If manufacturers provide additional points of viscosity over temperature, the tolerances are typically around ±10 %. Oftentimes the provided information covers only a small temperature range. In these cases it is necessary to extrapolate the v-T-behaviour. This can be more uncertain for grease because in the temperature range below 40°C base oil precipitates can additionally increase the viscosity (DIN 51563).

Besides the influence of viscosity the temperature has also an impact on the thermo-elastic behaviour of the bearings and its surrounding components. The deformed structure in and around the bearing can change preloads and with it frictional torques of the bearing. The characteristic of the thermo-elastic bearing behaviour may differ substantially and depends on: (a) the kinematic design to create the preload (e.g. X- or O- arrangement, row spacing), (b) the inner stiffness and the stiffness of the connection between bearing and surrounding components as well as (c) the thermal connection of the bearing to its outer heat sources and heat sinks.

The magnitude of the thermo-elastic behaviour may well have a relevant impact on the frictional torque. This is shown on the example of the bearing ZKLN30100 (Fig. 5). Assuming the realistic scenario of an increased temperature of the inner rings by 4 K, this results in a nearly free thermo-elastic deformation of the rings in axial and radial direction. By considering the kinematic contact conditions and the catalogue values for the bearing stiffness, the calculated preload rises by about 10 %.

6. Summary and Conclusions

The prediction of bearing heat loss under operational conditions of machine tools using available model approaches is significantly uncertain. This is especially true under the varying influences of long term behaviour, load and temperature. The causes of model uncertainties were investigated on the example of fixed bearings of ball screw drives and of the Palmgren friction model. Quantitative statements were given for essential uncertainties. Now, based on this, it is possible to estimate the overall uncertainties under the operational conditions and model assumptions present in each case.

The uncertainties can be classified into three categories: First, uncertainties related to limited abilities of available model approaches to reproduce relevant effects. Second, uncertainties due to the determination of model parameters and third, uncertainties belonging to varying bearing properties caused by manufacturing and mounting tolerances.

The amount of the tolerance related uncertainties is, despite of small manufacturing tolerances, quite significant. This has been shown on the example of the bearing type Schaeffler ZKLN30100. The sum of tolerance-related uncertainties can be estimated to not less than ±20 % of the nominal value. This estimation includes the presented tolerances of preload and base oil viscosity. The actual uncertainty is certainly higher because the considered values are lower limits and there are additional disregarded uncertainties which may have a significant value.

Finally, the uncertainties of friction models are one essential input value to determine the overall simulation uncertainties of thermal machine models. With these uncertainties it is possible to estimate the variation range of the real thermal machine behavior. This information in turn allows a more detailed interpretation of simulation results.

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References