COMPARING DIFFERENT COOLING CONCEPTS FOR BALL SCREW SYSTEMS

Josef Mayr[1], Markus Ess[2], Sascha Weikert[2], and Konrad Wegener[1,2]
[1]Institute of Machine Tools and Manufacturing (IWF)
[2]inspire AG
Swiss Federal Institute of Technology (ETH)
Zurich, Switzerland

ABSTRACT

Different design modifications to reduce the thermo-elastic deviations on a prototype lathe caused by moving linear axis are discussed. The machine tool has no linear encoders for the controlled axis whereby the TCP-displacements particularly at long-term machining can be enormous.

A simulation study was used to find out what design modifications can help to reduce the thermally induced deviations between the work-piece and tool and to what extent the deviations can be reduced. The example illustrates that the thermally induced TCP-displacements of the prototype can be reduced to about 15% of the original deviations by design modifications without using linear encoders.

Short-term variations in the boundary conditions cause unwanted long computation time. If possible, boundary conditions with short-term variations should be simplified. In an example the interpolation of the temperature range of the coolant temperature control reduces computation time by a ratio of almost five. With such simplifications the whole simulations study with more than 25 simulations can be realized in less than three days computation time including model modification.

INTRODUCTION

Internal and external heat sources and sinks initiate thermo-elastic deformations on machine tools as well as on the work-piece. In the end the thermo-elastic deformations lead to geometrical inaccuracies on the work-piece. Thermal influences are typically the major source of reduced machining accuracy on machine tools and prominently contribute to the overall geometrical inaccuracy of the work-piece [1,2].

The basic effects of thermal inaccuracies of machine tools have thoroughly been investigated for decades. Based on this, design rules have been derived, which thoroughly are applied in ultra precision machine design [4].

To improve thermal stability of linear axes typically linear encoders are used. The benefits of using linear encoders have frequently been demonstrated. Other methods like cooling the ball screw spindle with a temperature controlled coolant may have disadvantages. For example a shaft feedthrough may be required. An other disadvantage of cooling the ball screw spindle is that if no linear encoders are used the spindle is used as reference frame and therefore the control range of the coolant temperature swayed the axis position [5,6].

For precision machine tools ball screw manufacturers developed cooled nuts [7]. The cooled nut should inhibit the heat flux from the ball screw system into the machine tool structure.

BALL SCREW MODEL

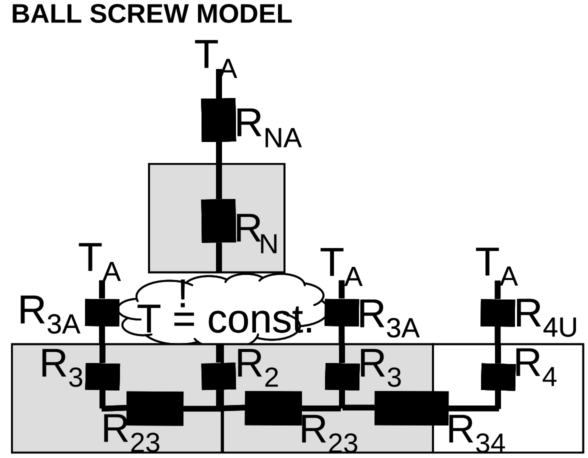

FIGURE 1. Thermal equivalent circuit diagram of the ball screw system, R: thermal resistance, T_A: ambient temperature, T: rolling body temperature.

In Figure 1 the thermal equivalent circuit diagram of the ball screw system is shown. The equivalent circuit diagram has three main elements: the rolling bodies, the spindle and the nut.

The model is used to determine the heat flux in the ball screw system. One part of the heat generated during operation flows into the spindle which leads to a temperature increase in this element. The temperature increase leads to a part of the heat being led down to the ambient air via convection. The other part of the heat generated in the ball screw system flows into the nut and from the nut to the ambient air.

The spindle is subdivided into 2 areas. On the one hand the area which is passed by the nut during operating (shown in Figure 1 highlighted in gray) and on the other hand the area which is just in contact with the ambient air (shown in Figure 1 highlighted in white).

The rolling body temperature is unknown. It is assumed that through the rotation of the rolling bodies the temperature on all contact points is equal.

Thermal effects on machine tools usually are long-term effects. Therefore having the same temperature at all contact points the ratio of the heat flux into the spindle and into the nut can be assumed as being the same as in steady state.

With these assumptions the thermal equivalent circuit diagram can be simplified to the equivalent resistance diagram shown in Figure 2 using Kirchhoff's laws.

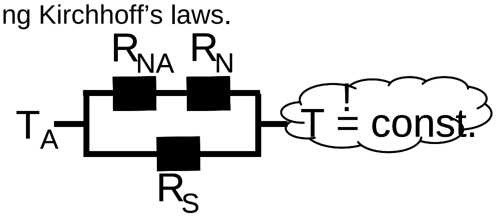

FIGURE 2. Simplified thermal equivalent circuit diagram of the ball screw system, R_S: thermal resistance spindle side, R_N: thermal resistance of the nut, R_{NA}: convective thermal resistance of the nut to the ambient air.

MACHINE TOOL MODEL

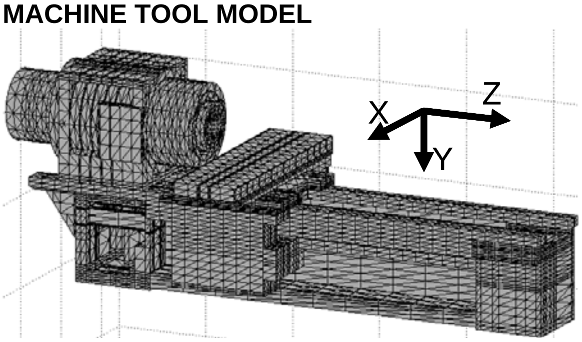

FIGURE 3. Model of the lathe.

The model of the lathe under investigation is shown in Figure 3. The model includes all necessary structural components of the machine tool like the spindle, the spindle cooling system, the bed and the linear axes. The motors e.g. are not included into the model. The motor temperature is included as thermal boundary condition based on measurements.

LOADING CONDITIONS

The simulation study was done for a load case consisting of a moving Z-axis: The Z-axis oscillates between the measurement position (the axis position shown in Figure 3) and the end of the axis travel at a distance of 318mm. The considered heat sources and sinks are the friction in the ball screw system and hydrodynamic linear guides, the coolant, the ambient temperature profile and the linear axes motor temperatures.

In the first 1.8 hours the Z-axis oscillates with 3m/min feed rate. Afterwards the Z-axis is 3.8 hours oscillating with 5m/min feed rate. At the end of the simulation cycle the z-axis is positioned at the measurement position for cooling down for another 14 hours. To ensure that the simulations can be compared with measurements the simulation cycle was chosen based on measurements previously done. From this measurement all boundary conditions like the ambient temperature profile are known. As examples the measured boundary conditions ambient temperature profile and the temperature of the Z-axis motor are shown in Figure 4. The measurement profiles have typically to be smoothed before they can be used as boundary conditions for simulation in order to reduce simulation time. As shown in Figure 4 different interpolation methods are used: E.g. the ambient temperature profile and the motor temperature interpolated based on linearly or on exponentially fitted curves.

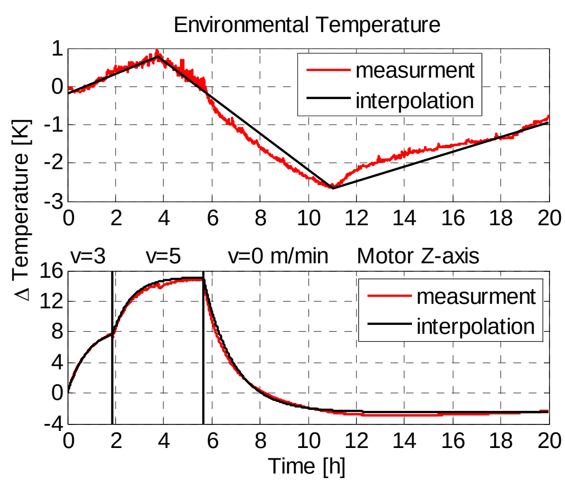

FIGURE 4. Measured and interpolated temperature profiles, top: environmental temp., bottom: temp. of the z-axis motor.

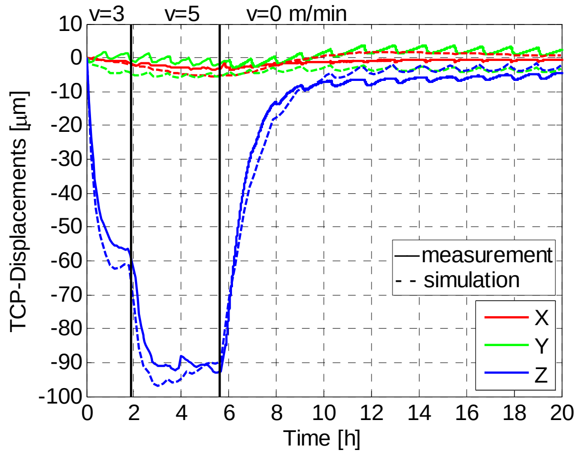

FIGURE 5. Measured and simulated TCP-displacements in X-, Y- and Z-direction.

As can be seen in Figure 5 the measured and computed TCP-displacements show a good correlation within a difference in maximum of about 10µm in Z-direction. In the other directions the TCP-displacements and also the difference between simulation and measurement are even smaller.

COOLANT TEMPERATURE

The coolant of the spindle is controlled to 21.5 ± 1.7°C. If the maximum temperature of 23.2°C is reached the cooling process starts while the temperature is higher than 19.8°C. The consequence of this control is a saw tooth profile in spindle coolant temperature. Applying this saw tooth profile in the simulation leads to abrupt changes in the boundary conditions. Such changes in boundary conditions usually cause an increase of computation time due to reduced step size. Therefore it is advisable to simplify those boundary conditions as shown in Figure 6.

The control range of the temperature of the coolant can be reproduced as sinusoidal profile or as being constant. The cycle period with the saw tooth profile is not constant. Here, the measured profile was used which is swayed by the ambient temperature and the heat input.

In Figure 7 the step sizes chosen by the solver (Matlab/Simulink, ODE 15s) are shown. As it can be seen every change in boundary conditions enormously reduces the step size. Therefore with the saw tooth profile the computation of the temperature distribution over a simulation time of 20 hours takes about 850 s using the FDEM simulation approach [3,4]. With a sinusoidal profile the same simulation just takes 181 s and with constant flow temperature of the coolant just 167 s are required. As it can been seen in Figure 7 with a sinusoidal flow temperature of the coolant the maximum step size of 1'000 s (chosen by the operator) cannot be reached by the step size control but the influence to the total cost of computation time is secondary.

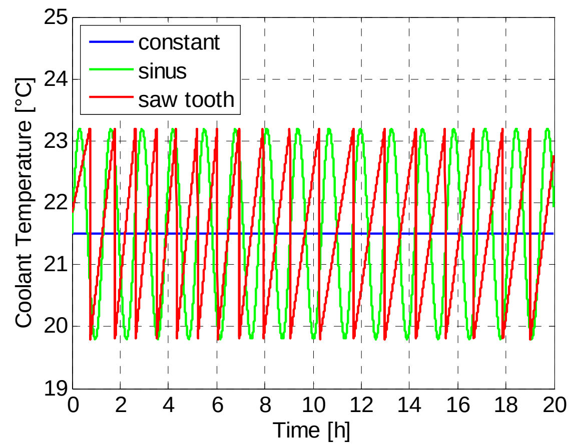

FIGURE 6. Flow temperature of the coolant used as boundary conditions in the simulation.

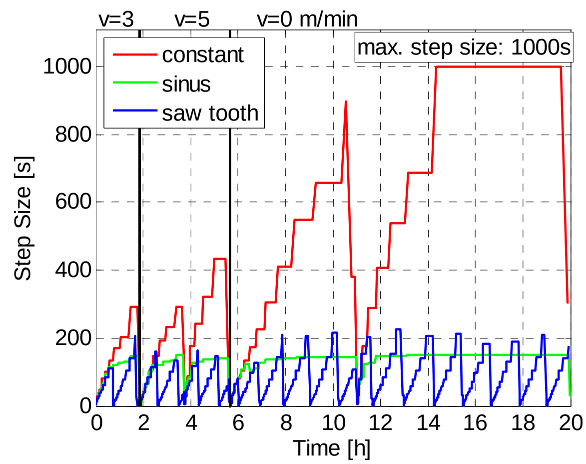

FIGURE 7. Step size with different interpolated coolant temperature profiles.

SIMULATION STUDY

With a simulation study it should be investigated whether the TCP-displacements can be reduced without applying linear encoders. For this, several design modifications are discussed. The ball screw manufacturer for example offers two types of cooled nuts. Another modification is changing the location between the ball screws thrust and floating bearing. At the end of the bed, at the opposite side to the main spindle, the Z-axis motor and the thrust bearing of the ball screws are located. On the prototype ball screws are all used in fixed-free configuration. With the modified assembly the thrust bearing is located

at the main spindle side and at the motor side an additional floating bearing is mounted to support the forces from the transmission.
Other ideas discussed are using cooling fins or cooling plates to isolate the thermal influence of the motor from the bed. The two assemblies discussed are shown in Figure 8.

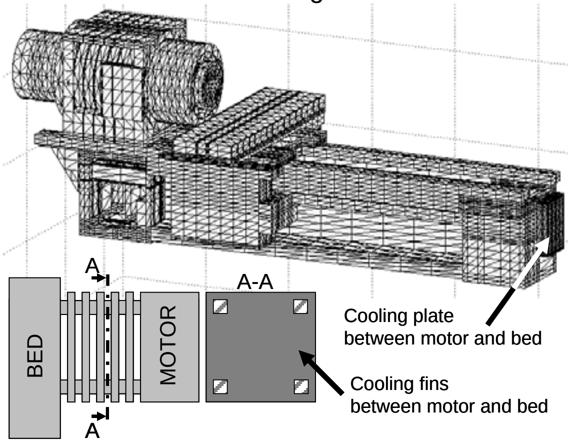

FIGURE 8. Cooling plates and cooling fins for isolating the motor temperature from the bed.

Cooling fins like the shown assembly in Figure 8 help to keep off the motor heat from the bed. The simulations have shown that through the isolation the temperature of the motor is rising. Therefore it is not unconditionally advisable to use such an assembly.
Cooling plates reducing the motor and bed temperature can always be used for isolating heat sources from machine tool structures.

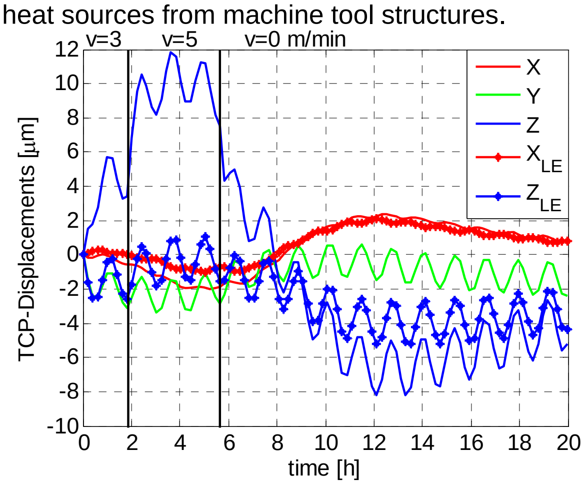

FIGURE 9. Simulation results, translatory TCP-displacements after implementing design modifications into the model, LE linear encoders are additionally considered, original curves see Figure 5.

Based on the ball screw model shown in Figure 2 the heat flow into the nut can be swayed when changing the boundary conditions. With a fluid cooled nut the convective resistance R_{NA} can be reduced. In such a case more heat flows into the nut while the thermo-elastic deformation of the spindle is reduced. The ball screw model has shown that over 97% of the friction heat generated in the ball screw can be conducted through the nut into the coolant. Nevertheless a limit in reducing thermo-elastic deformations of the spindle will be reached earlier because other heat sources like the friction heat of the bearings become more prominent.
In Figure 9 the minimum reachable translatory TCP-displacements after implementing the design modifications are shown. The greatest influence has the change of thrust bearing position and the cooled nut. Other changes like isolating the motor heat from the bed have secondary influence to the translatory TCP-displacements but help reducing rotatory TCP-displacements. The simulations study has also shown that the TCP- displacements can be reduced more when linear encoders for the X- and Z-axis are used additionally.

ACKNOWLEDGEMENT
The authors like to thank the Swiss Federal Office for Professional Education and Technology (OPET/CTI) for financial support.

REFERENCES
[1] J.Bryan. International Status of Thermal Error Research. Annals on the CIRP. 1990; 39/2: 645-656
[2] M.Weck, P.McKeown, R.Bonse, U.Herbst. Reduction and Compensation of Thermal Errors on Machine Tools. Annals on the CIRP. 1995; 44/2: 589-598
[3] J. Mayr, M.Ess, S.Weikert, K.Wegener. Simulation and prediction of thermally induced deformations on machine tools caused by moving linear axis using the FDEM simulation approach. Proceedings ASPE 2008 Annual Meeting. 2008
[4] J. Mayr, M.Ess, S.Weikert, K.Wegener. Thermal Compensation uning Three-Dimensional Physical Machine Tool Models. Proceedings of the 12th Mecha-tronics Conference. 2010; 1/2: 279-284
[5] J.Braasch. Genauigkeit von Vorschub-antrieben. Heidenhain Technische Information. 2004; 5
[6] J.Braasch. Optimizing feed axis precision. Motion System Design. 2002; 23-25
[7] HIWIN. Lineartechnologie. www.hiwin.de. 2009: 60-64