Journal of Machine Engineering, Vol. 10, No. 4, 2010

machine tool, tilting rotary table
thermal deformation, model

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MODELLING THERMAL DEFORMATION OF TILTING ROTARY TABLE WITH DIRECT DRIVE SYSTEM

Mathematical models representing power losses in tilting rotary tables with a direct drive system (increasingly often used in CNC machine tools) are presented. The process of creating an FEM model for computing the heating up and thermal deformation of tilting rotary tables is described. A simple way of verifying the FEM model on the basis of motor catalogue specifications is provided. Exemplary results of computations and thermal analyses carried out for a tilting rotary table with a synchronized two-sided direct drive system are reported.

1. INTRODUCTION

Today multiaxis (five and more numerically controlled axes) machining is used in many branches of industry, particularly the aviation industry and the car industry. Its main advantage is that complicated shapes, undercuts and hard to reach angles can be machined under one workpiece setting. Such machining is possible thanks to five-axis machine tools which instead of a standard table have a tilting rotary table with controlled rotation axes: A, C or B. Typical configurations of machine tools with linear and rotation axes are shown in Fig. 1 [12]. In the case of one-sided drives, power losses on the motor side are greater, which results in the lack of thermal symmetry of the tilting table. Tilting tables with a synchronized two-side drive system have no such drawback. The additional rotation axes increase the flexibility of the machine tool, but at the same time they constitute an additional source of linear and angular errors. The errors are the sum of geometric, kinematic and thermal errors.

This paper discusses problems connected with the modelling of the thermal behaviour of a two-axis tilting rotary table with a synchronized two-side drive system during an assumed duty cycle. The main objective was to develop a methodology and a tool based on

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the finite element method for predicting the heating up and thermal deformation (the phenomena affecting the total machine tool error) of the table's components.

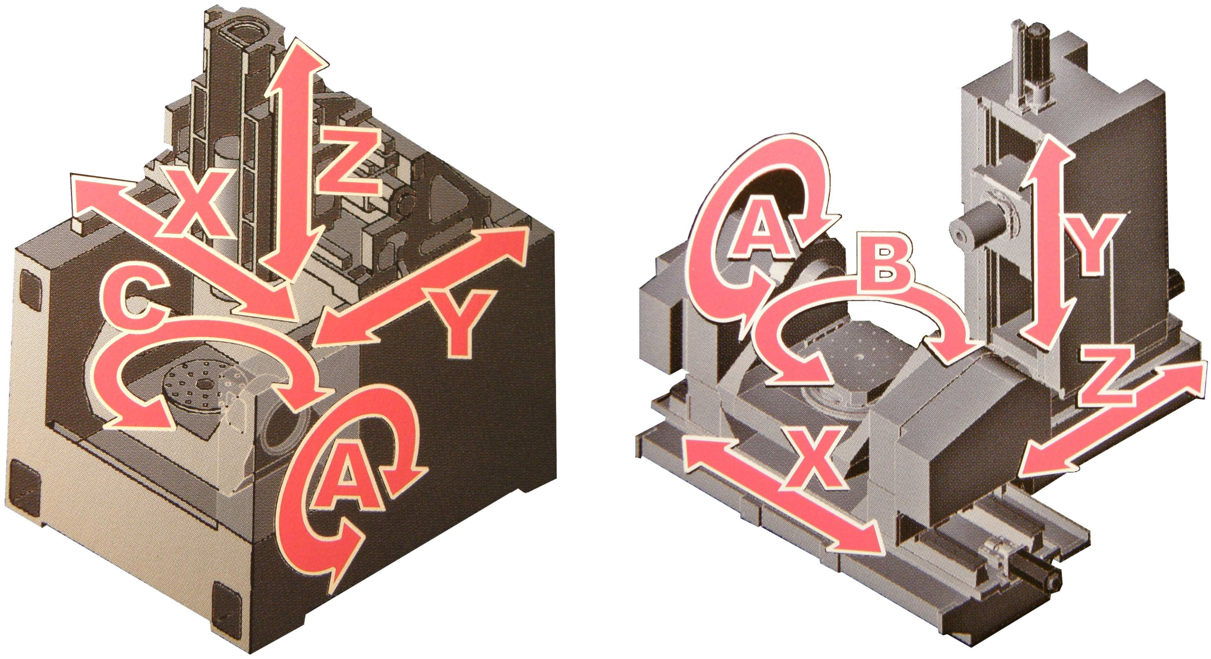


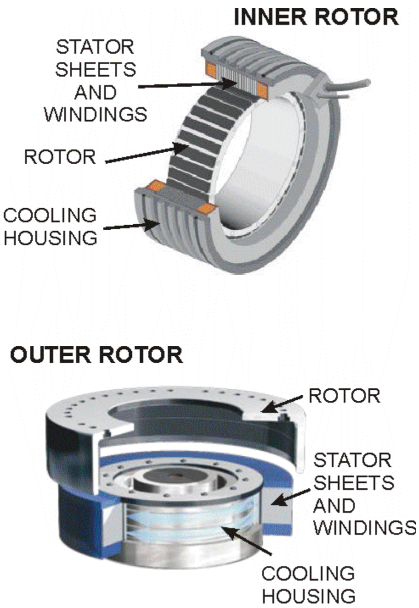
Fig. 1. Configuration of milling centre with rotation axes A, B & C, a) vertical centre, b) horizontal centre [12]

2. TILTING ROTARY TABLE DESIGNS

Currently there are two main machine tool tilting rotary table designs, differing in their drive transmission: a worm gear or a synchronized one-sided or two-side drive (Table 1). Owing to their clear advantages, designs with torque motors begin to predominate, especially in new machine tool models. Their major advantages include: high speeds of rotation axes C and A, high accelerations, no clearances, motion fluidity and high positioning precision and speed. The mechanical drive transmission entails difficulties in achieving the rotational speeds and torques required by modern HSC machine tools, and motion errors due to friction in the mechanical gear and to reverse clearance [8]. Other advantages stemming from the use of direct drives (torque motors) are presented in Table 2.

Torque motors eliminate the need to use gearboxes, worm gears and other mechanical ways of transmitting drive. They are also characterized by high dynamic response without hysteresis and a large gap (0.5-1.5 mm) between the stator and the rotor, which facilitates assembly. A major feature of the motors are their dimensions, i.e. large outside diameters at a relatively small length. For a diameter of over 2 m the motor's length may be less than a few tens of mm. Also their inside diameters are equally large since the rotor is a thin ring with permanent magnets (Fig. 2). The use of permanent magnets guarantees high efficiency of the motors.

Table 1. Typical tilting rotary table designs

<table>
<tr><th>Type of motor</th><th>Drive transmission</th><th>View of tilting rotary table</th></tr>
<tr><td>Servomotor
</td><td>Worm gear
</td><td>
[6]</td></tr>
<tr><td>Torque motor

</td><td>One-sided direct drive without support

One-sided direct drive with support

Synchronized two-sided direct drive</td><td>
[11]

[10]

[7]</td></tr>
</table>

Table 2. Comparison of specifications of indirect and direct drive motors [1]

Specification	Conventional Mechanical Drive	Direct Drive
Cost (conventional drive = 100%)	100%	97%
Mounting/assembly time	88 hr.	12 hr
Position index time	1 sec	0.33 sec
Position repeatability	2.5 arc sec	1 arc sec
Feedback system resolution	-	0.18 arc sec
Stiffness	$7.2x10^6$ Nm/rad	$13x10^6$ Nm/rad

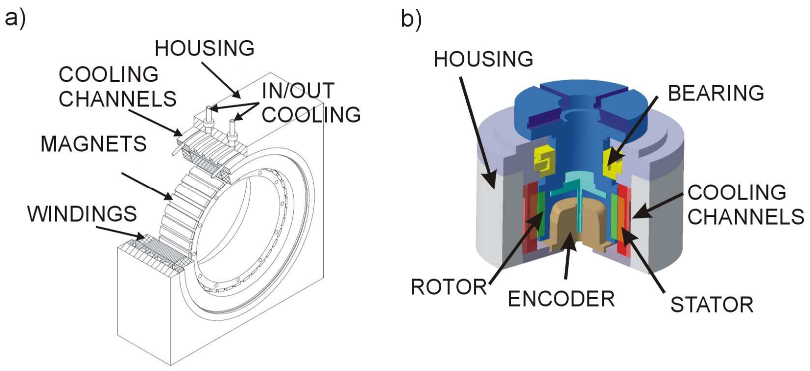


Fig. 2. Direct drive: a) torque motor [1], b) direct drive rotary Table

Thanks to the large outside and inside diameter, high torques can be achieved, which is the main advantage of such motors.

A model of the thermal behaviour of tilting rotary tables with a direct drive system should comprise (Fig. 3):

- models of heat generation in bearings and motors,
- a bearing stiffness model,
- a cooling model,
- a model of rotation and tilting axes operating conditions,
- a geometrical model,
- an FEM model.

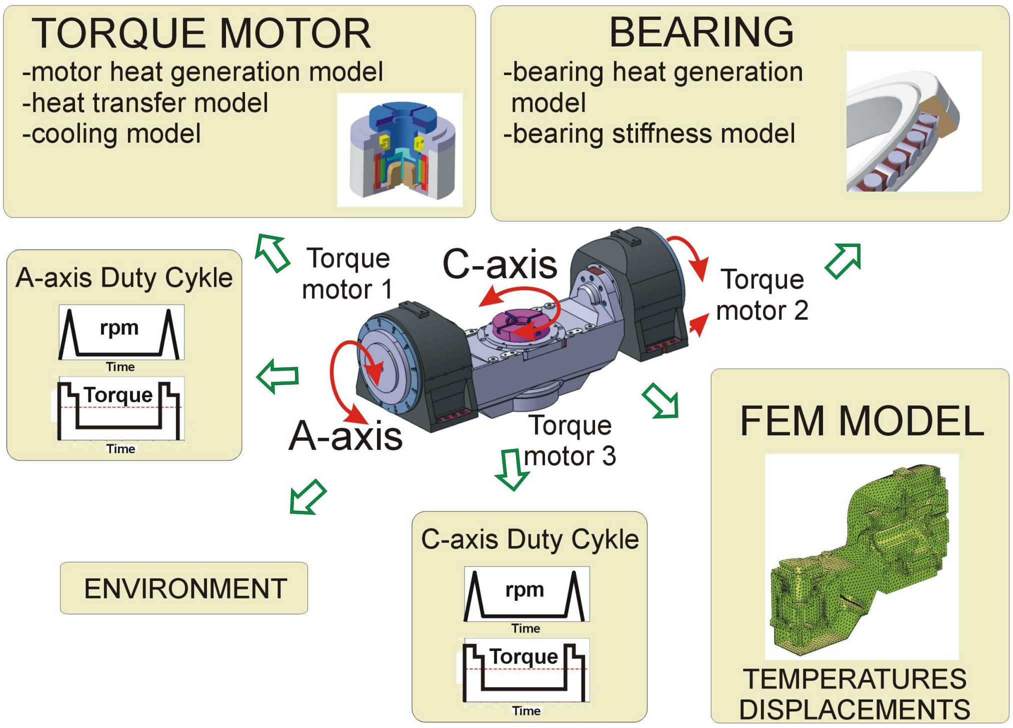


Fig. 3. Model of thermal behaviour of tilting rotary tables with direct drive

3. MODELLING OF THERMAL LOADS

3.1. AXIAL/RADIAL CYLINDRICAL BEARINGS

Conventional axial/radial bearings, e.g. of type RTC and YRT, have two rows of rollers transmitting longitudinal forces and one row of rollers transmitting transverse forces (Fig. 4a). An axial/radial cross bearing, e.g. of type RU and SX, has two rows of rollers set perpendicularly to each other in one plane (Fig. 4b). The two varieties of bearings can be mounted in both axis C and axis A of the tilting rotary table.

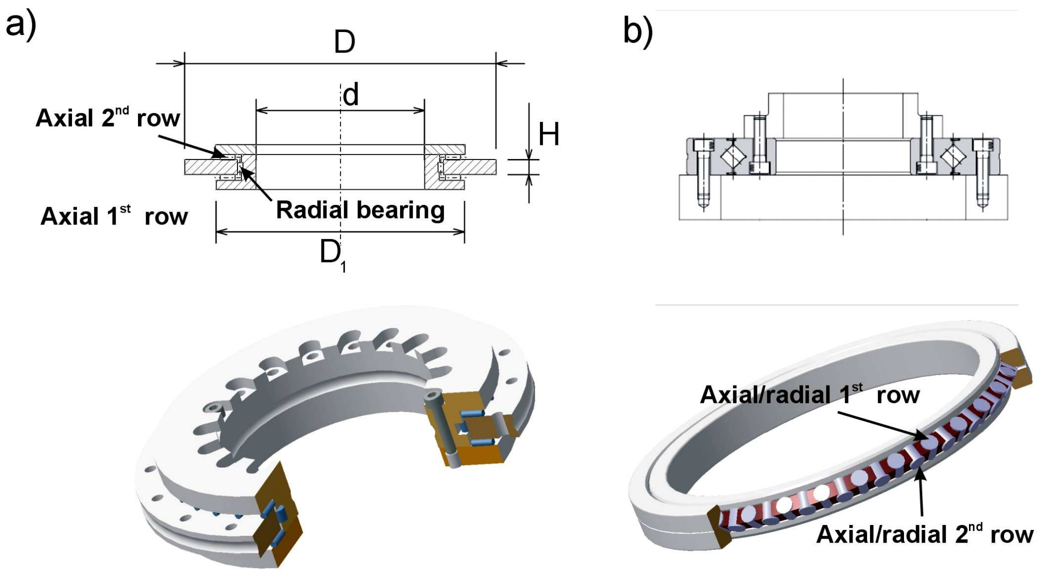


Fig. 4. Axial/radial bearings: a) conventional, b) cross

For the modelling of the thermal output of such bearings one can use the total power losses in each of them. The total power losses in a bearing of this type are the sum of the losses in two or three rows of rollers. The bearings should be preloaded during assembly or in the case of cross bearings (class CC0), by the manufacturer. The power losses in the bearings can be calculated from the modified Palmgren formulas [5] or from the following relations:

$$N = \frac{M \cdot n}{9{,}55}[W] \tag{1}$$

$$M = \sum M_o + \sum M1 \tag{2}$$

$$M_o = f_o \cdot 10^{-1} \cdot (\nu \cdot n)^{2/3} \cdot d_m^3 [Nm] \tag{3}$$

$$M1 = f1 \cdot P \cdot d_m [Nm] \tag{4}$$

where:
N – bearing power losses,
M – the total moment of friction,
M_o – the moment of hydrodynamic friction,
M1 – the moment of load induced friction,
d_m – the mean diameter of the bearing [m],
n – rotational speed [rpm],
ν – the kinematic viscosity of the lubricating medium [cSt],
Assumption 1 – longitudinal preload P_w=0,1Ca [N],
Assumption 2 – transverse preload P_p=0,1Cr [N],
C_a, C_r – respectively axial and radial catalogue dynamic capacity [N].
Coefficient f_o, f1 values for cylindrical roller bearings are shown in Table 3.

Table 3. Mean values of f_o, f1 coefficients for cylindrical roller bearings [3]

	Cylindrical roller bearings		
	Axial	Radial	Cross
f_o	0.0018	0.0003	0.0018
f1	2.5	2.0	2.5

Initially, postassembly loads of 0.1 Ca and 0.1 Cr can be assumed for RTC or YRT bearing calculations. This assumption will be proved when friction moment M1 close the value of M_{RL}.(rest friction moment) specified by some bearing manufacturers is obtained from the calculations. Power losses versus rotational speed for the bearings are shown in Fig. 5a.

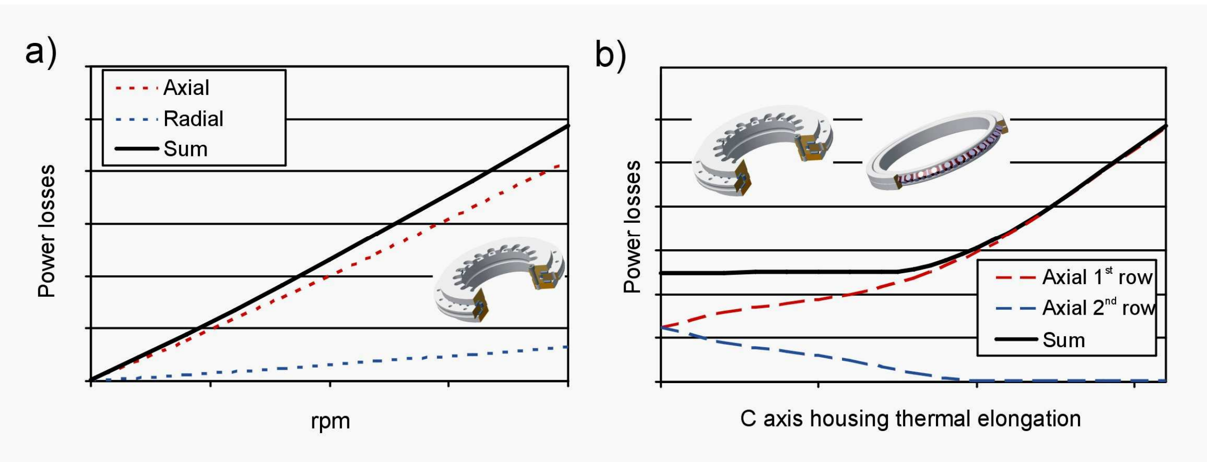


Fig. 5. Power losses in axial/radial bearings, as function of: a) rotational speed, b) thermal elongation of axis C housing

If RTC or YRT bearings are used for axis A, then as a result of thermal elongation Δx of the axis C housing, longitudinal load P_w in one row of the axial bearing increases while the longitudinal load in the second row decreases (Fig. 6). Also for cross bearings the load of the rollers set at an angle of +45° (row 2) will increase while that of the rollers set at an angle of -45° (row 2) will decrease. The magnitude of the load changes depends on, among other things, bearing stiffness. As a result, in the extreme case one of the rows of rollers may be completely unloaded, which may adversely affect the power losses and durability of the bearings (Fig. 5b).

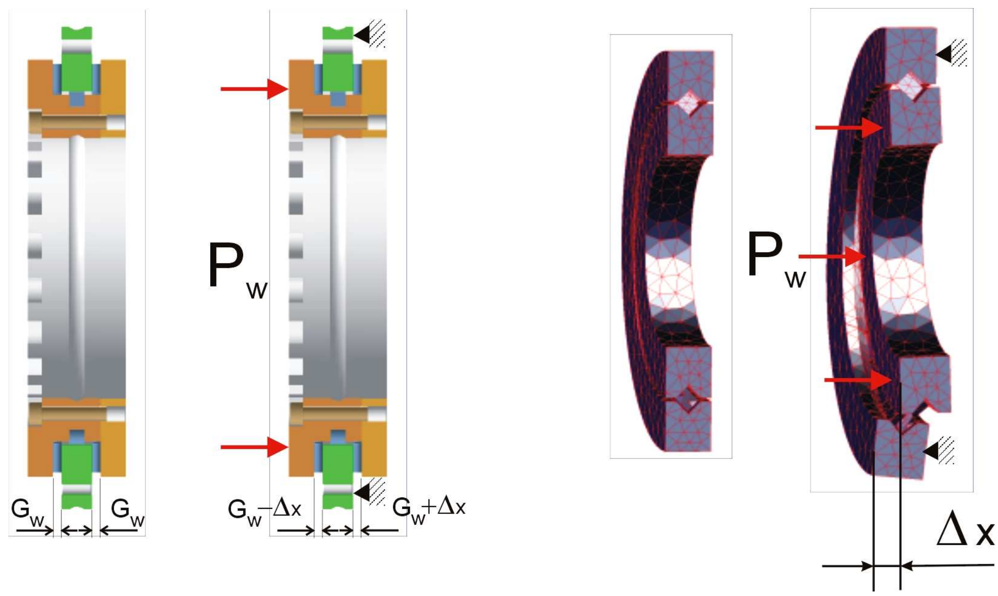


Fig. 6. Mutual displacements of raceways in axial/radial bearings: G_w, – distance between raceways in preloaded bearing, Δx – change of distance between raceways

For the complete unloading of one of the rows, when the thermal elongation exceeds the longitudinal preload of the bearing, the power losses in the latter practically do not depend the heating up of the tilting rotary table structure. Prestrain δ caused by the preload of the bearing can be determined from the relations:

reduced distance between raceways – $\delta = 3.06 \cdot 10^3 \cdot \frac{Q_o^{0.9}}{l_w^{0.8}}[\mu m]$ (5)

for cross bearings – $\delta = \frac{2}{\sqrt{2}} 3.06 \cdot 10^3 \cdot \frac{Q_o^{0.9}}{l_w^{0.8}}[\mu m]$ (6)

axial bearing roller load – $Q_o = \frac{P_w}{z}[N]$ (7)

cross bearing roller load – $Q_o = \frac{\pi P_p}{2z}[N]$ (8)

where:
P_p – transverse bearing load,
z – the number of rollers in one row,
l_w – roller length [m].

3.2. TORQUE MOTORS

Power losses in motors should be determined taking into account the repeatable tilting rotary table duty cycle consisting of the starting stage, a period of continuous running, and the braking stage (Fig. 7). The value of torque Tc loading the motor in the continuous running period is the sum of bearing friction losses, cutting power losses and losses due to other external influences.

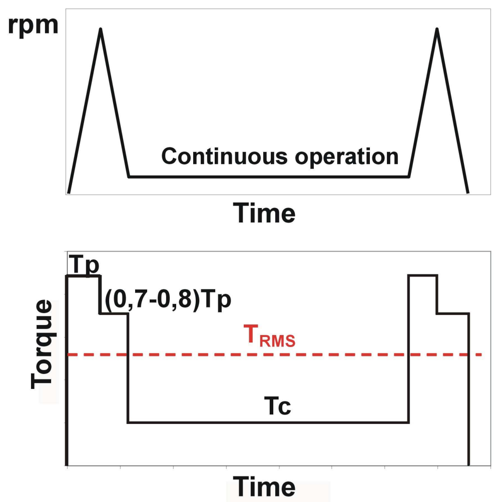


Fig. 7. Duty cycle of tilting rotary table for axis A or C

Equivalent torque T_{RMS} for the whole cycle is calculated as the weighted rms average for the three stages from the following relation [9]:

$$T_{RMS} = \sqrt{\frac{\sum_{i=1}^{N} T_i^2 \cdot t_i}{t_{cyklu}}} \tag{9}$$

Maximum motor dissipated power Pc, peak torque Tp and continuous duty torque Tc are specified in catalogues for a motor winding temperature of 130°C. But the actual temperature is determined by the motor duty cycle, the motor mounting structure (designed and made by the table manufacturer), the size of the heat exchanging surfaces and especially the cooling system. For the actual winding temperature other than 130°C, the dissipated power value should be corrected taking into account equivalent torque T_{RMS} in accordance with the relation:

$$Pc_{(\Theta \neq 130)} = \frac{1{,}5 \cdot Rc \cdot Ic^2}{\left(\frac{Tc}{T_{RMS}}\right)^2} \tag{10}$$

In the above formula:
Rc – the resistance of the windings at the actual working temperature.

$$Rc = R_{20}\left(1+\left(0{,}00392\cdot(\Theta-20)\right)\right) \tag{11}$$

R_{20}, Ic – catalogue resistance and current values,
Θ – the actual temperature of the windings.

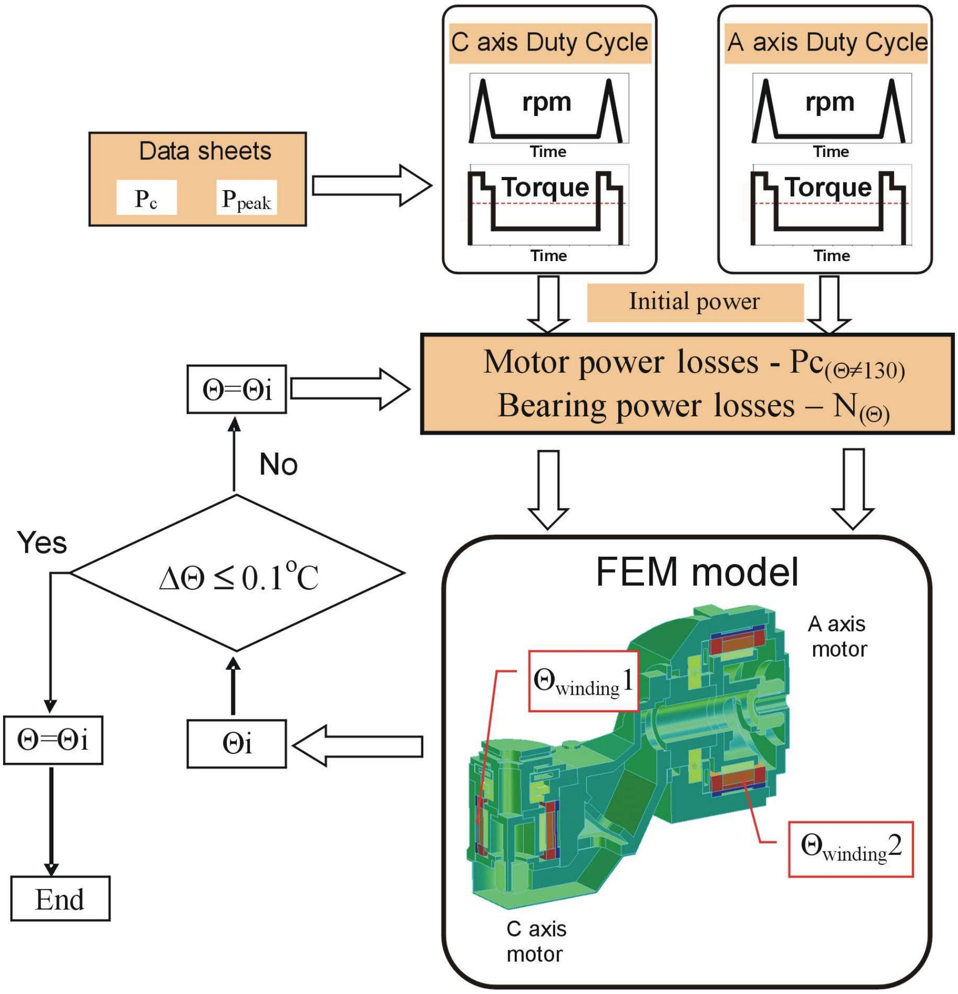


Fig. 8. Algorithm for calculating power losses in torque motors

In order to determine the actual motor power loss value one should assume operating winding temperature Θ for the motor mounting and the cooling conditions and then iteratively search for the actual temperature using an FEM model. The power value calculated on the basis of the catalogue data and the duty cycle should be the starting point for iterations. The algorithm for this iterative process is shown in Fig. 8.

After the FEM program calculates the actual operating temperature, the values of power loss $Pc_{(\Theta=130)}$ and the bearing power losses (which are also a function of temperature) are adjusted. The iterative process is continued until the temperature assumed for the motor power loss calculation becomes equal to the winding temperature obtained from the FEM model simulations.

4. MODELLING OF STIFFNESS OF BEARING ROLLING ELEMENTS

According to the Hertz theory, roller bearing strains at the roller/ring interface have a linear character. In order to represent the strains in the FEM model, the cylindrical rolling element can be replaced with a rectangular prism with similar linear elastic properties (Fig. 9).

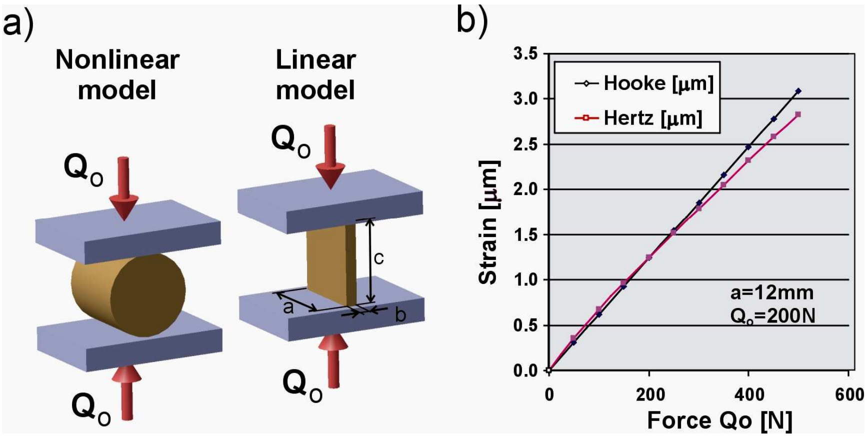


Fig. 9. Model of deformations in roller/raceway assembly in roller bearing: a) analysed models, b) comparison of models

Such dimensions of the rectangular prism should be selected that condition $\delta_{(Hertz)}=\delta_{(Hooke)}$ is satisfied for a bearing preloaded with force $P_w = z \cdot Q_o$. The nonlinear elastic deformations of the roller/raceway set can be calculated from the relation presented in section 3.1. The elastic deformations of the rectangular prism (the latter and the rings form one element) can be calculated from the relation:

$$\delta_{(Hooke)} = \frac{Q_o \cdot c}{a \cdot b \cdot E} \tag{12}$$

where:

E – a longitudinal modulus of elasticity,

a,b,c – the dimensions of the rectangular prism.

A comparison of the nonlinear strains of the roller/raceway set with the ones calculated using the linear cuboidal model is shown in Fig. 9b. In the area in which force Q_o (±200N) is predicted to change, the differences between the strains calculated from the two models do not exceed 0.2 µm. The calculations were done for a cross bearing with 12 mm long rollers, under load Q_o=200N.

5. MODELLING OF COOLING

Since the torque motor is fully integrated with the rotation axis, its heating affects more adversely rotary table or self-aligning axis motion precision than in the case of other ways of drive transmission. Therefore cooling is particularly vital for such motors. Because of its high heat transfer rate (2000 W/m^{2}K), mainly water is recommended as the cooling medium. Torque motors manufacturers in their catalogues specify radiator water flow rate Fw necessary to have temperature ΔTw=5^{o}C at the radiator's inlet and outlet at a winding temperature Θ=130^{o}C. Also pressure drop Δp at this rate of flow is specified. For the actual winding temperature the two values should be adjusted using the relation [9]:

$$Fw = 0{,}0143 \cdot \frac{Pc_{\Theta}}{\Delta Tw}, \qquad \Delta p = \Delta p_{(kata\log)} \cdot \frac{Fw}{Fw_{(kata\log)}} \tag{13}$$

where: Pc$_\Theta$ – motor power losse
inding temperature Θ (section 3.1.).

6. FEM MODEL

The first step in creating an FEM model consists in the ordinary simplification of the detailed CAD 3D model. This is reasonable considering the limited capacity of the software tools and less difficulty with the discretization of areas critical for calculation accuracy. In the case of tilting rotary tables with torque motors, elements such as screws, holes, cables, etc, having no significant effect on the behaviour of the calculated structure can be neglected in the calculation model. The simplified 3D model should have such a structure which in the course of model verification, will make it possible to identify major components, e.g. radiators, windings, bearings and so on. The integrity of the major assemblies or subassemblies is essential for the efficient setting of boundary conditions in the appropriate areas. The FEM model should also reproduce the actual total area of heat transfer to the environment.

In the case of geometrically, thermally and stiffness-wise symmetrical objects, it is reasonable to model only a certain fraction of the object, e.g. half, one fourth, etc. Geometrical objects are discretized, taking into account the declared connections between the assemblies and the subassemblies and using the available preprocessor.

It is mainly boundary conditions which determine the validity of the model. They come down to the assignment of specific properties, such as:

- convection and radiation coefficients, material specifications;
- heat sources, same temperature areas, areas being cooled;
- attachment points,
- interelement contact areas,
- ambient temperature,

to surfaces or nodes.

Motors and bearings are the sources of heat in the tilting rotary table. The heat they generate must be assigned to appropriate surfaces or nodes of the discretization mesh. The power losses calculated for each torque motor can be assigned to the volumes of the elements making up the stator and rotor windings (Fig. 10). When modelling the heat exchange through the radiator, the heat exchange coefficient can be assigned to all the discrete elements describing the radiator's surface. If the radiator's surface in the FEM model differs (as a result of, for example, the introduced geometry simplifications) from the detailed CAD model, the heat exchange coefficient value should be adjusted appropriately.

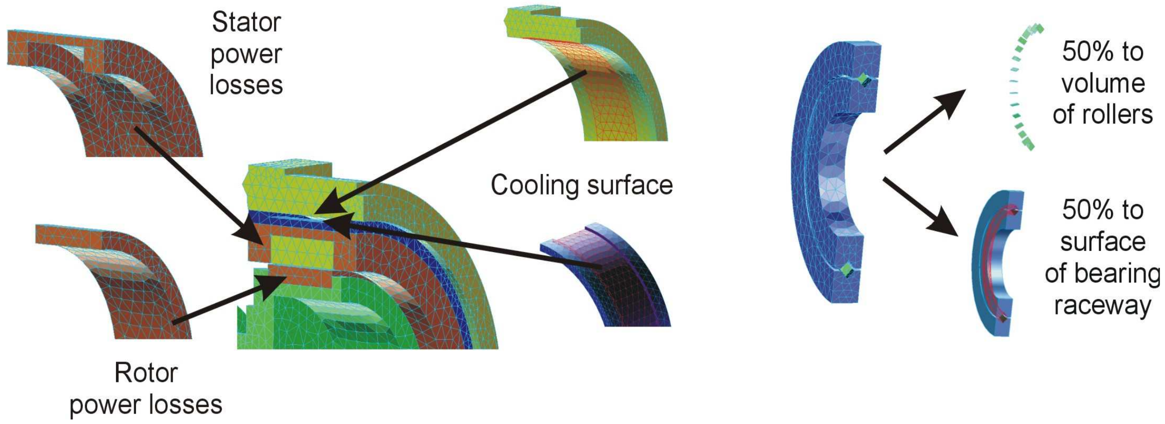


Fig. 10. Modelling of heat losses in torque motor and in bearings, and modelling heat dissipation by radiator

The power losses calculated for the bearings should be divided between their (two) raceways and rollers, allocating 50% to each group. The power loss percentage allocated to the rollers is distributed among them proportionally to the volume of the particular rollers. The other power loss percentage is distributed among the raceways proportionally to the geometric surface of the particular raceways.

7. COMPUTER SIMULATION

The FEM model can be verified using the manufacturer torque motor specifications also for the case when heat is removed to the environment only through convection and radiation. When water cooling is absent, the winding temperature should be around 130°C for the continuous running of the motor loaded with catalogue torque Tp. Then one can check whether the size of the heat removing surfaces is sufficient and if the assumed conditions of heat transfer from the surfaces to the environment have been accurately modelled. This verification can be done on the basis of the catalogue specifications for the water cooled motor. Computations performed for the motor loaded with the catalogue torque under continuous duty in the catalogue cooling conditions (the radiator inlet/outlet temperature difference, the rate of flow, the pressure drop) should yield a winding temperature close to 130°C.

The model verified in this way was used for computing a tilting rotary table with two torque motors, two cross bearings and one conventional axial/radial bearing. Because of the geometrical symmetry, a half of the motor in axis A and one fourth of the motor in axis C were modelled (Fig. 11).

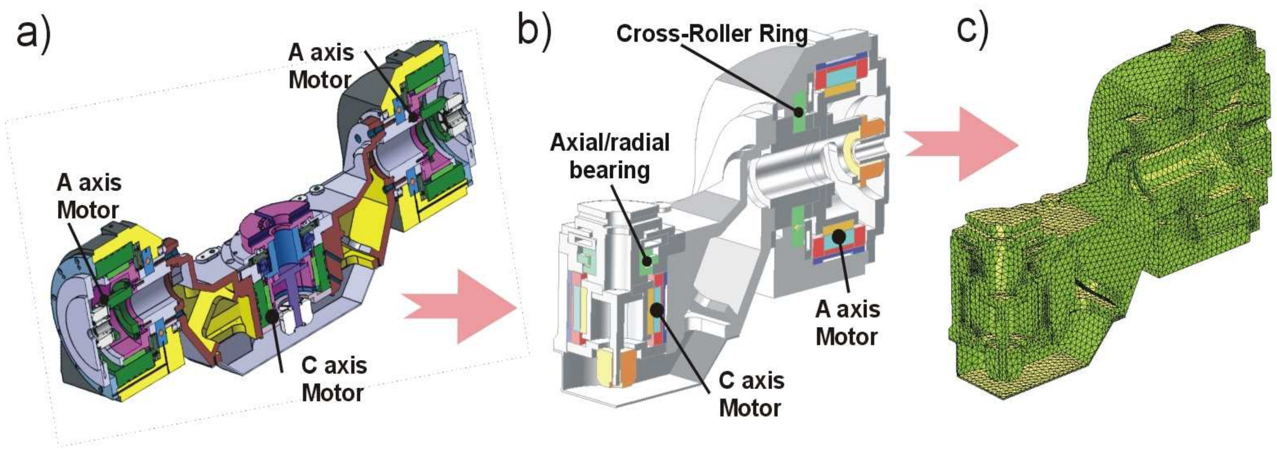


Fig. 11. Model of a tilting rotary table: a) real object, b) geometrical model, c) FEM model

The repeatable motor duty cycle was assumed to last 2.62 s and 1.52 s for respectively axis A and C, with the continuous running period constituting 70% of the whole cycle (Fig. 12). Equivalent torques T_{RMS} amount to respectively 228 Nm and 118 Nm and the rotational speeds during continuous running amount to 10 rpm and 20 rpm for respectively axis A and C. The equivalent torques were calculated taking into account a cutting torque of 30 Nm. A general picture of temperature and thermal deformations for the table duty cycle is shown in Fig. 13 and more detailed results are presented in Figs 14 and 15.

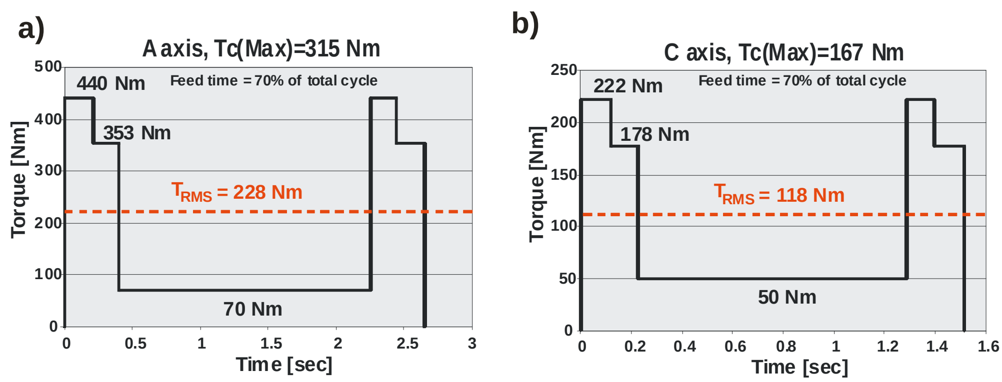


Fig. 12. Duty cycles: a) axis A, b) axis C.

A simulation of the heating up and thermal deformations of the tilting rotary table has shown that in the analyzed structure the motors heat up similarly in axes A and C despite the differences in their size and load. Differences between the winding temperatures and between the bearing temperatures do not exceed one degree. This is undoubtedly owing to the intensive and well suited cooling of the motors.

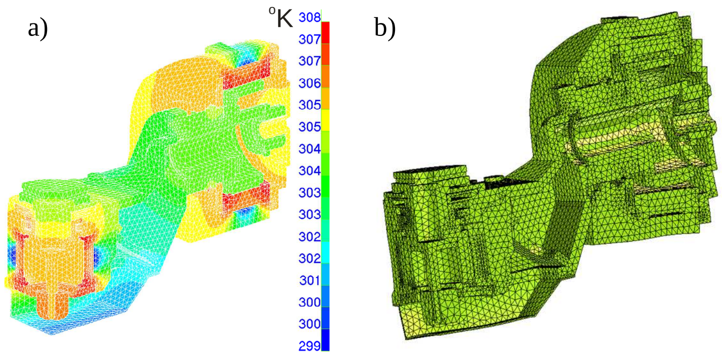


Fig. 13. Results of temperature and thermal deformation computations for duty cycles from Fig. 12: a) temperature, b) thermal deformations

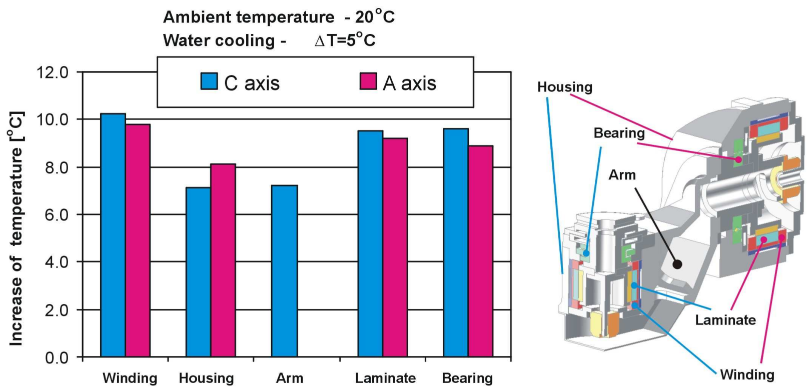


Fig. 14. Results of temperature rise computations for selected sections of tilting rotary table

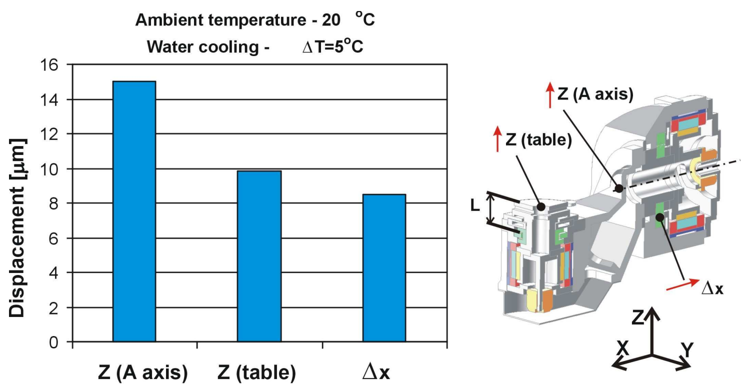


Fig. 15. Results of thermal deformation computations

A rise in housing temperature by about 8 degrees results in a significant displacement (15 μm) of axis A towards Z. This has a direct effect on the displacement of the table and so on the thermal stability of the machine tool. The thermal elongation of the arm along axis Z has a compensating effect on the resultant table displacement along this axis. Also the slanting of axis A has a compensating effect on the displacement of the table. Thus the total displacement (10 μm) of the table in direction Z comprises: the displacement of axis A, the slanting of axis A, the thermal elongation of the arm of axis C and the thermal elongation along axis Z, measured from the bearing of axis C to the surface of the table. Moreover, as a result of thermal elongation Δx of the axis C arm in direction X, amounting to over 8μm, the raceways of the cross bearing set an angle of $+45^o$ move closer to each other by this value while set at an angle of -45^o they move away from each other. This may lead to a considerable increase in the loading of some of the rollers and to the unloading of the rest, which may adversely affect the bearing lifetime.

8. CONCLUSION

The proposed model of the thermal behaviour of the tilting table with controllable axes C and A comprises all the major factors having an influence on the size of generated power losses and thermal displacements. The numerical simulation of the thermal behaviour of the table, the bearings and the self-aligning table bracket in selected operating conditions has shown that the model performs well as heat generation and removal and concerned. It will be further improved on the basis of its experimental verification for a real object with regard to: the power consumed by the motors, the object temperatures and the cooling conditions.

Once a complete thermal model of the tilting rotary table is obtained, it will become possible to reduce (through design and material modifications) the thermal displacement of the table in direction Z thanks to the compensating effect of the two displacement components.

REFERENCES

[1] HOLZKNECHT A., 2003, *Direct drive torque motors for machine tool applications*, ETEL, Inc, Schaumburg, German.

[2] ISO/CD 230-1.5.5., 2008, *Test code for machine tools - Part 1*: *Geometric accuracy of machines operating under no-load,* Committee Draft, International Organization for Standardization, Geneva, Switzerland, 2008. Reference number ISO/TC 39 / SC 2 N 1706.

[3] KRZEMIŃSKI-FREDA H., 1989, *Rolling bearing*. PWN, Warszawa, (in polish).

[4] MORIWAKI T., 2008, Multi-functional machine tool, CIRP Annals - Manufacturing Technology, 57, 736–749.

[5] JEDRZEJEWSKI J., KWASNY W., STRAUCHOLD S., 1991, *Numerical modelling of the thermal behaviour of spindle bearing assemblies for precise machine tools,* Proceedings of CIRP Conference on PE & MS, 525-535.

[6] *Rotary tables*, Catalogue, KOMA PRECISION, INC.

[7] *Rotary tables*, Catalogue, Peiseler Prazisionmaschinenbautechnik.

[8] SATO R., TSUTSUMI M., 2007, *High performance motion control of rotary table for 5-axis machining centres,* Int. J. of Automation Technology, 1/2, 113-114.

[9] *Torque Motors*, ETEL Doc. – Handbook, TORQUE 902/Ver. D/2/6/08.

[10] LCM-General Catalogue, www.Icmitalia.it.

[11] CyTurn, CyTorque Catalogue - CyTec - #1, http://pdf.directindustry.de/pdf/cytec/cyturn-cytorque-katalog/16094-111200.html.

[12] *Machine tools*, Mitsuiseiki Catalogue, http://www.mitsuiseiki.co.jp.