INTEGRATED MODELLING OF AXIAL MOTION ERROR SOURCES FOR 5-AXIS PRECISION MACHINING CENTRE WITH DIRECT DRIVES

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This paper discusses machine tool modelling, comprising the design process, degradation and repair processes and accurate identification, aimed at the effective improvement of machine tools. The importance of the holistic approach to modelling for the accurate identification of machine tool errors and the ways in which they arise and for their minimization, taking into account thermal and dynamic interactions, is highlighted. Models of the sources of errors (power losses) occurring in rotary direct drive motors and in rotary tilting table bearings are presented. Also the modelling of power losses in linear direct drive motors and in rolling guideways is described. Mostly original relations, supported by the results of studies on the modelling and computing of errors in the controlled axes of a 5-axes machining centre carried out by the authors, are used in the modelling.

Keywords
Holistic model, machining centre, direct drive, power losses, error

1. Introduction

Machine tool modelling evolves towards aiding design and operation processes to ensure the broadly understood maximum productivity of machine tools. The aim of such modelling is to assist in the accurate identification of the physical processes having a bearing on the static, thermal and dynamic properties of machine tools, the disturbances in machine tool operating parameters, and errors as well as in the minimization of errors, the taking of possible repair and self-repair measures and the prediction and prevention of critical states. Thus modelling is to aid the design of a machine tool to be characterized by the best possible service properties, and the identification of the causes of machine tool degradation in operating conditions. It also should take into account the complex interactions between dynamic processes and thermal processes and the resultant deformations and displacements affecting the precision and efficiency of the machining process.

Besides the identification of errors and critical states, their active minimization through optimization, self-repair and compensation is crucial for their reduction [Jedrzejewski 2004]. For this purpose dedicated models (the more complex, the more complex the modelled processes and the greater their dynamics) need to be created. The premises for creating such models are shown in table 1 in which machine tool degradation and repair processes have been assigned proper identification and modelling processes aiding the design and improvement of service properties and operation processes.

The degree of detail of a model depends on its role in improving the machine tool and on the requirements the latter must meet. Today an ever greater emphasis is placed on increasing precision. The number of precision and very high precision machine tools has significantly increased. Also the demand for multiaxis machine tools has grown. However, the greater the number of axes, the more difficult it is to identify spatial (especially thermal) errors and the greater the modelling complexity. It is also important to ensure that the machine tool is thermally symmetrical. The thermal requirements must be met especially by machine tools intended for the dry machining of hardly machinable materials. When modelling is to aid error minimization and compensation in controlled axes, the focus is on models of the errors occurring in these axes and on the evaluation and numerical prediction of the errors for compensation purposes. It is to these error models that this research is devoted.

2. Holistic modelling of 5-axis machining centre

According to the authors' experience, a model incorporating all the sources of thermal errors and errors due to dynamic forces, the totality of heat transmission conditions and the real-time interactions between deformations ensures the highest effectiveness of machine tool improvement.

In order to model such a complex object as the machine tool (especially a 5-axis machining centre), simplifications need to be made. Simplifications are introduced already in the modelling of geometry and phenomena (e.g. heat generation and transmission). Because of the use of external procedures for realizing (often through highly complex relations) mathematical models of phenomena, the computer programs become too complicated and time-consuming and the computations take too long. When creating a dedicated computing program or one based on a commercial FE system, several simplifications are made and then the assumptions are fine-tuned in order to obtain good agreement with the experiment. Modelling is particularly difficult when the heat sources move and affect the time- and space-variable dynamic loads.

Real processes taking place in machine tool		Virtualization of machine tool behaviour
Processes of degradation and repair	**Identification of properties**	**Processes of modelling and optimization**
Post-assembly service properties	Accurate identification of state of properties	Accurate models of properties
Disturbances	Identification of state sources and disturbance variation	Models of disturbance generation and behaviour in operating conditions
Geometry and motion errors in controlled axis	Identification of errors and their variation	Models of errors and their behaviour in operating conditions
Self-repair	Identification of repair process and properties	Self-repair modelling
Error compensation	Identification of compensation process	Predictive models of error and compensation
Generation of critical states	Prediction of critical states	Predictive models of critical states
Repair of critical states	Identification of repair processes	Models of repair processes and critical states
Permissible level of disturbances and errors	Continuous identification of state	Models of continuous error state monitoring

Tab. 1. *Identification and virtualization of processes taking place in machine tools.*

Through the use of the hybrid model (fig. 1) of the machining centre (with the heat sources determined using the individual isolated models) one can integrate the interactions between the components and the interactions taking place via the centre's support structure. A major challenge is the integration of all the impacts on the heat exchange and transmission conditions, especially in closed spaces (e.g. in the workspace). Similarly as the large body masses, such spaces accumulate heat, determining the thermal inertia (the delay in response to the heat sources). This phenomenon needs to be taken into account when identifying, e.g., errors due to a change in the ambient temperature. Many leading machine tool manufacturers try to take such errors into account. However, it emerges from the authors' comprehensive research into the problem that the latter has to be solved separately for each machine tool structure for the particular heat transmission conditions and the heat capacity of both the closed spaces and the heat accumulating masses [Mekid 2009], [Jedrzejewski 1996]. A way to reduce the effect of closed spaces is to cool/ventilate them (it is also a good way to reduce the thermal errors of the straightedges), but this an energy-intensive solution.

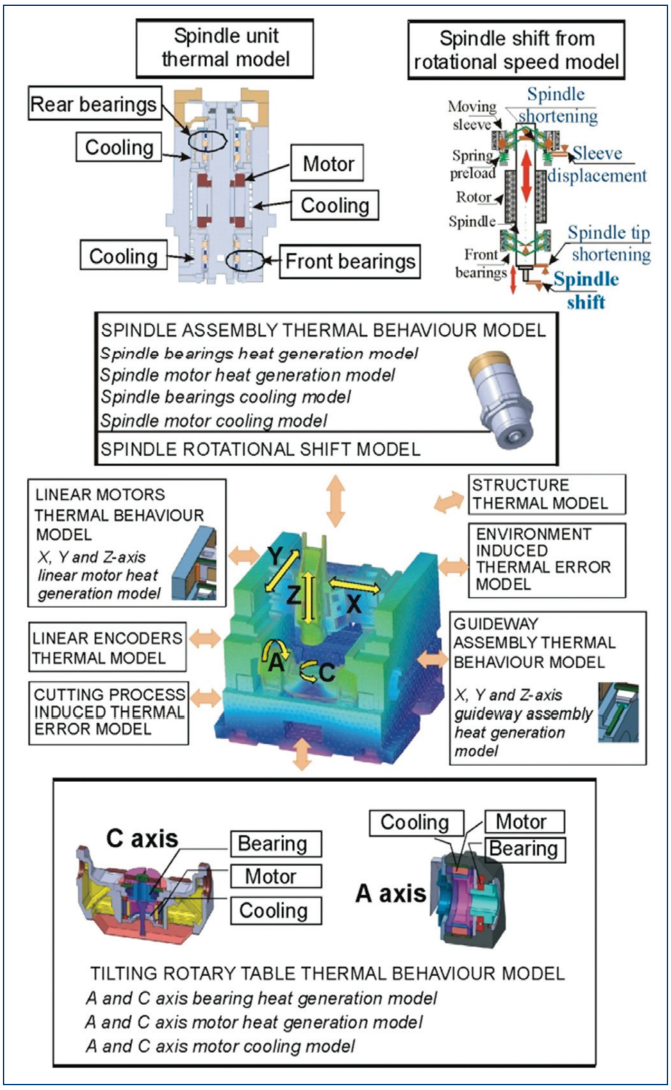

Figure 1. *Holistic model of 5-axis machining centre.*

When computing the distributions of temperature and thermal displacements for machine tool assemblies the authors use power loss models characterized by a different degree of complexity and accuracy of the description of power loss generation. Exemplary relations used by the authors are presented in tables 2 – 4.

The relations can be used to calculate instantaneous power losses, friction moments and friction forces in the main machine tool assemblies. However, they are not enough to properly describe machine tool thermal loads since most of them change in the course of operation as a result of thermally induced changes in the machine tool components' dimensions, changes in the lubricant's viscosity, changes in rotational and linear speeds, changes in accelerations and so on. If for computations one assumes constant outputs of the heat sources, one cannot expect results close to reality.

3. Power losses in rotary direct drive motors

The planned duty cycle, consisting of the starting phase, the continuous operation phase and the braking phase (fig. 2), is the basis for determining power losses in the direct motor. The torque loading the motor in the first phase and in the last phase is a resultant of the assumed maximum rotational speed, the accelerations and the moment of inertia of the moving masses, while during continuous operation it is the sum of the friction losses in the tilting table's bearings and the loads connected with the machining process and the external impacts.

Calculated quantity	Basic formulas
Power dissipation by coil	Total continuous power P_c to be dissipated by coil $Pc_{(\Theta=130^\circ C)}$, W (in catalogue data sheets) $Pc_{(\Theta=130^\circ C)} = \frac{1{,}5 \cdot Rc \cdot Ic^2}{\left(\frac{Tc}{T_{RMS}}\right)^2}$, W [ETEL 2009]
Torque	Tc – continuous torque for coil temperature of 130°C, Nm (in catalogue data sheets)
Current	Ic – continuous current for coil temperature of 130°C, Arms (in catalogue data sheets)
Resistance	Rc – resistance of windings at actual working temperature, $Rc = R_{20}\left(1 + \left(0{,}00392 \cdot (\Theta - 20)\right)\right)$, Ohm R_{20} – catalogue electrical resistance at 20°C, Ohm Θ – actual temperature of windings, °C
Equivalent torque	T_{RMS} – equivalent torque for whole cycle $T_{RMS} = \sqrt{\frac{\sum_{i=1}^{N} T_i^2 \cdot t_i}{t_{cycle}}}$, Nm T_i – torque values for step, N – number of steps t_i – step duration, t_{cycle} = complete motion time
Power dissipation by rotor	$P_{rotor} = 0{,}02\, Pc_{(\Theta\neq130^\circ C)}$
Water flow	$Fw = 0{,}0143 \cdot \frac{Pc_{(\Theta=130^\circ C)}}{\Delta Tw}$, l/min ΔT_w – water temperature difference between input and output coolant
Pressure drop	$\Delta p = \Delta p_{(catalogue)} \cdot \frac{Fw}{Fw_{(catalogue)}}$, bar

Tab. 2. *Power losses in rotary direct drive motors.*

Besides the load, the stator windings temperature and the temperature of the medium lubricating the bearings, which are closely dependent on the heat transmission conditions (and so on the motor support structure and the forced cooling conditions), determine power losses. In order to take the above interdependences into account when calculating power losses in the direct motor it is necessary to iteratively search for the real temperature of the windings, using a FE model.

The starting point for iteration should be the power calculated for the assumed windings temperature and determined torque TRMS loading the motor. The iteration process lasts until the initial temperature assumed for the motor power loss calculations is equal to the windings temperature determined from FE model simulations.

The temperature of the lubricant (grease, or more precisely its base oil) in the zone of friction between the bearing rollers and races should be determined in a similar way. The iteration ends when the temperature of the lubricating medium assumed for the

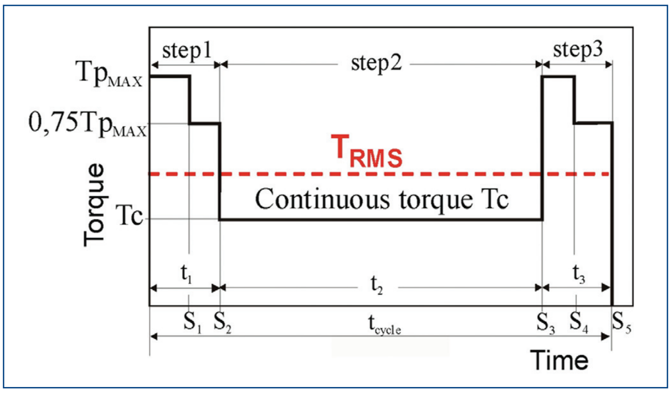

Figure 2. *Determination of equivalent motor load for assumed duty cycle.*

hydrodynamic moment of friction is equal to the average temperature of the bearing rings, obtained from FE model simulations.

4. Power losses in tilting rotary table bearing

In rotary tables and in tilting tables two types of roller bearings (fig. 3) are used:

- conventional bearings with two rows of rollers carrying axial loads and one row of rollers carrying radial loads;
- cross bearings with two rows set at an angle of 45°, each of which carries both axial and radial loads.

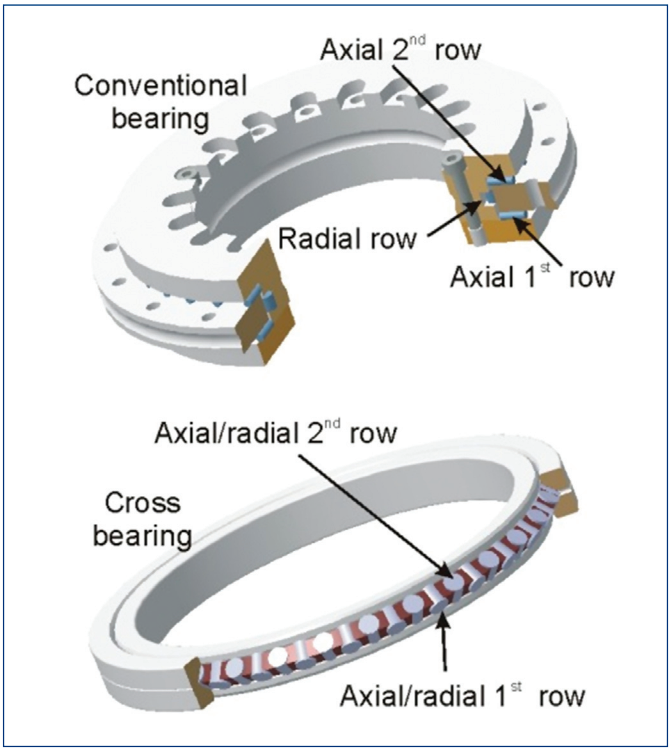

Figure 3. *Types of tilting rotary table bearings.*

The bearings work under preload ensuring their high axial and radial stiffness. As the whole structure heats up in the course of operation, the preload in one of the axial or angular bearings decreases while increasing in the other bearing. This has a significant effect on particularly friction moment M_1. However, from the design of the bearings and their arrangement it follows that an increase in friction moment in one row of rollers is compensated by a decrease in friction moment in the other row.

Consequently, as long as one of the rows of rollers is not completely unloaded, the sum of M1(1st row) and M1(2nd row) is approximately constant (fig. 4), which considerably simplifies the computational model of the bearings. The changes in torques M_o in the bearings are less important since the latter usually work at low rotational speeds and the product *vn* tends not to exceed 2000. Basic formulas for computing friction torque in such bearings are presented in table 3.

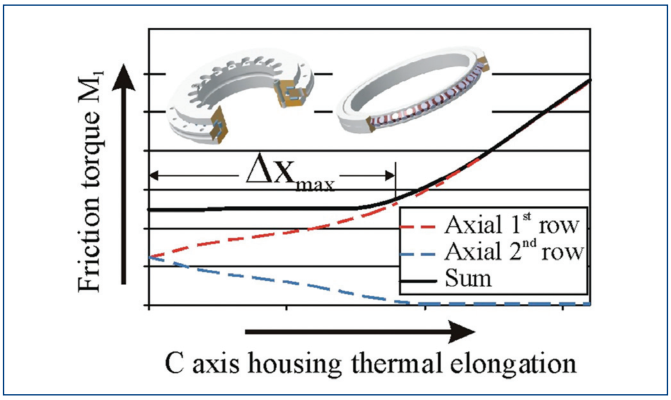

Figure 4. *Friction moment M1 as function of housing thermal elongation equal to change in race distance.*

Calculated quantity	Basic formulas
Total friction torque	Conventional and cross axial/radial cylindrical bearings $M = \sum_{i=1}^{k} M_{0i} + \sum_{i=1}^{k} M_{1i}$, [Blazejewski 2010] M_{0i}, M_{1i} – friction torque, [Palmgren 1964] k – number of roller rows k=3 – conventional bearing; k=2 – cross bearing
Torque due to applied load	For $\Delta X<\delta_{max}$ $M_1 = M_1(1^{st}\, row) + M_1(2^{nd}\, row) =$ constant ΔX – change in race distance due to heating up of structure δ_{max} – strain caused by bearing preload
Strain	**For conventional bearing** $\delta_{max} = 0{,}079 \cdot \frac{Q_o^{0.9}}{L_w^{0.8}}$, µm [Lundberg 1949]
Strain	**For cross bearing** $\delta_{max} = 0{,}079\sqrt{2} \cdot \frac{Q_o^{0.9}}{L_w^{0.8}}$, µm [Blazejewski 2010] Q_O – roller load:
Roller load	**For axial row** $Q_o = \frac{P_w}{z}$, N
Roller load	**For cross bearing** $Q_o = \frac{\pi P_w}{2z}$, N [Blazejewski 2010] P_w – axial bearing load, N z – number of rollers in one row, L_w – roller length, mm

Tab. 3. *Friction torque of tilting rotary table bearings.*

5. Power losses in linear direct drive motors

In order to calculate the amount of heat generated in linear motors one must carry out comprehensive analyses of the current flow, the thermal effects, the magnetic field and the electrical field in both the fixed part and the moving part of the motor. Power losses are determined by the motor design, e.g. the size of the air-gap between the motor's primary part and secondary part, the materials used (for, e.g., the permanent magnets), the supply voltage and the external loads (e.g. the cutting forces and the masses being moved). Also the high dynamic performance requirements which the linear motors used in precision machine tools must meet contribute to the amount of generated heat. In conditions without cooling the temperature of the two motor parts and the temperature in the air-gap may reach a few hundred degrees (fig. 5) [Abdou 2000]. As the air-gap increases, temperatures decrease in the whole supply voltage range. A linear motor with a large gap (6,350mm) is more sensitive to the applied external force than motors with smaller air-gaps.

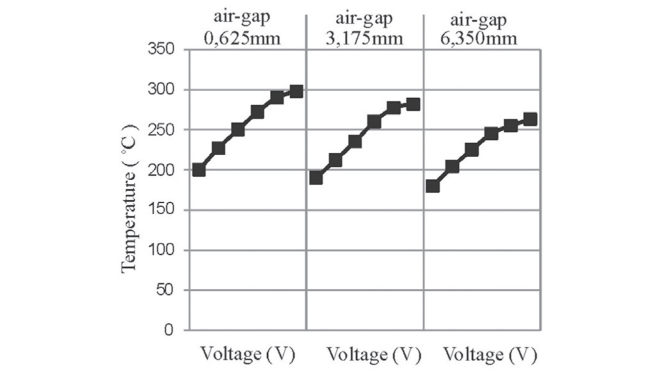

Figure 5. *Air-gap temperature versus voltage of 50-265 V for selected air-gaps [Abdou 2000].*

Therefore, linear direct drive motors must be intensively cooled to thermally separate the motor from the machine tool support structure. The applies particularly to the motor's moving (primary) part in which nearly 88% of the total heat is generated. The heat flux, which, despite the intensive cooling, may find its way to the machine tool support structure, depends rather on the effectiveness and stability of the cooling systems than on the amount of generated heat.

The modelling of the linear motor's thermal impacts on the machine tool structure is usually based on the motor manufacturer specifications or the investigator's practical experience. The existing practical knowledge boils down to the finding that as a result of the thermal impact of the motor on the machine tool, the latter's temperature increases locally, usually by 2-4 degrees.

This has been confirmed by experimental studies carried out on the 5-axis milling centre by the authors. Figure 6 shows the results of such measurements for the maximum (100%) feed rate which the control system permitted and for a much lower rate (amounting to merely 40% of the max. feed rate). Thus one can model the flow of heat from linear motors to the machine tool in such a way that the temperature of the machine tool components surrounding the linear motors will increase by 2-4 degrees. The drive's maximum feed rate at which the temperature increased by 6 deg. should not be regarded as representative for most machining processes.

In some cases the rise in motor temperature will be higher than a few degrees. For example, according to the experimental results for a heavy machine tool with linear drives (a horizontal machining centre) reported in [Kim 2004] (fig. 7), the temperature rise measured on the housing (the primary part) would reach nearly 25 deg.

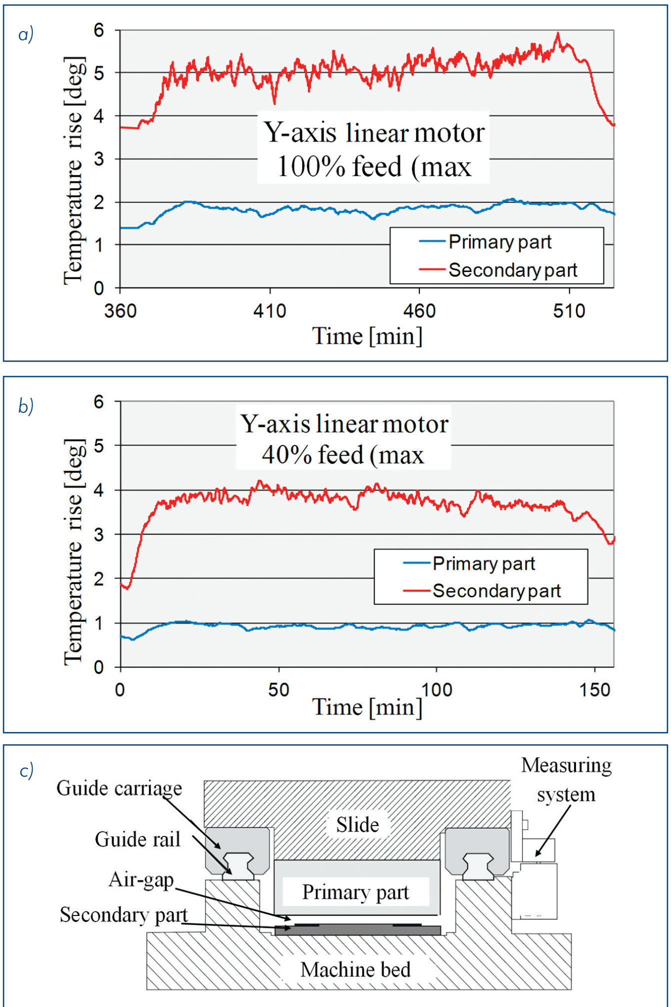

Figure 6. *Temperature rise for linear motor working in medium-size 5-axis machining centre at:*
a) maximum feed rate, b) 40% feed rate, c) linear drive components.

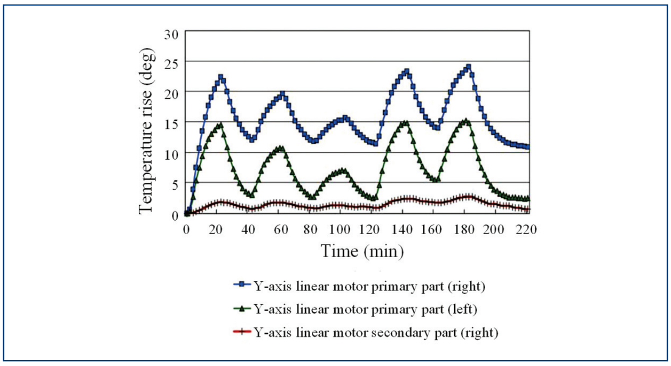

Figure 7. *Rise in temperature of linear motor moving heavy headstock [Kim 2004].*

The rise in the temperature of the magnets (the secondary part) in this case would not exceed 3 deg.Such a high rise in the temperature of the primary part of the linear motor installed in the heavy machine tool and moving a heavy headstock can be due to the great forces needed for machining and to the inadequate effectiveness of the cooling system. According to the dependence between the power, the force and the attainable maximum rate, in order to maintain constant V_{max}, the increase in force should be accompanied by a proportional increase in motor power.

6. Power losses in rolling guideways

Frictional resistances *F* in ball or roller Linear Motion (LM) systems depend on the external load, the preload, the viscosity of the lubricating medium and the number of seals.

Heat generated on the guideway by the balls or rollers set in the LM block is directly proportional to the frictional resistances and to the linear motion velocity. Loads F_m (table 4) acting on a set of rolling LM blocks may be different for each of the blocks. Therefore when determining them one must take into account the location of the gravity centre, the points of application of the loading forces and the dynamic forces during acceleration and deceleration.

The efficiency of rolling LM systems is very high (>99%), which is mainly owing to the very low rolling friction coefficient. This coefficient amounts to 0.002-0.003 and 0.005-0.01 for respectively

Calculated quantity	Basic formulas
Power losses	*Ball and roller LM systems* $P = (\mu \cdot F_m + f) \cdot v$, W µ – friction coefficient v – speed of motion, m/s Fm – average load, N f – frictional resistance of seals, N
Average load of block	$F_m = \sqrt[3]{\frac{1}{L} \cdot \sum_{i=1}^{n} (Fw_i^3 \cdot L_i)}$ Fi – varying load L – total distance travelled Li – distance travelled under load Fi
Total load of block	$Fw = F_o + k \cdot f_w \cdot F_{out}$, [THK 2007] F_O – preload F_{out} – preload k – external load factor k_w – dynamic load factor

Tab. 4. *Power losses in linear motion systems.*

ball elements and roller elements. Thus the thermal impact of such systems on the machine tool structure should not be significant in comparison with the other heat sources. Moreover, since the difference between static friction and dynamic friction is very small, the troublesome stick-slip effect does not occur.

The above conclusions have been confirmed by the research results reported in [Kim 2004] and by the authors' experimental studies carried out on a medium-size machining centre with linear drives (fig. 8). The increases in temperature shown in fig. 8, measured on the side block housing and on the guideway, were determined relative to the machine tool housing situated far from the guideways

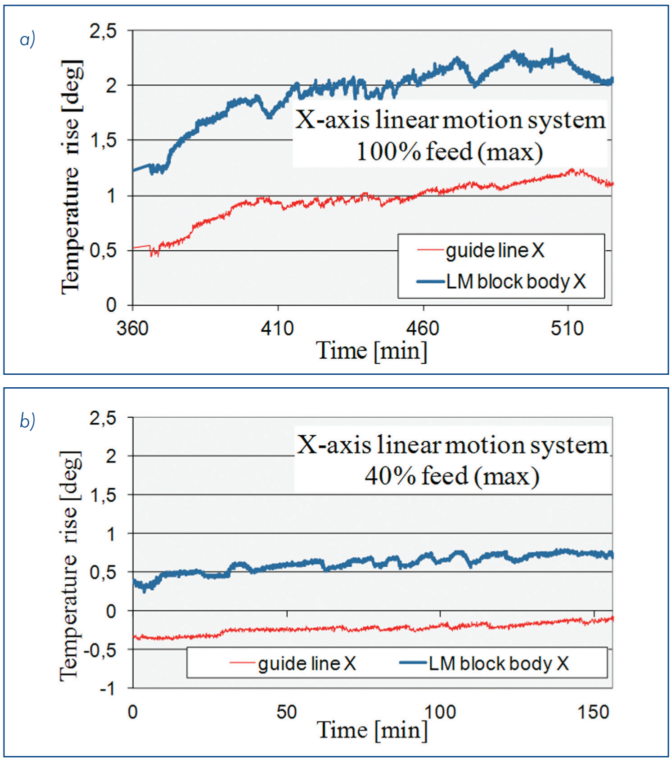

Figure 8. *Rise in temperature of linear motion system working in average-size 5-axis machining centre for: a) maximum feed rate, b) 40% of maximum feed rate.*

and the linear motor of the X-axis. Actually, they are the result of the combined impact of the heat generated in the rolling blocks and the heat generated by the linear motor since the two impacts are inseparable.

For the maximum attainable velocities of motion of all the controlled axes the rise in temperature of the LM block working in the Y-axis (fig. 8a) amounted maximally to about 2,3 deg. Some of the heat from the rolling blocks is transferred to the environment, whereby the increases in the temperature of the guideways are even smaller (1,1 deg.) than those of the rolling blocks. In average drive system operating conditions (40% of the max. feed rate), the rise in guideway temperature did not exceed 0.5 deg. (fig. 8b).

It is important that the rises in guideway temperature be small since they affect not only machine tool geometric errors, but also the accuracy of distance measurement (usually) by the linear scales fixed to the guideways.

7. Conclusion

It has been demonstrated that not only the processes taking place in machine tools, but also the degradation processes taking place in the course of service and the introduced repair processes should be modelled for proper machine tool design. Using as an example a 5-axis precision machining centre it has been shown that such modelling should be holistic in order to include the interdependences between deformations and thermal and dynamic loads, having a bearing on errors in the controlled axes.

Owing to the presented models of sources of errors in the controlled axes, relating to power losses in rotary direct drive motors, tilting rotary table bearings, linear direct drive motors and rolling guideways, close agreement (within 2mm) between the displacements/volumetric errors of the machining centre and the results of measurements was obtained.

As regards the modelling of the thermal interactions for linear direct drive motors, the authors think (on the basis of their studies) that valuable information can be gained from the measurements of temperature rises. It has been shown that at average motion velocities small rises in the temperature of the rolling blocks and the guideways occur.

The presented models have been successfully used in simulation computations and are being fine-tuned to new materials and constantly increasing velocities of rotational and feed motions. The models are supplemented by models of power losses in the rotating rolling units in the direct drive, discussed in Procedia CIRP [Jedrzejewski 2012].

References

[Abdou 2000] Abdou, G. and Tereshkovich, W. Performance evaluation of a permanent magnet brushless DC linear drive for high-speed machining using finite element analysis. Finite Elements in Analysis and Design, 2000, 35, 169-188.

[Blazejewski 2010] Blazejewski A. et al., Modelling thermal deformation of tilting rotary table with direct drive system. Journal of Machine Engineering, 2010, 10/4, 26-40.

[Etel 2008] Torque Motors Handbook. Etel Motion Technology, 902/Ver. D, 2008.

[Jedrzejewski 1996] Jedrzejewski, J. and Kwasny, W. Generalized indexes for evaluating intensities of heat sources within machine tool structure components. Edit. Institution of Wroclaw Univ. of Technology, 1996, 58-168, ISSN 0867-7778.

[Jedrzejewski 2004] Jedrzejewski J. et al., Hybrid model of high speed machining centre headstock.
CIRP Annals – Manufacturing Technology, 2004, 53/1, 285-288.

[Jedrzejewski 2012] Jedrzejewski, J. and Kwasny, W. Holistic precision error model for 5-axis HSC machining centre
with rotating rolling units in direct drives. SciVerse ScienceDirect, Procedia CIRP, 2012, (in print).

[Kim 2004] Kim, J-J. et al., Thermal behaviour of a machine tool equipped with linear motors. International Journal of Machine Tools & Manufacture, 2004, 44, 749–758.
[Lundberg 1949] Lundberg G. Cylinder compressed between two plane bodies. SKF Reg. 4134, 1949.
[Mekid 2009] Mekid S. (editor), Jedrzejewski J. at al., Introduction to precision machine design and error assessment. CRC Press, Taylor&Francis Group, 2009, ISBN 978-0-8493-7886-7.
[Palmgren 1964] Palmgren A. Grundlagen der Walzlagertechnik. Francklische Verlagshandlung, 1964, Stuttgart.
[THK 2007] Caged Ball LM Guide. CATALOG No.356-2E, THK Co., Ltd. 2007, Tokyo, Japan.

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