SENSITIVITY ANALYSIS AND VALIDATION OF A THERMO-MECHANICAL MODEL OF A SPINDLE

Václav Marek

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Abstract

Displacements of milling centres caused by thermal load is a vital issue of a present research. The Final element analysis, the computed fluid dynamic and other predicting tools are strong sources of information about thermo-mechanical behaviour of a machine. Validation and verification of a solution is required as many factors and coefficients influence the solution. The presented study describes the thermo-mechanical behaviour of a spindle of a milling centre. There is demonstrated the thermo-mechanical model and the verification is shown. Simulated data are compared with the measurement. Data generated by simulated options are processed. The goal of this processing is a sensitivity analysis of a thermo-mechanical system. Compensation of thermal displacements of machining spindles is currently a requested issue. Presented sensitivity analysis might help with identification of a critical area of a heat load. Improvements in these areas increase symmetry of warming-up, cooling and other heat distribution parameters. These thermal parameters influence structural displacements. Improvements decrease negative effects of heat loading, such as thermal dilatations and angular deviations. This approach leads to increase the quality and the performance of machining.

Keywords: final element analysis; thermal displacements; spindle; validation; verification

1. Introduction

The production accuracy of machine tools is effected by the thermo-mechanical behaviour of the components and structural design. Heat distribution and temperature is changed in time due to the time-dependent internal and external heat sources [3], [7], [10]. Generated temperature differences causes thermal induced displacements of the machine structure. These displacements degrade the positioning accuracy and resulting geometrical deviations of manufactured workpiece [6]. Thermal-induced errors might cause up to 70% of all production errors [1]. Precise design with respect to the thermo-mechanical structure could increase the production quality of a machine tool and it increases the accuracy of the machining. Creation of compensation algorithms is based on thermo-mechanical studies too [9].

For the prediction and the simulation of the complex thermal behaviour are required powerful tools. The complex thermal behaviour consists of heat source locations, intensities, thermal resistances and heat transfer coefficients. Due to the nonlinear thermal deformations, nonlinear modelling approaches are usually used [5]. Methodology of the complex thermo-mechanical model definition is the aim of this paper.

1.1. Neural web and AI thermal displacement compensation

Neural webs and AI principles are perspective trends in a prediction of the thermal induced errors. [5], [11] High precision, accurate CFD/FEM simulation at the beginning of a design is an expensive step with low-value results. The goal of the improvement is a combination of a prediction by a macro-element calculation and a simulation by CFD/FEM. The Macro-model computation provides very fast predictions of a thermal transfer. It is a significantly faster computation. It is not a full substitution of the CFD simulation in this context. It quickly provides basic data for the predictions in the design process. It can be seen in Figure 1 [4], [11].

Non trivial task of derivation displacement by thermal measurement can be solved by artificial intelligence. Application of machine learning is already effective for high accuracy machine tools. AI technology is able to process many information about dilatation effect based on heat load. Algorithm based on machine learning is able to predict displacements in many different cases [13].

Fig. 1. Computation with super-element predictive computation

2. Heat sources identification

Identification of heat sources is the first issue for prediction of heat load. Many studies describe base characteristics of heat sources, which are included in machine tools [2], [4]. A precise definition of heat sources is vital like a definition of boundary conditions for structural analysis [5]. Heat sources can be sorted by position to internal and external. Gears, bearings and electric motors are the main internal heat sources of machines. Sun light, convective heat flows and powerful lamps are the external sources which effect the machine structures. Characteristics of these sources are based on empirical measurements. [8] Figure 2 describes the base identification of heat sources in the spindle case. Equation 1, 2 show examples of base statements of heat loads.

Fig. 2. Heat sources in the spindle case

2.1. Heat generation in the ball bearing [14]

$$\dot{Q}[W] = 1.05 \cdot 10^{-4} \cdot F \cdot r \cdot n \cdot \mu \quad (1)$$

2.2. Heat generation in gears

$$\dot{Q}[W] = \mu_o \cdot P \quad (2)$$

where F is preload, r is outer radius, n is revolutions, μ is bearing coefficient, μ_o is overall efficiency, P is performance.

3. FEA thermo-mechanical model

A precision of FEA model is a significant factor for an accuracy of a prediction. Determination of heat sources and heat transitions between bodies are the main factors of a computation. Figure 3 shows the solution setup of the computation. The heat transfer coefficients between bodies, sources of heat load and convection coefficient are defined [7], [3].

Fig. 3. Thermo-mechanical model, NX12 Simcenter setup

3.1. Modelling of a heat generation in the bearing

The study is focused on spectre of revolutions between 600 rpm and 4,000 rpm. Presumed generated heat loads in bearings are written in the Table 1. The ball bearing FAG DLR with outer diameter 360mm is defined: total surface area of balls 79150mm². Simplified model is designed, 76608mm² surface area for convection, 5138mm² for contact; heat transfer coefficient 2500W/m²K for surface contact, forced convection area coefficient 30W/m²K [7]. Due to the presumed heat conduction in the shaft around 30% [12], heat permissivity of a bearing is presumed 8,5W/K. Numerical simulation indicates that 280W of generated heat power is distributed out of the bearing by outer rings to housing and shaft, 120W of heat is cooled by convection, based on heat transfer coefficient 30W/m²K and flow velocity of rotating air 30m/s [10].

Revolutions [min ⁻¹]	Preload [N]	Heat load / bearing [W]	Set of 2 bearings [W]
1000	3000	164	328
1250	3000	204	408
1500	3000	245	490
1750	3000	286	572
2000	3000	328	656

Table 1. Heat load of bearings

Figure 4 shows solution of heat generation and transfer in the idealized bearing housing. Results of the simulation are comparable with the bearing measurement. Balls in the bearing are simplified, heat generation and convection cooling are defined. These results determine definition of bearing heat load in the overall thermo-mechanical structure. Real simulation is shown in the Figure 4, on the right side. Real simulation indicates heat loading of the rings, it is able to define convection with better accuracy [15] Including the real model of bearing into thermo-mechanical structure of overall spindle case is not perspective. It is because of high computational requirements, quantity of boundary condition and limits of the verification and validation. Computation of the real transient thermal simulation of the bearing for 1minute running took 60 hours (Xeon 3,4GHz, 8cores).

Fig. 4. Thermal model of a bearing, simplified on the left, real on the right

4. Sensitivity study

The thermo-mechanical model of the spindle unit was tested by parameter fields. Parameters can be seen in the Table 2. Displacement of a tool clamping was analysed, depending on parameters. First set of parameters shows variation of loading revolutions. Precision of a model is indicated by presumed dilatation dependent on loading revolutions. The second set shows effect of an isolation (heat transfer coefficients – htc) of a spindle body and inner bearing rings. Isolated bearing achieves higher temperature, but average temperature of a spindle body is lower, which means dilatation reduction. The third set represents data for comparison with measurement.

		z0	Z1	Z2	z3	z10	z11	z12	z13	z20a	z20b	z20c
Bearing 12	w	300	400	500	600	300	300	300	300	300	600	800
Bearing 34	w	300	400	500	600	300	300	300	300	300	600	800
Bearing 5	w	150	200	300	400	150	150	150	150	200	400	600
Bearing 6	w	200	300	500	700	200	200	200	200	200	500	700
Final gear	w	300	400	400	400	300	300	300	300	200	300	400
F.g. case	w	100	150	200	250	100	100	100	100	150	200	250
Tool	w	120	120	120	120	120	120	120	120	0	0	0
Htc	w/m ² k	0	0	0	0	400	600	800	1000	2000	2000	2000
Revolutions	n/min	1000	1250	1500	2000	1000	1000	1000	1000	1000	2000	2500
Time	min	0-60	0-60	0-60	0-60	0-60	0-60	0-60	0-60	0-30	30-60	60-120

Table 2. Options of a sensitivity study

The thermo-mechanical simulation of a deformation corresponds with the experimental measurement. Simulation in the Figure 6 shows crucial dilatation of a boring spindle. Figure 5 shows the final dilatation with Z13 options. Results of thermal calculation by NX Simcenter solver are exported to structural transient solver NX Nastran and solved with respect to a contact conditions, mechanical structure and preloads [8], [14].

Fig. 5. Results of a thermo-mechanical simulation Z13, dilatation in direction Y-axis

Processed dilatation results identify crucial nodes and elements, which can reduce dilatation after optimization. Option z10-z13 use thermal isolation between shaft and bearing. Isolation decrease dilatation effect, as can be seen in the Figure 6. Option z0-z3 confirm dependency of dilatation on revolutions. Other simulation sets need to be solved to find the best way for optimization. Bearing heating is strongly dependent on the quality of an assembly on the other side. It is almost impossible to predict this factor. Negative change of parameters leads to depreciation of the optimization. Measurement and verification is significant because of that.

Fig. 6. Graphic results of a sensitivity study

5. Validation and verification

Test running was performed by test rig. Running speeds are 1000/min for 0-30 minutes, 2000/min for 30-60 minutes, 2500/min for 60-300 minutes. Machine run without any process heat loading. Temperature measurement option can be seen in the Figure 7. Results are able for sensors T1, T2, T4. These sensors show temperature of the main spindle bearings. Thermocouples measure body temperatures of housing, actually close to outer ring of bearing.

Fig. 7. Measurement options and values

The comparison of the temperature simulation and the measurement can be seen in the Figure 8. Measurement also includes infra-camera pictures which shows thermal effect area around bearings. The simulation responds to the measurement in the time gap 0-7200 seconds.

Fig. 8. Heat load of bearing L12 (T1)

6. Conclusion

The simulation and the prediction of a thermal load in the machine tools are high value source of information. The verification of these simulations is required due to wide spectre of used coefficients. Lack of the precise models of these structures motivates research. Described thermo-mechanical model of a spindle was validated by comparison with temperature measurement. The sensitivity study also shows critical nodes of this thermo-mechanical system. The methodology of the thermo-mechanical model definition was stated. The research will continue through detailed verification with detailed boundary conditions. A convection to close environment and specification of bearing cooling by air-oil lubrication will be examined. Gained data and optimized thermo-mechanical models will be applied for the solution of a problem of a spindle dilatation compensation. The solution is strongly requested by machine tool producers. Approach in this area brings optimization of the thermal behaviour of the machine tools and allows increase of the overall machining performance. Described progress induce request of development in affiliated parts of the system.

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