Precision of prestressed ball screw thermal behaviour in machine tool operating conditions

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Abstract: This paper presents an FEM model of a feed drive with a ball screw stiffness-wise, thermally and motion-wise integrated with the machine tool structure. The model takes into account the moving heat sources, the frictional and inertial interaction of the masses being shifted and the variable thermoelastic and stiffness interactions within the drive and the machine tool load-bearing structure. The results of heating up and displacement calculations for the ball screw, the bearing supports and the whole machine tool are analysed with regard to positioning accuracy.

Keywords: machine tool; ball screw; model.


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1 Introduction

The ball screw still remains the principal component of the feed motion drive in numerically controlled machine tools. Owing to the fact that it is about 80% less expensive than a drive with a linear motor and that it does not adversely affect (no magnetic field) the human body and electronic circuits, intensive efforts are made to improve it so that it meets the requirements of high speed cutting (HSC) and high performance cutting (HPC).

The research on the ball screws used in high speed machine tools covers the experimental identification of the behaviour of the ball screws (Altintas et al., 2011; Mayr et al., 2012) and attempts at reducing the positioning (mainly thermal) errors generated in them (Winiarski et al., 2010). The aim is to reduce power losses and heating up (Wu and Kung, 2003) and to increase dynamic stability (Altintas et al., 2011). From among the works devoted to the study and modelling of the physical processes taking place in ball screw assemblies the ones cited below deserve special attention. Wu experimentally identified the distribution of temperature along the ball screw for several rates of feed and determined the dependence between the feed rate and the magnitude of the heat flux flowing to the ball screw. In his FEM calculations of the positioning error he assumed a trapezoidal distribution of the flux of heat (proportional to the contact time) transmitted from the nut to the ball screw, taking into account the bearings supporting the ball screw as heat sources (Wu and Kung, 2003). Heisel et al. (2006) in order to calculate positioning errors would add up the elongations of a few mm long sections of the ball screw, determined from its thermal images obtained by means of a thermal imaging camera. A similar procedure was adopted by Horejs et al. (2007), who also undertook an attempt to include the variation in power losses in the ball screw bearings as a function of rotational speed and the axial force transmitted by the ball screw, in the FEM computations. Kim et al. (1997) in order to calculate the moment of friction in the screw-nut joint used the values of the current drawn by the motor at different feed rates. Having included an additional element (gap element), to which he assigned the heat generated
between the screw and the nut, uniformly distributed along the length of their contact, he would analyse the distributions of temperature in the ball screw. Gleich (2007) proposed an FEM model of a pseudo-moving heat source, described by a small number of elements and nodes and affected through repeated semiautomatic coupling of the screw and nut discretisation meshes. Finally, Yang et al. (2013) adopted a multi-zone heat load between the nut and the screw, simulating the reciprocating motion of the nut. The analytically determined power losses in the nut took into account the acceleration, constant speed and braking of the drive.

However, none of the models also Miyaguchi and Arai (2013), Neugebauer et al. (2007) and Schulz and Schmitt (1994) proposed to date takes the totality of the dynamics of the changes in the heat and force loads in the feed drive with a ball screw. In order to achieve this it is necessary to model a moving heat source and such neglected phenomena as:

- the change in the pulling force in the screw, connected with the acceleration of the masses and their braking
- the decline in the initial tensile stresses in the screw with increasing temperature
- the variation in the power losses in the bearings and in the nut as a function of the force in the screw
- the deflection of the bearing supports
- the variation in the friction forces on the slideways as a function of speed.

The model presented here is an attempt at meeting the above requirements.

2 Model of distribution of forces in machine tool feed drive

The positioning error of a drive with a ball screw is the result of the axial displacements of the ball screw caused by changes in loads (forces) and in thermal displacements. The value of this error depends on the value of the axial force in the screw, produced by the stretching during the assembly of the screw, and on the changes in this force and the displacements resulting from the thermal deformation of the whole machine tool.

The basis for the model are the moving heat sources (the nut and the table slideways) which make it possible to faithfully model the actual operating conditions of the drive. In order to build a comprehensive model of the feed axes of the machine tool, integrated with its whole structure, compromises between modelling accuracy and computing power need to be worked out.

A change in the position of the moving parts of the lathe over time entails changes in the force loads stemming from the mass and inertia of the moving unit. Both the factors cause reactions on the slideways. The reactions usually depend on the speed-variable coefficient of friction. Computations were performed for a given work cycle (i.e. in the successive position of the feed axis), using the FEM, to determine the variation in the forces acting on the moving parts of the machine tool (Figure 1).

The conditions of connection with the slideways in the normal direction (the contact stiffness) and the variable conditions of friction in the tangent direction (the coefficient of friction) were defined for the moving component in each time step. Figure 2 shows an
friction force as result of the tangent stresses (friction) automatically computed on the machine tool bedways in the model, changing their sign with a change in the speed direction.

**Figure 1** Forces acting on moving machine tool component (axis Z) (see online version for colours)

![Friction coefficient](image)

**Figure 2** Change in sign and position of friction forces on bedways as function of time and speed direction (see online version for colours)

Friction forces: Friction 1 and Friction 2 generate power losses on the two bedways, causing a temperature rise and deformations of both the bed and the slidable machine tool parts, and overload the other kinematic pairs of the drive.

The phenomena connected with deformations in the screw-nut joint (Figure 3) were reduced to:

- the reaction in the nut to the forces generated by the resistances to motion of the slidable component, defined as

  \[ R_n = \text{Inertia} + \text{Friction1} + \text{Friction2} \]  

(1)
• a known value of nut displacement \( Sz \) in the place of contact with the ball screw, being a function of speed and time
• rigid connection between nut and ball screw for every position of nut.
• the axial force in the screw \( (Rs = -Rn) \) as a function of speed and time.

Having calculated the reactive forces in the slideways one can determine the friction forces and knowing the reactive force in the nut one can determine the force in the screw, loading the bearings connected with the machine tool bed.

• computed force in the nut and in the screw enable using analytical equations determine power losses in nut and front and rear bearing
• for every position of nut power losses \( Qn \) are transferred to screw causing changes of ball screw temperature and positioning accuracy.

**Figure 3** Axial forces \( Rn, Rs \), boundary nut displacement \( Sz \) and calculated power loses in nut \( Qn \) (see online version for colours)

Figure 4 shows the forces taken into account in the computations, which act on the ball screw and on the machine tool bed. Among them there is ball screw initial tension force \( R \) the function of which is to reduce the thermal deformations of the ball screw and to improve the positioning accuracy of the machine tool. Such an initial value of the force was adopted that the deformation at the end of the screw amounted to 50 µm. This was achieved by introducing an appropriate interference between the nut and the washer in the right-hand set of bearings. As the ball screw heats up as a result of the motion of the nut, instead of thermal elongations, the relaxation of the initial ball screw stresses takes place. The stiffness properties of the right-hand support bearings were modelled through appropriate contact properties mapping the nonlinear characteristic of the bearings. The linear characteristic of the preloaded bearings in the left support was realised by a separate screw element which was assigned an appropriate Young modulus.

In Figure 5 the resistances to motion, represented by the calculated friction force and the force of inertia (in newtons), are compared with the current drawn by the feed motor measured for a turning centre. It is visible that computed main load and measured current representing motor torque have the same shape. The diagram also shows the variation in speed and support position along axis \( Z \) in a single work cycle.
This verification of the computations indicates that the model is highly accurate in estimating force loads for both moving and stationary machine tool components. It appears from Figure 5 that because of the high variation of the forces during the cycle, time steps not larger than 0.1 s need to be adopted in order to obtain accurate results. In the case of an analysis of the whole machine tool structure, this entails many hours of computations.

Figure 5 shows tangential stress (friction) on the bed guides, changing its value and position over time and its sign with a change in the sense of the speed vector (change in the direction of motion). The friction forces: Friction 1 and Friction 2 on the two guides generate power losses which thermally load the machine tool structure.
3 Thermal model of feed drive

In order to develop a tool for computing temperature distributions and the associated deformations in machine tools with moving assemblies it was necessary to find ways of modelling power losses in the machine tool fixed and slidable components. The assumptions for power loss computations are shown in Figure 6.

Figure 6 Sources of power losses in feed axis Z (see online version for colours)

Power losses (marked green) on the surfaces of the slideways are automatically calculated on the basis of the friction force and the work cycle-dependent instantaneous speed of the unit. This friction results from the pressure exerted on the slideways by the mass of the moving unit and from the action of the inertial forces and the speed-variable coefficient of friction, as illustrated in the upper part of Figure 6. The losses are generated in actual places of contact with the slideways, consistently with the instantaneous axis unit position in a given time. Power losses between the nut and the screw change cyclically consistently with the speed and are generated by the moment of friction in the nut, which changes with the force driving axis Z (the pulling force in the screw). The losses in the nut/screw joint, presented in Figure 6, were calculated following the guidelines provided by NSK and HIWIN. Power losses in the bearings depend on their
preload and on the changes in speed and stretching force in the screw (Jedrzejewski and Kwasny, 2013). The power losses were calculated using the Palmgren (1964) relations, taking into account the preload of the front set bearings (on the motor side) and its absence in the rear bearings as well as the load produced by the temperature-variable screw tension. Also the hydrodynamic losses determined by the rotational speed and the viscosity of the lubricating medium were taken into account. The power losses in the slideway joints were automatically calculated as a function of the speed and the friction forces and distributed among the components proportionally to their heat flux reception capacity. The power losses in the other heat sources were calculated from the analytical relations presented earlier and have been assigned to the volumes and the surfaces on which they arise.

A major factor having a bearing on the intensity and location of the thermal load is the drive work cycle (which determines the frequency and height of heat peaks). Figure 7 shows that at a speed of 30 m/min and a distance of 400 mm the drive work cycle amounted to 3 s, which included acceleration + uniform motion + braking –1 s and a break of 2 s.

**Figure 7** Comparison of computed and measured ball screw nut temperatures (see online version for colours)

![Increase of nut temperature](image)

**Figure 8** Comparison of computed and measured ball screw nut temperatures in first 10 seconds of operation (see online version for colours)

![Increase of nut temperature](image)
Under the above conditions the computed and measured temperatures on the screw gear nut were compared in Figures 7 and 8. Because of the quick changes in temperature over time caused by the movement of the nut and by the change in its load a wide band of temperatures is obtained from the computations. One can assume that the model performs correctly. An attempt to improve it would require a very precise experiment, accurate temperatures in the nut itself (consistency of the locations of the measuring points with those of the heat generation points) and the measurement of the ambient temperature. The agreement between the temperatures in Figure 7 is good for the short gear operation time proper for the natural operating conditions. In a longer time interval the influence of the ambient temperature in the whole machine tool model becomes clearly evident. The pattern of changes in nut temperature in the first seconds of heating is shown in Figure 8.

The increasing temperature values result from the 1-second feed along axis Z, generating power losses in the nut, whereas the 2-second drop in temperature arises during standstill. It is evident that the model renders well the character of the temperature changes taking place in the nut as a moving heat source. Simulation curves are not very precise, because at the end of the movement to early cooling takes place. It was caused by automatic changing time step from 0.1 s to 0.25 s. The time step 0.25 s create possibility of omitting important phenomena increasing power losses during breaking moving part which take 0.2 s. It is therefore important not to automatically determine the time step, especially in the case of expecting very precise calculation results.

4 Thermal error reduction for turning centre feed axis

The total positioning error is understood as the total elongation of the ball screw itself and the elastic deflection of the supports and the bearings, caused by a change in load and temperature in the whole machine tool. Computations were carried out using the above model in order to assess the possibility of reducing the feed drive positioning error through ball screw preloading. A ball screw preload equivalent to a 50 μm deformation of its tip. a speed-variable friction coefficient and a motion cycle: pause 2(s) ▶ movement 30 m/min (distance +400 mm) ▶ Pause 2(s) ▶ 30 m/min (distance +400 mm) were assumed.

An analysis of the computation results shows that the initial deformations of the supports, caused by the screw preload, after 1,200 seconds almost disappear as a result of the increase in the temperature of the screw, and stress relaxation. Moreover, changes in the displacement of the points situated along the screw would amount to max 10 μm. The increment in temperature amounted to 6 K.

The beneficial effect of screw preload is apparent in Figure 9 where the displacement values in one of the points on the screw surface (point of maximum displacement) are compared for the situations with and without preload. The changes of the position of the selected point on the screw are clearly larger in the case of the screw without preload.

This has a direct bearing on the accuracy of positioning the slide along axis Z. Also the thermal elongation of the screw in the screw/nut contact caused by the moving heat source, reduced during downtimes, affects this accuracy. Therefore the influence of downtime must be taken into account in positioning error modelling. Figure 10 shows that downtime has a considerable influence on both the screw temperature and the positioning error. The latter can be reduced by introducing a longer pause.
5 Conclusions

Using the proposed model one can extensively analyse the behaviour of a feed drive within the machine tool structure and examine the mutual interactions, particularly with regard to the forces connected with the inertia of the moving masses. The following conclusions emerge:

- Proposed methodology enables one to model most of the phenomena occurring in real machine tools feed drives.
- Computing model comprising moving units makes it possible to precisely describe the power losses the heat transfer between parts of feed drive and its thermal behaviour.
- Forced by changeability phenomena sampling time result long computing time.
- Preload (50 μm) of the ball screw in the Z direction reduces the changes of displacement to 12 μm in comparison with 55 μm for preload = 0 μm over a time of 1,200 s.
• Owing to preload the displacement of a large part of the ball screw working length is close to zero.

• Increase of pause in recurring cycle of turning centre carriage enables to reduce positioning error by amount 15 μm (from 65 to 50 μm).

• The future research should be focused on considering in model higher dynamics of ball screw drive.

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