Journal of Materials Processing Technology 162–163 (2005) 615–621

Journal of Materials Processing Technology

www.elsevier.com/locate/jmatprotec

High-speed precise machine tools spindle units improving

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Abstract

A hybrid high-speed precise machining centre headstock model based on two computation methods: the finite element method (FEM) and the finite difference method (FDM) is presented. The model allows one to calculate precisely the headstock's indices on the basis of which its optimal operating characteristics can be determined. The presented modelling methods allow one to evaluate a design from thermal, stiffness and durability points of view. By way of illustration, the behaviour of three machining centres headstocks with: an electrospindle on rolling bearings, a conventional spindle and an electrospindle on aerostatic bearings are modelled using the hybrid model. The bearing units and cooling system influence on machine tool preciseness has been described.

Keywords: Machine tool; Modelling; Spindle

1. Introduction

The development of modern machining centres proceeds towards very high machining speeds and precision. At the same time the static, dynamic and heat loads of machining centres become increasingly complex. Consequently, the creation of an accurate computational model is a very difficult and time-consuming task. Precise modelling is the basis for design improvement through the analysis and optimisation of the machining centre's operational properties and the minimization of its prototyping time and cost [1–8].

Hence the creation of machine tool models becomes highly important. Effective models are holistic hybrid models, which accurately represent all the involved phenomena and their effects. The thermal phenomena, which occur in such complex machine tools as machining centres, are particularly difficult to model. It is necessary to search for accurate functions and procedures to describe the energy losses, the geometry, the heat exchange and the dynamics and nonlinearity of the involved phenomena. To achieve a high model precision it is often necessary to investigate in minute detail the behaviour of the headstock's electrospindles.

In this paper an idea of a holistic hybrid high-speed machining centre headstock model is presented and illustrated using as examples three machining centre headstocks differing in their design.

2. Idea of hybrid model

The thermal state of high-speed precision machining centres is affected by a whole range of factors, such as machine tool operating conditions, energy losses in kinematic system elements, temperature distributions and thermal displacements of machining centre elements and assemblies, and the interrelationships between them. Power losses are a function of the machine tool operating conditions and depend on the design features, the geometrical structure and the material properties of the particular components and assemblies, the lubricating media and methods, the cooling, etc. Power losses result in time-variable temperature distributions, which in turn affect, among others, the running clearance in the bearings, the viscosity of the lubricating medium and the surface film conductance. Consequently, the power losses change. The interactions between the factors result in a complex, dynamic and nonlinear state of thermal equilibrium characterized by continuous changes in power losses, temperatures and displacements. The hybrid model of the

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doi:10.1016/j.jmatprotec.2005.02.149

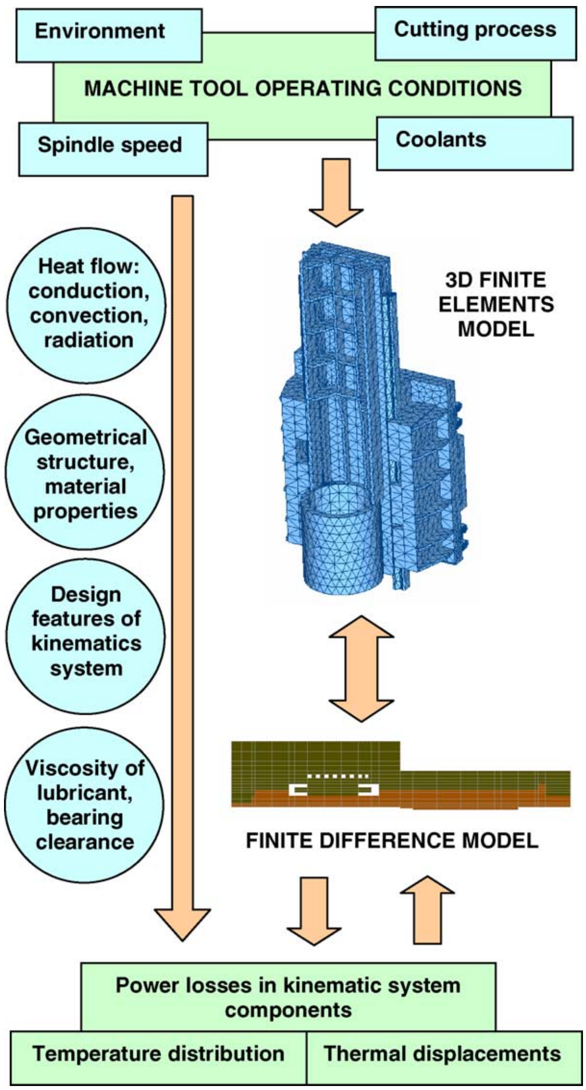

Fig. 1. Schematic of hybrid model of machining centre headstock.

high-speed headstock takes into account the above interactions (Fig. 1) and makes possible a highly precise partial and full analysis.

The model is based on two computation methods: the finite element method (FEM) and the finite difference method (FDM). The components of the headstock's body are modelled by FEM using tetrahedral elements. The axially symmetric headstock assemblies (the spindle with bearings, the rollers, the gears, etc.) are modelled by FDM using ring elements, which in a natural way render the geometry and the phenomena that occur in the assemblies. The two computational submodels are integrated into one holistic model whereby it is possible to model very precisely the thermal phenomena in the axially symmetric kinematic system components, such as rolling bearings, toothed gears, belt transmissions, motors, electrospindles, etc., taking the natural proportions of the pattern of the generated heat fluxes into account. In this approach spindle assemblies are treated as modules, which can be modelled separately and integrated into the headstock body whose geometry does not change.

The hybrid model owes its high accuracy to the precise modelling of the heat sources and flows and to the fact that all the complex interactions are taken into account, e.g. when modelling rolling bearings their design (the type of bearings, the dimensions, type and number of rolling elements, the material of the bearing elements), the rotational speed of the spindle, the load, the lubrication conditions, the elastic and thermal properties of the bearing sets' components, the forced cooling and the bearings' lifetime can all be taken into account.

3. Hybrid model of electrospindle

3.1. Model of electrospindle

The electrospindle model is based on the FEM and FDM (Fig. 2). Plane and spatial finite elements and cylindrical finite difference elements are used. The hybrid model is particularly useful when the temperature distributions resulting from the heat generated in the assemblies are to be calculated and it allows one to:

- freely shape the geometry of the headstock's housing;
- avoid the computation of body deformations when calculating temperature distributions—owing to the analytical models for determining power losses in the bearing sets and in the motor;
- avoid excessive mesh density within bearings when determining running clearance; and
- store models of typical bearing set designs in the system's database.

Selected analyses carried out for the electrospindle by means of the hybrid model are presented below.

3.2. Bearing set

Spindle bearing set modelling requires special attention since accurate evaluation of power losses is not possible unless the influence of the set's temperature and deformations (operating clearance) is precisely known. Power losses are

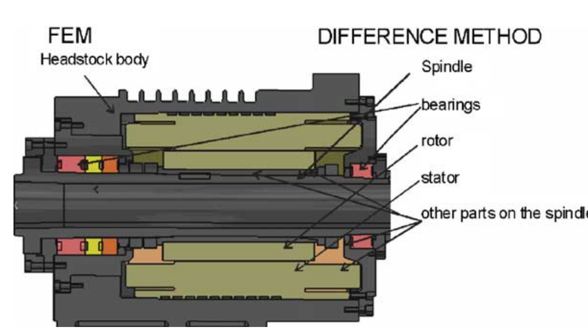

Fig. 2. Electrospindle model.

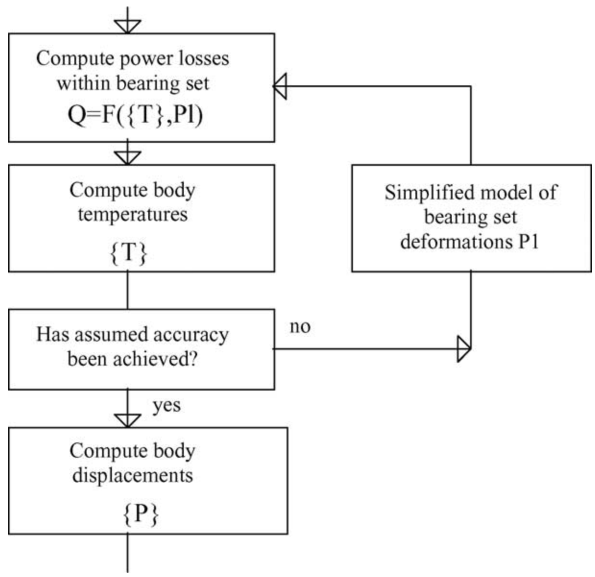

Fig. 3. Procedure for determining temperatures and thermal displacements.

determined from temperature relations while bearing losses are computed from a simplified model consisting of a body wall fragment, the spindle and the bearings.

A sequence of steps needed to compute temperatures and thermal displacements in machine tools is shown in Fig. 3.

Bearing set deformations $P1$ are needed to determine running clearance L, which in turn is needed to calculate the power losses and the bearing wear-out time. Working clearance L it is after mounting clearance L_p, changed by deformation of bearing assembly components. The final clearance value for a given design depends on the fit between the bearing set's components, the bearing preload and the bearing set thermal deformation.

3.3. Initial clearance in bearing set

Owing to the recognition of clearances in different service conditions, proper fit values (assuring the optimal operating characteristics for the bearing set and the headstock) can be selected. Precise initial fit values are also needed for the computation of running clearances. Clearance $W1$ between the bearing and the housing and clearance $W2$ between the bearing and the spindle (see Fig. 4) are computed.

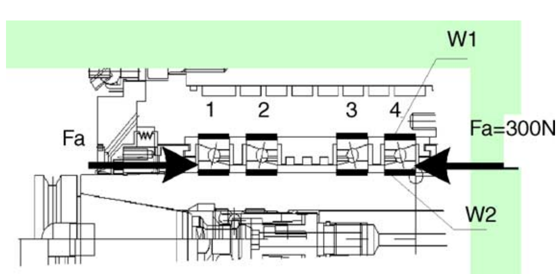

Fig. 4. Clearance $W1$ between bearing and housing and clearance $W2$ between bearing and spindle.

Table 1
Tolerance analysis of spindle front bearing set

Tolerance of housing bore diameter	Tolerance of spindle outer diameter	**Bearing type** 4x7013CEHCP4AQBC sets
Φ100H4	Φ65j4	**Fits** H4 / j4
L_{max}=18µm	L_{max}=4µm	Preload for bearing set
L_{min}=0µm	L_{min}=-11µm	Fa= 300 N

Most probable W1 and W2 for considered fits

Clearance before/after preload Fa		Radial clearance after mounting			
$W1_{init}$ /$W1$ Housing/ Bearing	$W2_{init}$/$W2$ Bearing/ Spindle	Bea-ring 1	Bea-ring 2	Bea-ring 3	Bea-ring 4
6.00 / 2.92µm	-6.00 / -6.34µm	-7.49	-7.49	-7.49	-7.49

L_{max}: maximal clearance fits for bearing class P4; L_{min}: minimal clearance fits for bearing class P4; $W1$: clearance between bearing outer ring and housing; $W2$: clearance between bearing inner ring and spindle. Example: For considered fit H4/j4 the most probable initial (before preload) clearances are: $W1_{init}$ = +6 µm, $W2_{init}$ = −6 µm. After preloading with 300 N, $W2$ and $W2$ are: $W1$ = 2.92 µm, $W2$ = −6.34 µm.

The procedure for determining clearances $W1$, $W2$ and L in service conditions is shown below for bearing set 4 × 7013. In order to estimate clearances $W1$ and $W2$ one must determine their values after preloading as shown in Table 1.

In the same table one can find radial clearance after mounting L_p (clearance resulting from assembly and bearing preload). Since there are many possible clearance values for the housing bore diameter and spindle outer diameter tolerances, the most probable clearances were adopted as the basis for computations. This means that in practice there may be cases differing from the above. The ranges of variation in post-assembly clearance in the bearings for the adopted fits are shown in Fig. 5. The middle value in the grey range is the most probable value.

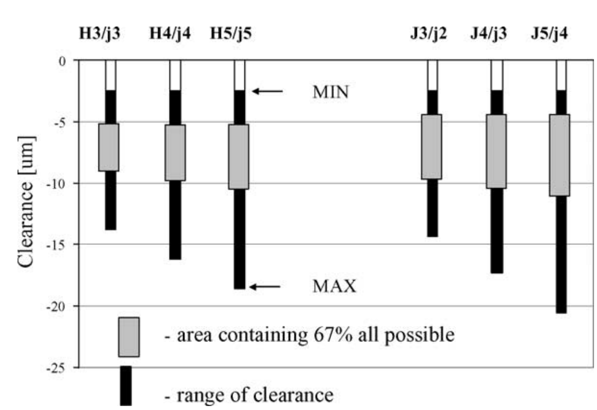

Fig. 5. Ranges of variation in post-assembly clearance depending on fits.

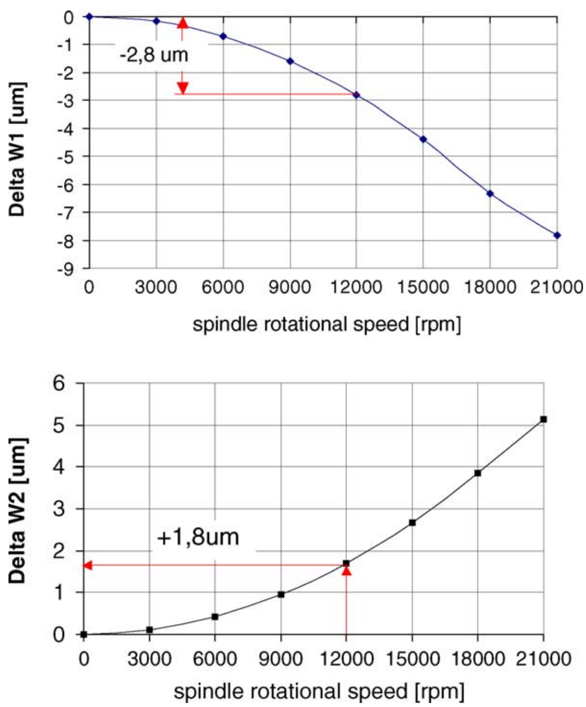

Fig. 6. Ranges of variation in clearance $W1$ and $W2$ depending on spindle rotational speed.

3.4. Change in clearance caused by rotational speed

Under the influence of rotational speed a sharp change in values $W1$ and $W2$ (caused by the centrifugal forces in the bearings) occurs. As a result, running clearance L changes too. For the provided example changes in the two clearances: delta $W1$ and delta $W2$ are shown in Fig. 6. As the rotational speed increases so does the centrifugal force, which reduces the pressures in the spindle-bearing inner ring connection while increasing the pressures on the bearing's outer ring.

3.5. Running clearances

Except for the initial clearance, running clearance in the bearings depends on changes in the dimensions of all the bearing set components caused by the distribution of temperature. The bearing set components whose longitudinal and radial deformations determine running clearance are shown in Fig. 7.

LH: length of bearing set in housing
LS: length of bearing set on spindle
ΔLH(t): thermal increase in length LH
ΔSLH(t): thermal expansion of ring set $\Sigma zi + \Sigma qi$
ΔLS(t): thermal increase in length LS
ΔSLS(T): thermal expansion of ring set at spindle $\Sigma wi + \Sigma pi$

From the conditions of the equilibrium of forces in the bearing set, taking the forces generated by the radial and axial changes in the dimensions of the bearing set components into account, the running clearance in each of the bearings could be determined.

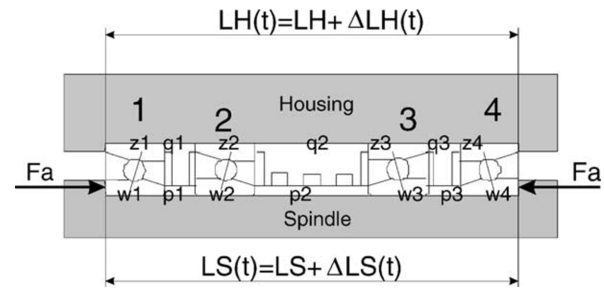

Fig. 7. Simplified model of spindle bearing set for calculating its deformations.

Changes in running clearance in the particular bearings for different spacer lengths are shown in Fig. 8. The differences in clearance in the bearings are due to the different temperature distributions in their sections (see Fig. 9) and the conditions of the equilibrium of forces in the bearing set.

3.6. Power losses in bearing

If the load (loading with external forces, e.g. machining forces, and with internal forces generated by negative running clearance) and the lubricating conditions (the kind, viscosity

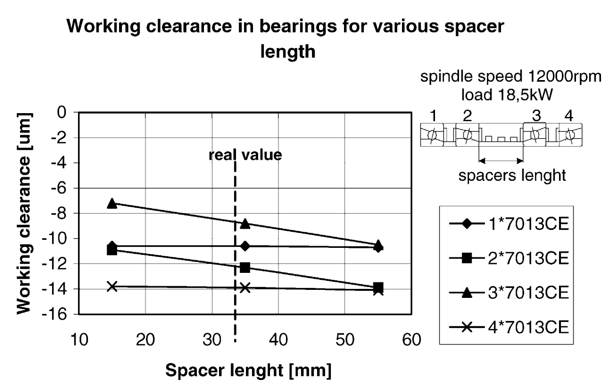

Fig. 8. Running clearance in bearings vs. spacer length.

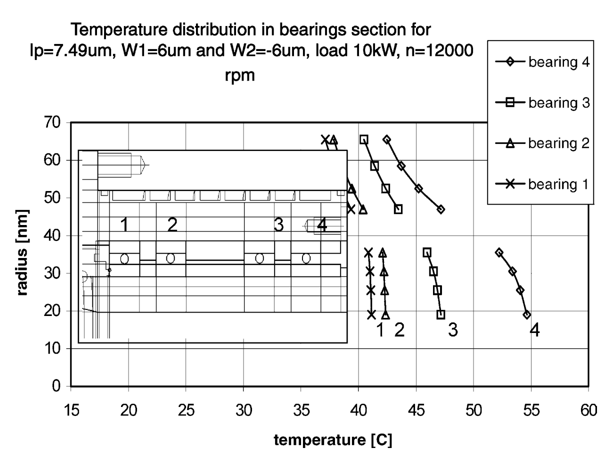

Fig. 9. Temperature distribution in bearing sections.

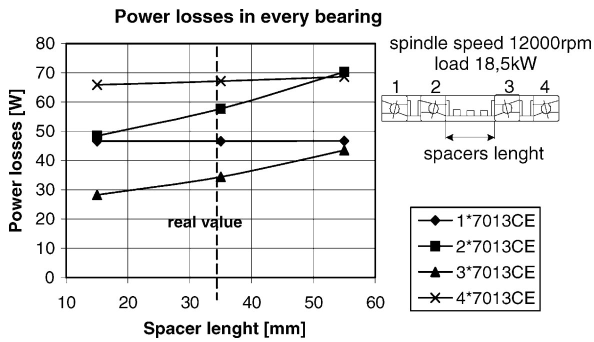

Fig. 10. Power losses in bearings vs. spacer length.

and amount of lubricant) are known one can determine the power losses in each bearing as shown in Fig. 10.

3.7. Bearing lifetime

Another important function of the proposed system is the calculation of the durability of the bearings working in the bearing set. Bearing lifetime is calculated assuming that the bearings are under equivalent bearing load Fe resulting from radial running clearance. According to Fig. 11, the lifetimes of bearing 4 and bearing 2 are the shortest in extreme operating conditions (high speed and heavy load).

Lifetime values are highly sensitive to clearance values. This means that the accuracy of the calculated lifetime depends greatly on the accuracy of the computing model and on the quality of parts fitting and assembly.

The bearings' thermal loads determined by means of the hybrid model constitute the boundary conditions for the 3D FEM model.

4. Hybrid model of machining centre head-stock

4.1. Machining centre with 8000 rpm spindle

A machining centre headstock with an 8000 rpm spindle was modelled. The spindle is equipped with angular ball bear-

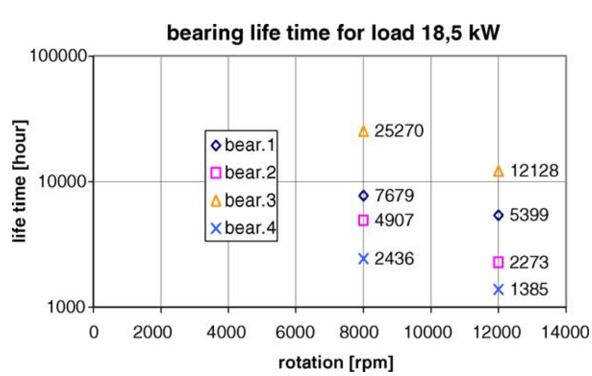

Fig. 11. Lifetime of bearings under maximum load.

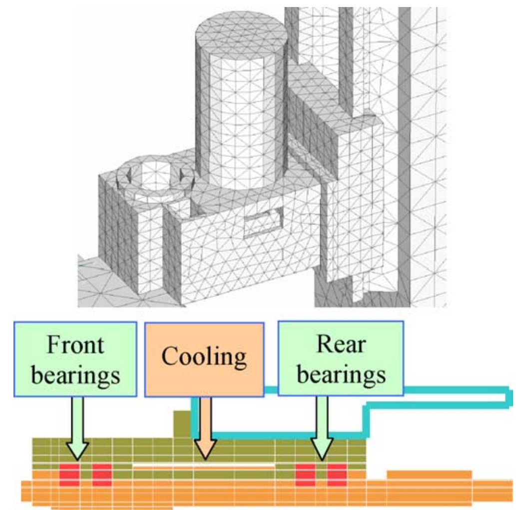

Fig. 12. Model of machining centre headstock.

ings with reduced diameter steel balls or with hybrid angular ball bearings with ceramic balls. The spindle is driven by a motor via a belt transmission and it is intensively cooled by a forced water or oil flow. The hybrid model of the machining centre headstock (Fig. 12) consists of an FEM based headstock body model and an FDM based spindle assembly model.

The holistic hybrid model of the machining centre headstock integrates several submodels of the thermal phenomena, which occur within the headstock, such as:

- a model for computing the amount of heat generated in the rolling bearings;
- a model of the heat flow in the spindle assembly;
- a model of the forced cooling of the spindle bearings; and
- a model for calculating the amount of heat generated in the belt transmission driving the spindle.

The hybrid model was used to analyse the effect of the spindle bearings on the thermal behaviour of the machine tool. An example of such an analysis is shown in Fig. 13, where temperatures $T1$–$T3$ at three points of the machining centre headstock are compared. The analysis gave the designers an insight into the effect of the bearings' design and their operating conditions.

The aim of the analysis was to check whether the thermal behaviour of a machining centre prototype was correct. For the assumed proper thermal load conditions the thermal behaviour of the hybrid model of the machining centre was found not to be wholly consistent with the experimental results. Divergences were found in the belt transmission area (see values $T3$ in Fig. 13). A hybrid model analysis of different possible cases of belt transmission operation indicated what changes had to be made in the belt transmission design and assembly.

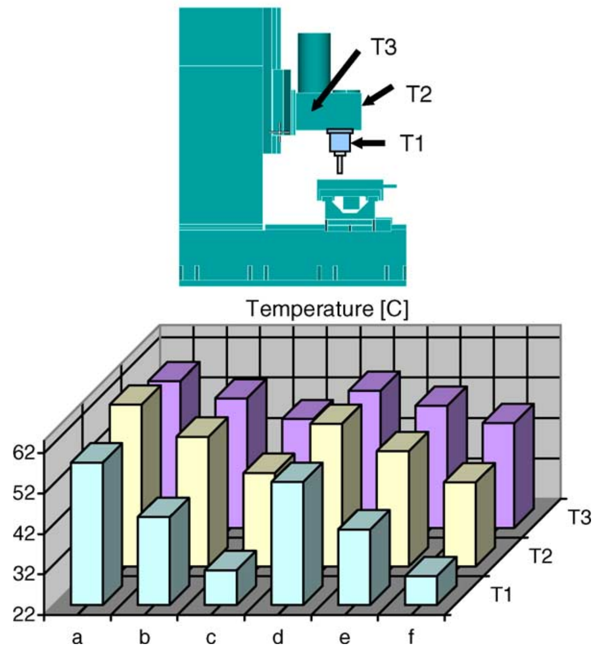

Fig. 13. Temperatures at three points of headstock (8000 rpm, steady state, ambient temperature 22 °C): (a) bearings with steel balls; (b) bearings with steel balls + cooling with oil; (c) bearings with steel balls + cooling with water; (d) bearings with ceramic balls; (e) bearings with ceramic balls + cooling with oil; and (f) bearings with ceramic balls + cooling with water.

4.2. Precise machining centre with 50,000 rpm spindle

Also a precise machining centre headstock with the spindle rotating at a very high speed of 50,000 rpm was modelled. An electrospindle with an asynchronous motor and aerostatic bearings was used. The electrospindle motor stator is intensively cooled by the forced flow of water or oil. The machining centre headstock model (Fig. 14) consists of a headstock body model integrated with an electrospindle model.

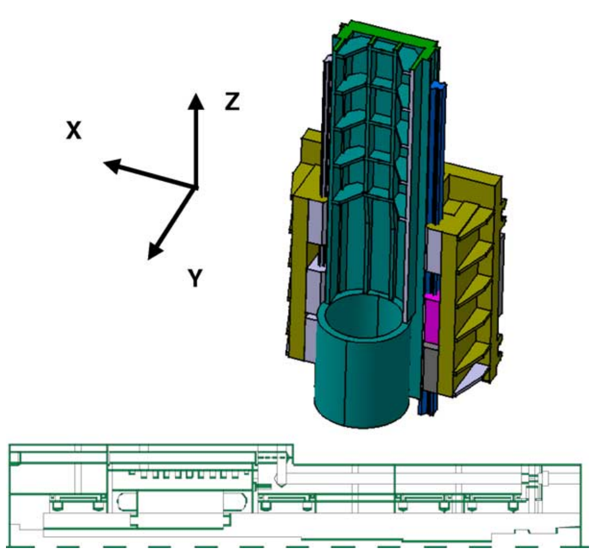

Fig. 14. Model of a machining centre headstock body and electrospindle.

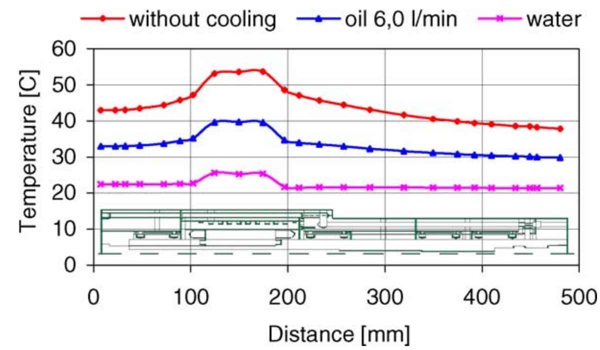

Fig. 15. Effect of stator cooling conditions on temperature of the fixed part of an electrospindle (50,000 rpm, steady state, ambient temperature of 20 °C).

Thanks to the hybrid model it became possible to integrate in the holistic headstock model the following submodels:

- a model for computing the amount of heat generated in the electrospindle motor;
- a model of heat distribution to the stator and the rotor,
- a model of the stator cooling system;
- a model for computing the amount of heat generated in the aerostatic bearings; and
- a model for computing the heat flow in the spindle assembly.

The hybrid model was used for analysing the effect of motor stator cooling on the thermal behaviour of the machining centre electrospindle. An example of this analysis is shown in Fig. 15 where the temperature of the electrospindle's fixed part is compared for three cases: no stator cooling, stator cooling with oil and stator cooling with water. The analysis showed a very strong effect of cooling on electrospindle temperature, particularly for intensive cooling with water. The results of temperature distribution and spindle thermal displacements obtained by means of the hybrid model made it possible to indicate the best solution and to quantitatively assess the improvement in the electrospindle's thermal properties. The spindle thermal displacements, precisely simulated using the hybrid model assuming water cooling, was decreased in the Y-axis from 100 to 20 μm and in the Z-axis from 110 to near 0 μm. It is a good enough result to precisely design error compensation functions.

5. Summary

Precise modelling of the thermal behaviour of high-speed machining centre headstocks is a complex and time-consuming problem. Owing to its high accuracy, ensured by the precise modelling of the main heat sources, the presented hybrid model can be successfully employed in the design of high-precision machine tool headstocks and spindle assemblies. The model has been repeatedly verified in the design of machining centres. Thanks to detailed hybrid model analyses the lifetime of the electrospindle assemblies was significantly increased while heating and thermal displacements were reduced.

References

[1] J. Jędrzejewski, W. Modrzycki, A new approach to modelling thermal behaviour of a machine tool under service conditions, Ann. CIRP 41 (1) (1992) 455–458.
[2] J. Jędrzejewski, J. Kaczmarek, Z. Kowal, Z. Winiarski, Numerical optimization of thermal behaviour of machine tools, Ann. CIRP 39 (1) (1990) 379–382.
[3] J. Jędrzejewski, W. Modrzycki, Improving machine tool accuracy using intelligent supervision model, in: CIRP International Seminar on Intelligent Computation in Manufacturing Engineering, Capri, 1998, pp. 465–470.
[4] B. Bossmans, J. Tu, A power flow model for high speed motorized spindles—heat generation characterization, J. Manuf. Sci. Eng. 123 (2001) 494–505.
[5] B. Bossmans, J. Tu, A thermal model for high speed motorized spindles, Int. J. Mach.Tools Manuf. 39 (1999) 1345–1366.
[6] S.M. Kim, S.K. Lee, Prediction of thermo-elastic behavior in a spindle-bearing system considering bearing surroundings, Int. J. Mach. Tools Manuf. 41 (2001) 809–831.
[7] T. Moriwaki, E. Shamoto, Analysis of thermal deformations of an ultraprecision air spindle system, Ann. CIRP 47 (1) (1998) 315–319.
[8] M. Weck, G. Spachtholz, 3- and 4-contact point spindle bearing—a new approach for high speed spindle systems, Ann. CIRP 52 (1) (2003) 311–316.