Analysis of Thermal Volumetric Effect on Machine Structure Using FEA Model

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Abstract

Thermally induced errors are the dominant source of machine tool (MT) inaccuracies and are often the most difficult types of errors to reduce. Software compensation of thermally induced displacements at the tool centre point (TCP) is a widely used technique to reduce these errors. An extensive set of measured data is necessary for calibration of these compensation models. For a compensation model valid across the entire workspace of a horizontal milling centre, many measurement points are typically needed, making experimental analysis costly and time-consuming. This paper presents an analysis of the thermal behaviour of a horizontal milling center using the finite element analysis (FEA) model without experiments at different positions of the Y and Z machine axes. The most significant thermal behaviour is observed when the Y-axis position is changed. Based on the simulation results, critical measurement points were identified to guide subsequent experiments and support the development of a compensation model. The proposed method reduces the number of necessary experiments, thereby shortening the time required for the experimental analysis of the thermal behaviour and lowering costs. In addition, the paper discusses alternative descriptions of spindle bearings as one of the main heat sources in the FEA model.

Machine tool, Finite element method (FEM), Volumetric error, Thermal error compensation

1. Introduction

The machine tool (MT) is loaded during the working process by various dynamic phenomena that result in deformations, e.g. vibrations, chatter, gravitational forces and thermal effects, etc. Geometric errors resulting from non-stationary thermal phenomena can be described as displacements from the desired tool and workpiece path. These errors cause 40-70% [1] of the total MT inaccuracy. Given the importance of this source of error, minimizing thermal errors has been a long-standing goal of MT designers and programmers. There are different approaches to mitigating thermal errors. In general, solutions to thermal errors can be divided into three basic groups: machine tool design to reduce sensitivity to heat flux (e.g., thermally symmetric machine tool design, thermal insulation, etc.), control of the temperature of the machine tool and its environment (e.g., control of the machine tool cooling system, etc.), and thermal error compensation [2]. This paper presents an analysis of the thermal behaviour of a horizontal milling centre using an FEA model without experiments at different positions of the Y and Z axes of the machine. The results enable minimization of measurement points required in the subsequent experimental phase, thereby reducing overall analysis time and cost.

2. Target machine

The target machine is a horizontal milling centre. The dimensions of the working area of the machine are 5000 x 3000 x 3000 mm, the maximum spindle speed is 5000 rpm. The machine frame is made of cast iron. In addition to the three primary linear axes (X, Y, Z), the machine is also equipped with a working extension spindle (W) with a maximum extension of 800 mm. According to ISO 10791-1:2015 [3], the kinematic chain

of the machine can be described as H[w-(B')-X'-b-Z-Y-W-(C)-t]. The schematic kinematics of the machine is shown in Figure 1.
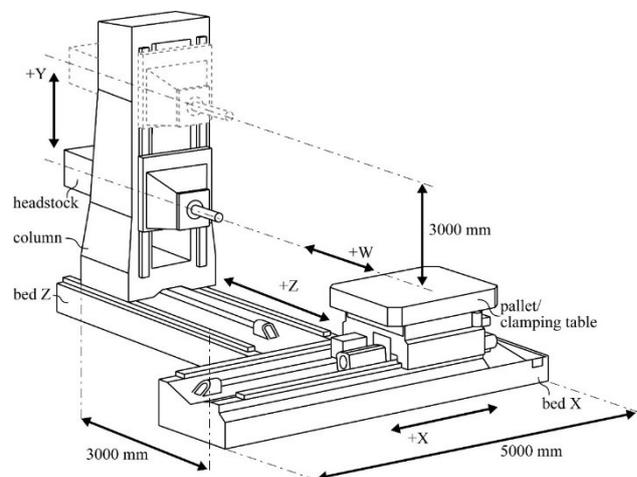

Figure 1. Kinematic description of horizontal milling centre.

For the purpose of thermal behaviour mapping, this machine is loaded only by spindle rotation at reference speed of n = 4000 rpm. During both experiments and each individual simulation, all machine axes remain fixed.

3. Description of the FEA model

To investigate the thermal behaviour of the horizontal milling centre, a FEA model was created (Figure 2).

The model consists of simplified main supporting parts: the Z-axis bed, the column and the spindle with the working-extending spindle. Because the X-axis table is structurally independent of the Z-axis bed, it was excluded from the model. The main

drive and the machine axis drive motors are replaced by mass points corresponding to the masses located in the centres of gravity of the replaced motors. The model does not include the machine fairing, electrical, pneumatic and hydraulic systems, pressure reservoirs, etc. The finite-element mesh consists of quadratic elements and contains approximately 630,000 nodes.
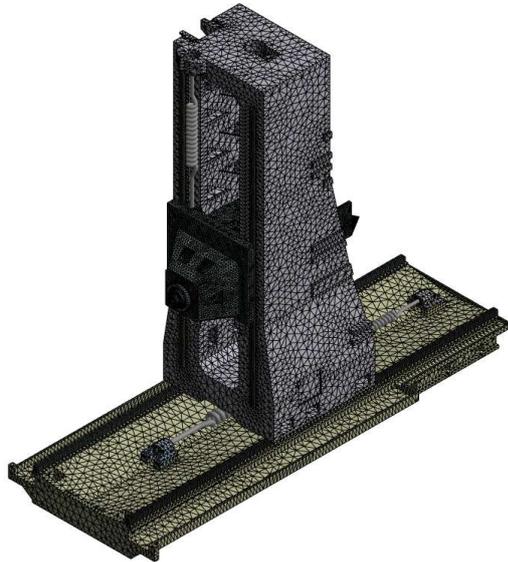

Figure 2. FEA model of the horizontal milling centre.

The column part model also includes replacements for the spindle bearing and the linear guide carriage. The stiffness of these replacements is taken from the manufacturers' catalogues. The ball screws of the Y, Z and W axis machine drives are modelled as springs of infinite stiffness.

Thermal boundary conditions are also defined. The thermal influence of the axis drives is described by applying constant temperatures to the contact surfaces of the machine structure with the drives. Heat losses generated by spindle bearings and pulley through which the spindle is driven were defined. Bearing losses (bearing power losses) loads are introduced into the model via temperature boundary conditions on bearings, using experimental data by manufacturer for different spindle extensions. The forced convective boundary condition for the aforementioned simulation setup was determined from the criterion equations for the individual spindle component surfaces. The values of the free convective boundary condition (5 - 6.5 $Wm^{-2}K^{-1}$ depending on the direction and size of the surface) are determined for all external surfaces of the machine structure. Radiation values were determined on the external surfaces of the spindle bracket.
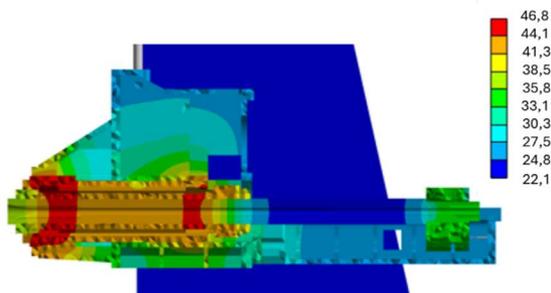

Figure 3. Simulated temperature field after 4 hours of spindle rotation in the configuration Y = 0 mm and W = 0 mm.

This model can be used to simulate the temperature field and its changes during loading by rotating the spindle in different configurations of the machine Y and Z axes (W - extension of the working spindle in the direction of the Z axis). Figure 3 illustrates an example of the simulated temperature field after 4 hours of loading in the configuration Y = 0 mm and W = 0 mm.

The result of the FEA model is a displacement field in various configurations of the Y and Z machine axes depending on the specified boundary conditions. Figure 4 shows the corresponding displacement field after 4 hours of spindle rotation in the configuration Y = 0 mm and W = 0 mm.
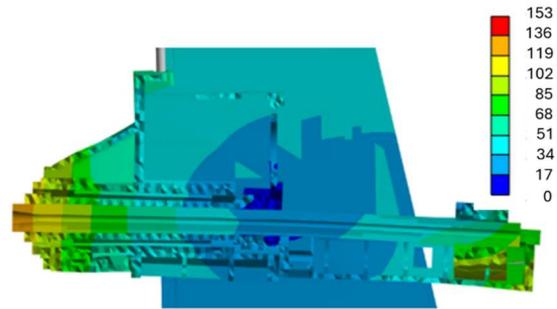

Figure 4. Simulated displacement field after 4 hours of spindle rotation in the configuration Y = 0 mm and W = 0 mm.

4. Simulation of machine configurations

Given the above ranges of the Y and W machine axes, a network of 18 points in the YZ plane was created in which simulations were performed. This network is shown in Figure 5.
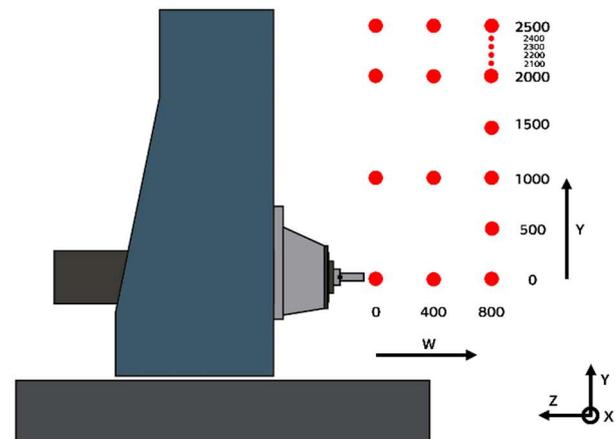

Figure 5. Network of analysed points in the YZ plane.

Spindle rotation at constant speed of n = 4000 rpm is the only load considered in simulations.The displacement of the tool centre point (TCP) in all machine axis directions is evaluated for 4 hours, when the thermomechanical system stabilizes.

The most significant error is in the Z-axis direction. However, in this direction, there is a minimal change in error depending on the change in position in the Y-axis. Conversely, in the Y-axis direction, the error exhibits dependence on the Y-axis position, accompanied by evident variations in the deformation trend over time at different Y positions. In the lower half of the Y-axis range, the error increases slightly even after the relative stabilization of the thermomechanical system (after approx. 1.25 hours). Roughly in the middle of the Y-axis range, the displacement curve remains constant after the system has stabilized. In the second half of the Y-axis range, the displacement curves show a downward trend after

the system has stabilized. Due to this behavior, the paper will focus only on displacement in the Y-axis direction.

With increasing extension of the working spindle in the W-axis, the displacement changes only linearly, so this paper deals only with the maximum extension W = 800 mm.

The result of FEA (simulated displacements) for each machine axis at different Y-axis and W-axis positions are shown in Figure 6.
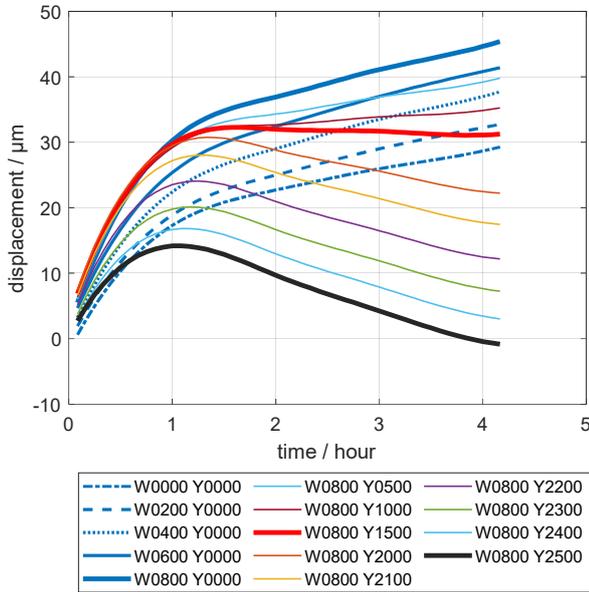

Figure 6. Simulated displacements for each machine axis.

5. Definition of measurement points

Based on the simulations performed, five measurement points were defined, which are shown in Figure 7. Points 1–3 were chosen to verify the effect of W-axis extension (Z direction), while points 3–5 were selected to verify the influence of the Y-axis position at maximum W extension. These points were selected to cover both extreme and intermediate positions within the workspace, ensuring that the most critical trends in thermal behaviour are captured with the minimum number of experiments.
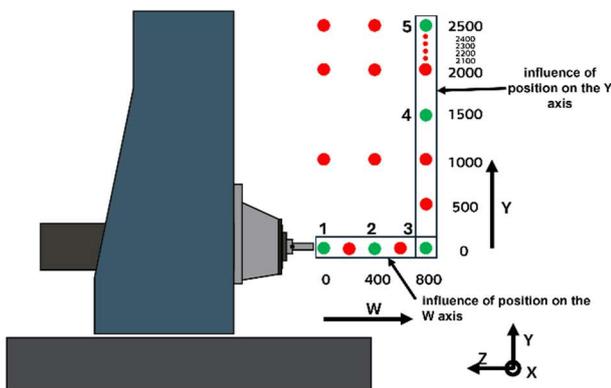

Figure 7. Selected measuring points (green).

Experiments were carried out under the same conditions as the analysis using the FEA model. Measurements were taken during loading at a constant speed of n = 4000 rpm for a period of 4 hours. There was no change in position of the machine axes during the measurements. A comparison of the simulation results and measured data is shown in Figure 8.

The comparison shows that the absolute values of the measured deformations do not correspond

to the simulation results. However, the simulations can be used to define the trend of thermal behavior in individual positions, which corresponds to the measured behavior with a certain degree of inaccuracy. The observed discrepancies are caused by additional influences acting on the machine. For the purpose of an initial analysis of the machine's thermal behavior and for defining critical measurement locations in the workspace, these discrepancies are considered negligible.
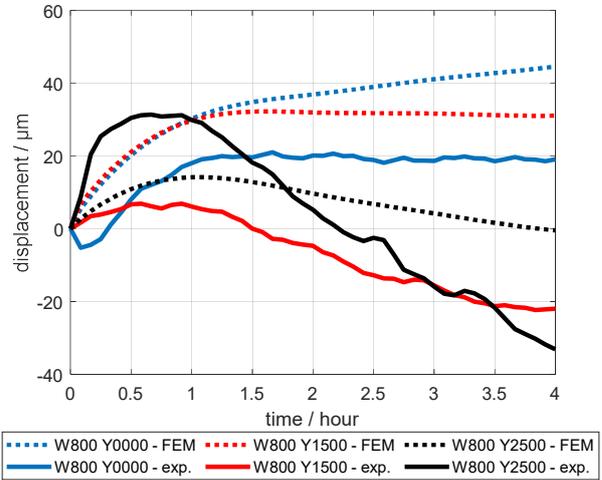

Figure 8. Comparison of measured and simulated displacements.

6. Analysis of inputs into the FEA model

One of the key inputs into the FEA model is the power loss of the front and rear bearing groups of the spindle. These losses are introduced into the model via the temperature boundary conditions on the bearings, using experimental data by manufacturer for different spindle extensions. To verify the suitability of this replacement, simulations were also performed using bearing power loss calculated analytically from spindle speed and bearing parameters; details of the calculation method are given in [4, 5]. The comparison of the three data sets (measured displacement, FEA with measured bearing temperatures, and FEA with calculated bearing power loss) is shown in Figure 9. To facilitate comparison of the trends, the curves were normalised to relative scales. The individual colours correspond to three critical positions in the machine's working space. The solid line denotes measured displacement, the dotted line denotes simulated displacement using an FEA model with measured temperature input on the spindle bearings, and the dashed line denotes simulated displacement using an FEA model with calculated power loss on the spindle bearings.
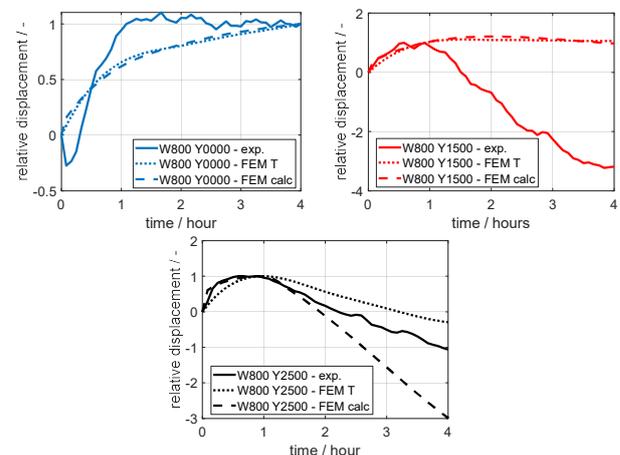

Figure 9. Comparison of the simulation results and measured displacement in relative scale.

The comparison indicates that both methods provide very similar results. Thus, using measured temperatures as inputs for the FEA model appears to be a suitable and practical solution. This confirms that the simplified modelling of spindle bearing heat sources is sufficient for the purpose of defining thermal error trends.

Nevertheless, the comparison also reveals a certain shift in the defined trends within the Y-axis. The simulation in the lower half showed an increasing shift trend. However, the measured data show a constant trend, which was observed in the simulated behavior only up to about half of the Y-axis range. Above this point, the measured data show an increasing downward trend. Therefore, a shift in the measured behaviour compared to the simulated behaviour can be observed. This shift may be caused by uncertainties in the heat source model and boundary conditions. One of the contributing factors may be the different ambient temperature, which was used as a constant in the FEA model but changed by approximately 3 °C during the experiments. Such variations may affect convective boundary condition during spindle rotation, especially for the extended spindle configuration.

Given that the presented approach is intended primarily for initial workspace analysis in situations where precise heat-source characterisation is not available, the identified shift in trends does not invalidate the method for its intended application

5. Conclusion

This paper presents an analysis of the thermal behaviour of a horizontal milling centre using an FEA model evaluated at different Y and Z positions, without the need for extensive experiments. In the Y-axis direction, the error shows the most significant dependence on the Y-axis position. The result is the definition of different behaviour trends within the range of this axis. The analysis leads to a reduction in the number of measuring points from 18 to 5 representing significant savings in time and experimental effort in the subsequent experimental phase.

Further research will focus on developing a mathematical compensation model based on the measured data valid throughout the machine's working space. The analysis described in this paper suggests that a single global compensation model may not be sufficient; instead, the workspace may need to be divided into several zones, with a dedicated sub-model for each. A switching or interpolation strategy between these models will likely be required when moving within the workspace. The main limitation of the present approach is the simplified treatment of boundary conditions, particularly the assumption of constant ambient temperature, which can deviate during experiments and influence convective effects. Nevertheless, the method provides a practical and efficient basis for reducing the scope of experimental analysis while preserving the essential trends of thermal behaviour. This demonstrates that FEA can be effectively used as a front-end tool to guide experimental design and accelerate the development of thermal error compensation models.

Acknowledgement

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