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

machine tool, spindle unit
temperature, thermal deformations, model

Zdzisław WINIARSKI[1]
Zbigniew KOWAL[1]
Wojciech KWAŚNY[1]
Jae-Yong HA[2]

THERMAL MODEL OF THE SPINDLE DRIVE STRUCTURE

A computation system dedicated for machine tool spindle drives is presented which integrates a FDM model with a FEM model. Modelling of heat exchange through coolers are discussed, as well as modelling of heat exchange between cooling-lubricating oil and gear transmissions, shafts and headstock walls. In modelling of heat exchange inside headstock, the use of additional elements - type "Fluid" and "Air" - is proposed. Sample modelling results are presented concerning a headstock from a drilling-milling machine tool for machining with high loads and are compared with experimental data.

1. INTRODUCTION

Main drives of turning, milling and grinding machine tools, as well as machining centres are usually equipped in synchronous or asynchronous electric motors. Types of drive transmission from a motor to a machine tool spindle can be various. The introduction of a gear box between the motor and spindle allows, in a broad range, adjusting rotational speed and torque on the spindle for the needs of the technological process. While selecting drive parameters it is important to achieve required rotational speed quickly which requires high acceleration and at the same time suitable motor power surplus. In such solutions there is a possibility of the appearance of large radial forces, increased vibrations, noise and other unfavourable phenomena. At required high torques between the motor and the spindle, gear transmissions are applied. At lower torques such transmission is adequately shortened all the way to a direct integration of the motor with the spindle (electrospindle).

There are also solutions of main drives with high-torque motors or motors coupled in series in order to increase torque on the spindle. Such solutions are however much more expensive from these described before. The selection of one of ways for transmitting drive will always be a compromise between the required torque value and rotational speed, precision and cost.

[1] Institute of Production Engineering and Automation, Wroclaw University of Technology, Wroclaw, Poland
[2] DOOSAN Infracore Corporation, Korea

Difficulties in modelling main drives rise together with the complexity of their kinematic structure and the need of applying motor, bearing and transmission cooling systems. It also causes the rise of the amount and diversity of generated heat sources, which require proper computation models, as well as the rise of the complexity of modelling forced cooling of particular drive components.

The possibility of using commercial FEM programs, such as: Nastran, Ansys, ABAQUS or Simdesigner for CATIA to simulate the behaviour of machine tool structures is limited, because they do not have many required boundary conditions, which enable taking into account significant interactions between the efficiency of internal heat sources and temperatures and deformations of the entire structure. This especially applies to bearing assemblies, gear transmissions, couplings, etc. placed in closed spaces, in which power losses are a relation of temperature with thermal and elastic dimension changes. In such situations, the best results are yielded by dedicated systems. In this paper, a dedicated system, called SATO[6], will be described, which has been elaborated and is still developed at the Wroclaw University of Technology, as well as thermal and elastic phenomena models used in it, enabling the analysis of complex kinematic structures of spindle drives in varying operational conditions and in transient thermal states.

2. DEDICATED SYSTEM SATO FOR THERMO-ELASTIC ANALYSES

This system is designated to simulate thermal and static behaviour of machine tools. During its development, various methods were used, such as a Finite Element Method (FEM), Finite Difference Method (FDM), as well as a series of procedures supporting modelling of boundary conditions [5]. This system integrates computations of temperature and displacement distributions with determining power losses in kinematic links of drive systems (Fig. 1.).

Such integration ensures high precision of simulations. It is especially important for the spindle assembly, where interactions take place between power losses, temperature and thermal dimensional changes. These can significantly change values of power losses in relation to these defined by simplified mathematical models without such conjugations.

Boundary conditions included in the system ensure modelling of thermal and elastic phenomena taking place in the structure, as well as on stationary and mobile links between machine tool units and components. These include: radiation, natural convection and convection forced by the movement of elements, carrying of heat by means of coolant flow, contact conductivity and contact stiffness. Boundary conditions included in the system enable modelling of heat exchange with the environment, as well as in closed spaces inside housings, as it takes place e.g. inside headstock bodies.

Internal procedures for computing power losses in subassemblies and kinematic pairs of a machine tool require preparation of input data in the initial (cold) state with help of external procedures. The most complex ones include procedures for ball bearings in the assembled state. Fig. 2 shows an algorithm for determining such input data concerning spindle angular contact bearings taking into consideration contact angle changes in a bearing according to ball bearing theory [2].

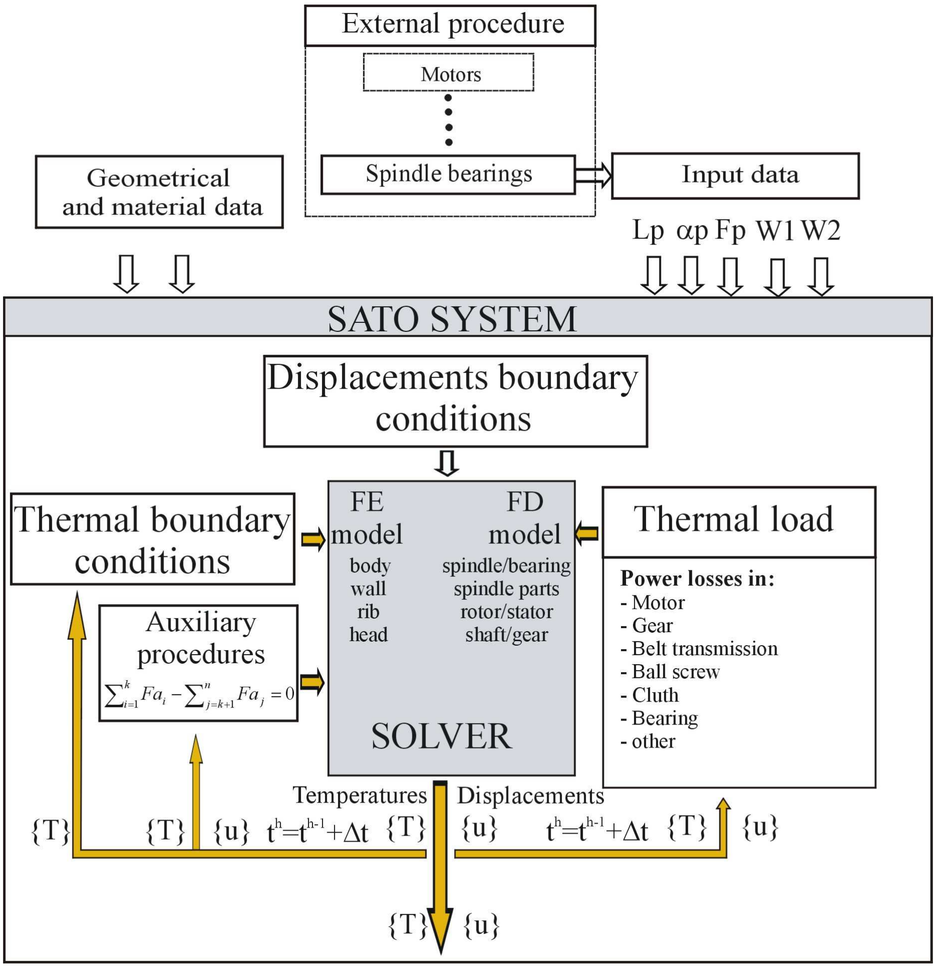


Fig. 1. Dedicated SATO system general structure

The result of the operation of such algorithm are: after-assembly load of a bearing set Fap or negative after-assembly clearance Lp, after-assembly contact angles α_p, as well as resulting interferences or clearances between a bearing and housing W1p and between a bearing and spindle W2p. Fit W1p, W2p take into account not only dimensional deviations of a hole in a housing and spindle pin, but also surface quality expressed as roughness height Rz.

Angular contact ball bearings are assembled in sets of two or more bearings, arranged in “X”, “O” arrangements or their combinations. Each of bearings in such set can have different assembly conditions, due to e.g. varying dimensions of a hole in the spindle or dimensions of the external diameter of a housing (Fig. 3).

Thermal changes of bearings set node dimensions during operation disrupt the elastic force equilibrium on both sides of a set, taking place in an assembled bearing set, and lead to the establishment of a new equilibrium of axial forces, but on a different level. In special cases it can lead to partial or full unload of bearings. The consequence of such phenomenon will be the alteration of after-assembly bearing parameters, such as: Fap, Lp, α_p, as well as W1p and W2p, which significantly influence stiffness, power losses and lifetime of bearings.

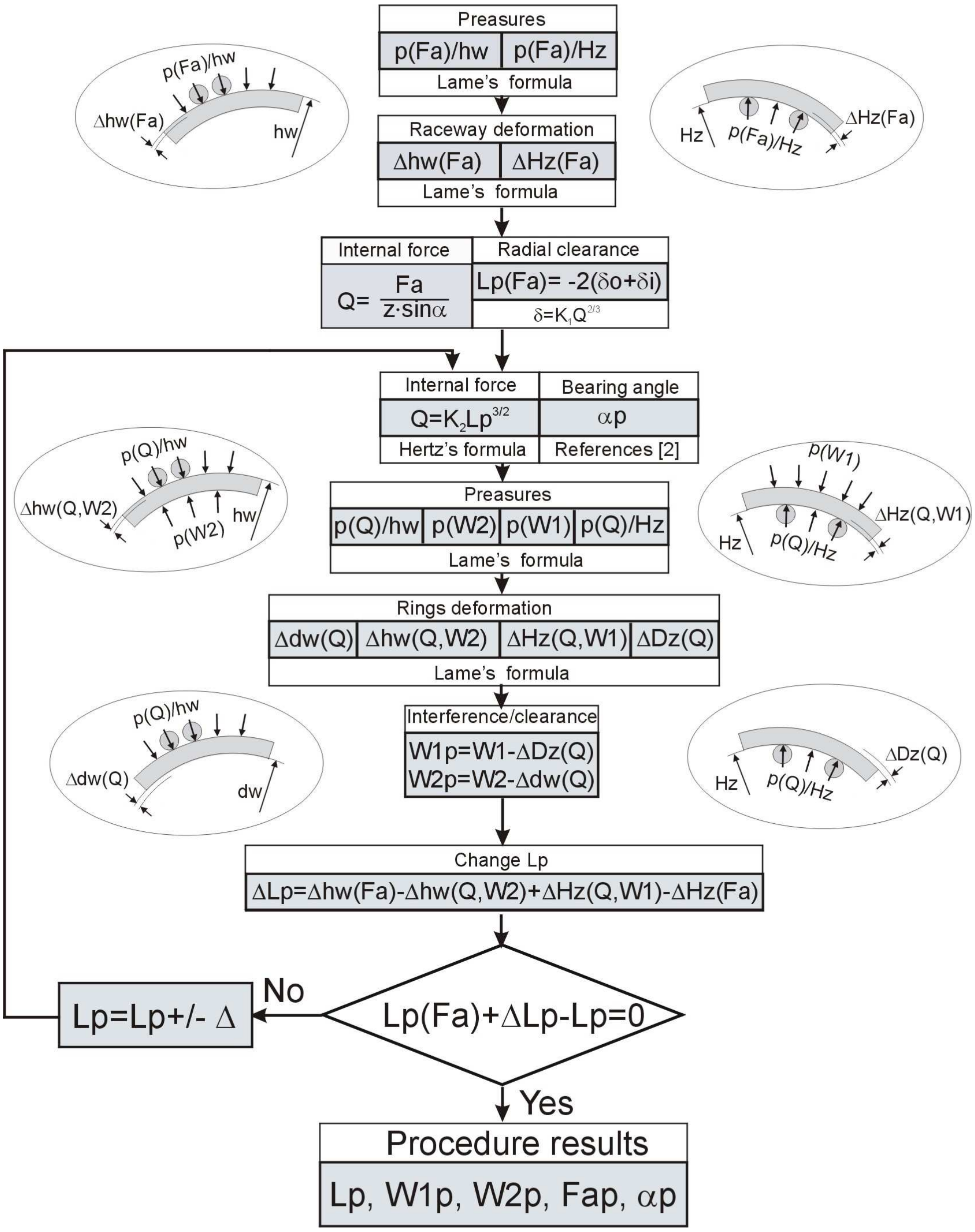


Fig. 2. Algorithm for determining parameters of angular ball bearing after its assembly [9]

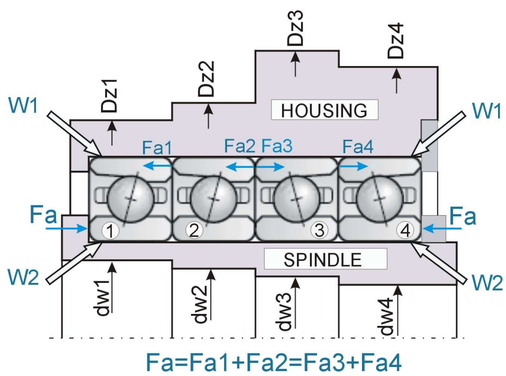


Fig. 3. Equilibrium of axial forces in a set of four preloaded bearings

The search for force equilibrium parameters is conducted iteratively in order to fulfil the following condition:

$$\sum_{i=1}^{k} Fa_i - \sum_{j=k+1}^{n} Fa_j = 0 \tag{1}$$

where:

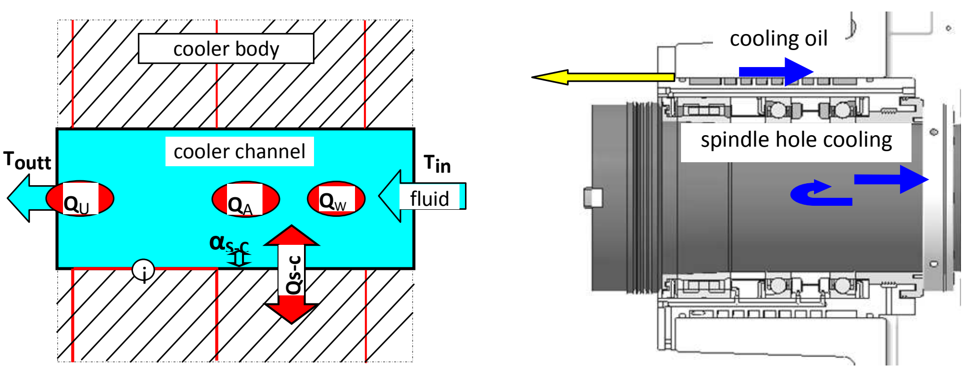
Fa_i – axial force loading the *i*-bearing on the left side of a set,
Fa_j – axial force loading the *j*-bearing on the right side of a set,
k – amount of bearings on the left side,
n – amount of bearings in a set.

Balancing of forces in a bearing set, in the SATO system, is realised for each time step. The determination of a new force equilibrium in bearings, new operation temperature, thermal changes in the dimensions of a housing, spindle and bearing elements, is needed for the automatic update of power losses in bearings and other kinematic pairs, update of heat exchange conditions, etc.

3. MODELLING OF HEAT EXCHANGE INSIDE HEADSTOCK

Constructional solutions of high-speed, as well as high-torque spindle drives, usually require intensive cooling. This usually applies to motors, spindle bearings, bearing holes, internal surfaces of a housing and sometimes also gear transmissions, their shafts and bearings. In such cases, of dominant importance for the correctness of obtained temperatures and thermal deformations, is modelling of heat exchange in different types of coolers, as well as modelling of heat exchange between the coolant and gear transmissions, shafts and headstock walls.

For this purpose, two types of volumetric elements have been defined - "fluid" and "air" types. First of them, Fluid1, is used for modelling fluids flowing through channel coolers, e.g.: spindle bearing coolers, spindle or motor coolers (Fig. 4).



Element type „Fluid1”

$$\sum_{i=1}^{n} Q_{S-C}^{i} + Q_W + Q_A + Q_U = 0$$

i – number of surface element

Fig. 4. Model of heat exchange in a bearing cooler and in a spindle hole cooler

In many complex headstocks of large machine tools, both air and oil fed to rotating elements can be observed. The purpose of such is both lubrication of kinematic pairs and cooling of internal elements of the housing (Fig. 5).

For modelling of heat exchange to the fluid flowing on internal housing surfaces, which at the same time exchanges heat with air, a Fluid2 element is used (Fig. 5). The second type of "air" element allows modelling heat exchange between air and surfaces inside the headstock. Air can circulate inside the housing closed space, as well as flow through it in order to cool or heat it.

Heat fluxes included in balances shown in Fig. 4 and Fig. 5 are:

$Q_{S\text{-}C}$ – heat flux exchanged between a wall and fluid,
Q_W – flux of heat generated in fluid as a result of its flow resistances in a cooler channel,
Q_U – flux of heat carried by fluid or air,
Q_A – flux of heat accumulated in fluid or air,
$Q_{P\text{-}C}$ – heat flux exchanged between air inside headstock and fluid,
$Q_{S\text{-}P}$ – heat flux exchanged between a wall and air inside housing.

The model of heat exchange through cooler channels utilises FDM and FEM discretisations of a headstock, as well as the heat balance equation for transient heat conduction. With the use of "Fluid1" element it describes heat balance concerning the cooling fluid and heat flux on a cooled housing surface presented in Fig. 4. Fluid movement in the cooling channel ensures convective heat exchange between fluid and wetted surfaces of channel walls. Heat flux exchanged between the channel wall and fluid is in form:

$$\mathbf{Q}_{S\text{-}C} = \boldsymbol{\alpha}_{S\text{-}C} \sum_i \mathbf{F}_i \,(\mathbf{T}_{Si} - \mathbf{T}_C) \tag{2}$$

where:

$\mathbf{T}_C$ – temperature of fluid in the channel (in the area of taking over heat from the cooler). It was assumed that it is the mean temperature of fluid temperatures at the input to the cooler T_{in} and at the output T_{out} $\mathbf{T}_C = 0.5(\mathbf{T}_{in} + \mathbf{T}_{out})$,
$\mathbf{T}_{Si}$ and $\mathbf{F}_i$ – temperatures and surfaces of discrete elements "i" which exchange heat with fluid,
$\boldsymbol{\alpha}_{S\text{-}C}$ – coefficient of forced convection on channel surfaces.

The value of this coefficient depends on properties and parameters of fluid flow, construction and dimensions of flow channels.

Second heat flux $\mathbf{Q}_W$ which appears in the balanced fluid is a result of resistances to its flow through sleeve channels and is defined by a relationship:

$$\mathbf{Q}_W = \Delta p \, q_v \tag{3}$$

in which:

q_v – volumetric flow intensity,
Δp – loss of fluid pressure resulting from movement resistances.

Last elements of Fluid1 type element are heat fluxes: carried $\mathbf{Q}_U$ and accumulated in flowing fluid $\mathbf{Q}_A$. These fluxes are defined by relationships:

$$\mathbf{Q}_U = q_m \, c_p \, (\mathbf{T}_{out} - \mathbf{T}_{in}) \tag{4}$$

$$\mathbf{Q}_A = m_c \, c_p \, (\mathbf{T}^h{}_{out} - \mathbf{T}^{h-1}{}_{out}) \, / \Delta t \tag{5}$$

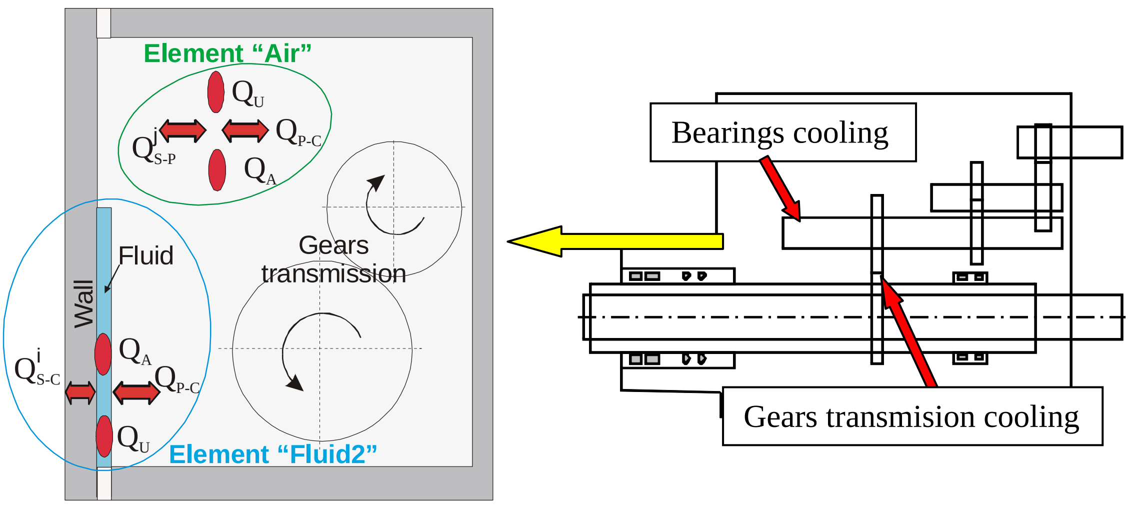
in which:
q_m – mass intensities of fluid flow,
c_p – fluid specific heat,
m_c – fluid mass contained in cooler channels,
$\Delta t = t^h - t^{h-1}$ time segment between analysed time level h and previous one h-1.

Full heat flux balance equation in transient state for "Fluid1" element is in form:

$$\mathbf{Q}^h{}_{S\text{-}C} + \mathbf{Q}^h{}_W + \mathbf{Q}^{h,h-1}{}_A + \mathbf{Q}^h{}_U = 0 \tag{6}$$

The inclusion of this equation to balance equations of remaining discrete elements allows determining temperature distributions on a fluid-cooled element and fluid outlet temperature.

Fluid movement inside the housing causes convective heat exchange between this fluid and headstock internal surfaces, as well as with air inside it and is described by a "Fluid2" type element (see Fig. 5).



Element type „Fluid2"

$$\sum_{i=1}^{k} Q^i_{S-C} + Q_{P-C} + Q_A + Q_U = 0$$

i – number of surface element with fluid layer

Element type „Air"

$$\sum_{j=k+1}^{n} Q^j_{S-P} + Q_{P-C} + Q_A + Q_U = 0$$

j – number of surface element without fluid layer

Fig. 5. Model of heat exchange inside the headstock with oil fed to gear transmissions and shafts

Heat fluxes characteristic for the balance of Fluid2 type element are described by the following relationships:

$$\mathbf{Q}_{S\text{-}C} = \boldsymbol{\alpha}_{S\text{-}C} \boldsymbol{\Sigma}_i \mathbf{F}_{i\,(TSi\,-TC)} \tag{7}$$

$$\mathbf{Q}_{P\text{-}C} = \boldsymbol{\alpha}_{P\text{-}C} \mathbf{F}_j (\mathbf{T}_P - \mathbf{T}_C) \tag{8}$$

where:

$\mathbf{T}_C$ – temperature of fluid in the area of heat exchange with wetted surfaces inside the housing defined as $\mathbf{T}_C = 0.5(\mathbf{T}_{in} + \mathbf{T}_{out})$,

$\mathbf{T}_{Si}$ and $\mathbf{F}_i$ – temperatures and surfaces of discrete elements "i" which exchange heat with fluid,

$\mathbf{F}_j$ – surfaces of discrete elements "j" covered by fluid, exchanging heat with air,

$\mathbf{T}_P$ – air temperature inside headstocks. In case of headstocks with air flow it was assumed that it is a mean temperature of air inlet T_{we} and outlet T_{wy} temperatures,

$\boldsymbol{\alpha}_{P\text{-}C}$ –coefficient of forced convection between fluid and air inside housing. The value of this coefficient depends on properties and parameters of fluid flow, as well as on construction and dimensions of wetted elements in housing.

Heat flux accumulated in fluid $\mathbf{Q}_A$ and carried by fluid $\mathbf{Q}_U$ are defined according to relationships 4 and 5. Heat flux which is a result of fluid flow resistances inside housing $\mathbf{Q}_W$ can be neglected, then heat flux balance equation in transient state for "Fluid2" element will be in the following form:

$$\mathbf{Q}^h_{S\text{-}C} + \mathbf{Q}^h_{P\text{-}C} + \mathbf{Q}^{h,h\text{-}1}_A + \mathbf{Q}^h_U = 0 \tag{9}$$

Air and fluid movement inside housing causes convective heat exchange between air and dry surfaces which are in contact with it, as well as with fluid. To describe such exchange, "Air" type element is used and fluxes taking place in it are in the following form:

$$\mathbf{Q}_{S\text{-}P} = \boldsymbol{\alpha}_{S\text{-}P} \boldsymbol{\Sigma}_i \mathbf{F}_i (\mathbf{T}_{Si} - \mathbf{T}_p) \tag{10}$$

$$\mathbf{Q}_{P\text{-}C} = \boldsymbol{\alpha}_{P\text{-}C} \mathbf{F}_j (\mathbf{T}_P - \mathbf{T}_C) \tag{11}$$

where:

$\mathbf{T}_{Si}$ and $\mathbf{F}_i$ – temperatures and surfaces of discrete elements "i" which exchange heat with fluid,

$\boldsymbol{\alpha}_{S\text{-}P}$ – coefficient of convection between dry surface and air inside housing.

Heat flux accumulated in air $\mathbf{Q}_A$ and carried by it $\mathbf{Q}_U$ is defined based on analogous relationships as (4 and 5). Only difference is that q_m, c_p, m_c properties and temperatures relate to air inside housing. Heat flux which is a result of air flow resistance inside housing $\mathbf{Q}_W$ can be neglected, similarly as the flux connected with resistances in a fluid. Flux balance equation in transient state for "Air" element is in form:

$$\mathbf{Q}^h_{S\text{-}P} + \mathbf{Q}^h_{P\text{-}C} + \mathbf{Q}^{h,h\text{-}1}_A + \mathbf{Q}^h_U = 0 \tag{12}$$

Presented models of heat exchange in coolers and inside complex headstocks require the user to have broad experience, favourably based on experimental analyses and available literature [1],[3],[4],[8]. It is because in order to achieve good results of analyses it is required to conduct proper division of internal surfaces to elements covered by oil and dry ones, as well as to determine proper values of forced convection coefficients.

4. HEADSTOCK THERMAL MODEL AND ITS VERIFICATION

For the drilling-milling machine headstock shown in Fig. 6 a discrete model has been elaborated, which is based on a simplified CAD 3D model, where details were omitted which are insignificant for the behaviour of the computed structure, such as: screws, holes, wires, etc. During the elaboration of such model, efforts were made to maintain such structure, so in the process of applying boundary conditions it would be easy to identify important construction elements, such as: coolers, bearings, gear wheels, etc.

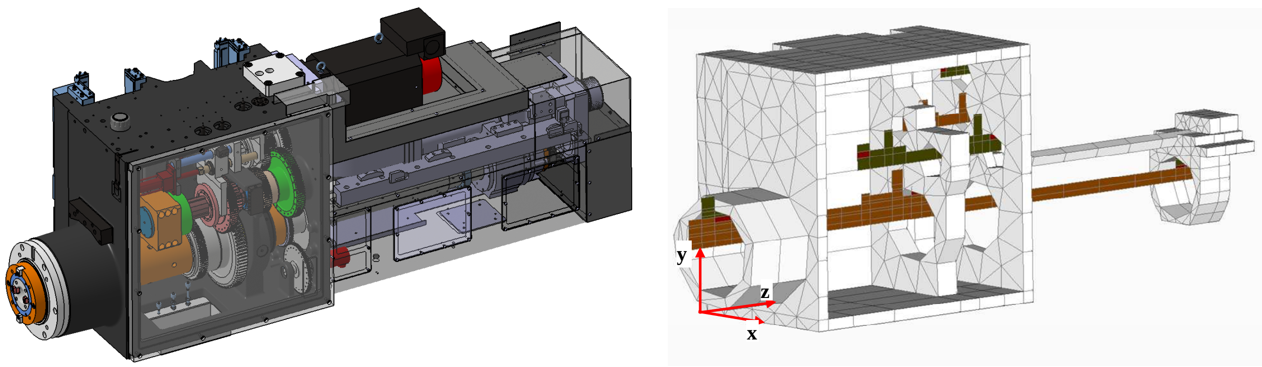


Fig. 6. Discrete model based on a headstock simplified geometry: - prismatic elements (body, ball screw, ball nut housing), - cylindrical elements (spindle, bearings, gear wheels, shafts, nut)

In a temperature model of a headstock with intermediate shafts and gear transmissions equipped with bearings, placed inside the housing, it was required to take into consideration a large number of heat phenomena. The most important of them are:

- stabilised and transient heat conduction in the housing material and in spindle unit drive elements,
- heat exchange force by fluid flow through:
  - sleeve for cooling of front spindle support roller bearings,
  - spindle hole,
- heat exchange by a cooling-lubricating fluid fed to:
  - gear wheels, intermediate shafts and their bearings,
  - internal housing surfaces,
- heat exchange with air inside housing,
- heat exchange by couplings,

- transmission of heat to the environment by forced and free convection and by radiation,
- generation of heat flux by spindle and shaft bearings and by gear wheels.

In a displacement calculation model, proper degrees of freedom were constrained in FEM mesh nodes located on slideways and in the location of transmitting feed movement on the headstock.

In order to verify the presented approach to modelling headstock with complex drive, assessment of compliance between the calculated temperature values and these from experiments was made (Fig.7).

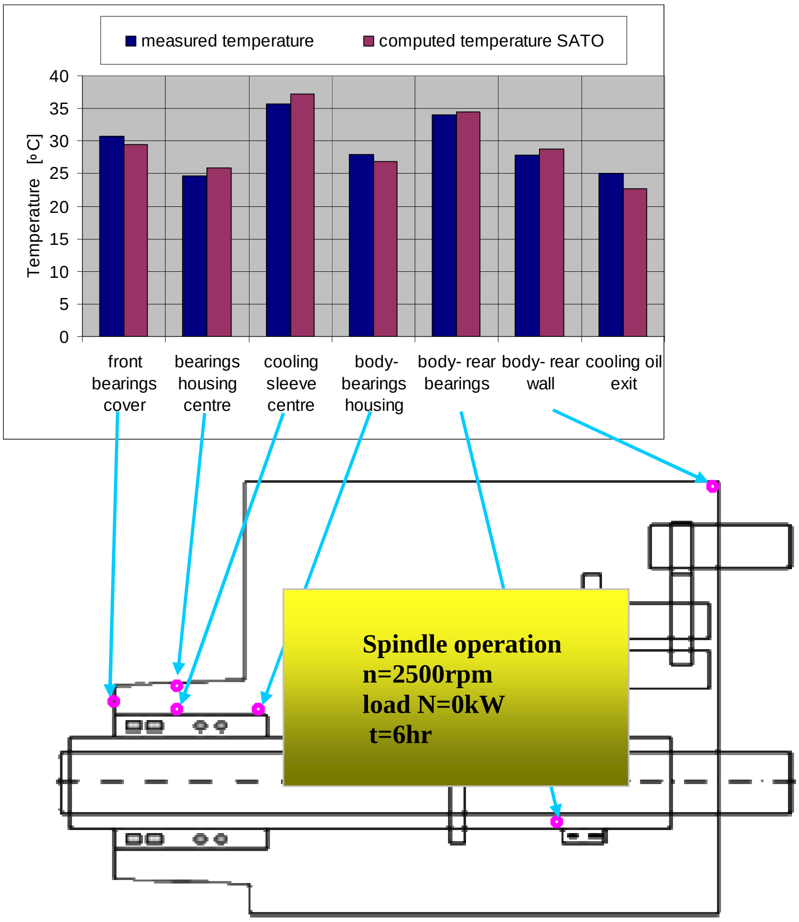


Fig. 7. Comparison of computed and measured temperatures for selected headstock points

Computations were carried out for the following conditions:
- lubrication of front spindle bearings by oil mist,
- stream lubrication of gear wheels and remaining bearings,
- oil flow intensity through:
  sleeve for cooling front bearings 14,8l/min,
  spindle hole 6,4l/min,

headstock interior, gear wheels, shafts and their bearings 8,2l/min,

- ambient temperature 19,5^{o}C,
- cooling oil inlet temperature 20^{o}C.

Discrepancy between measured and computed temperatures in seven points on the structure was in range of 0,5^{o}C to 1,5^{o}C.

The quality of elaborated headstock thermal model was also assessed in transient state (see Fig. 8), in case of which it is possible to observe transient phenomena and these can have significant influence on headstock thermal behaviour. Such types of phenomena can be seen in Fig. 8a and 8b, on which the character of measured temperatures in some points of the housing differs from typical runs, such as the one recorded in Fig. 8c.

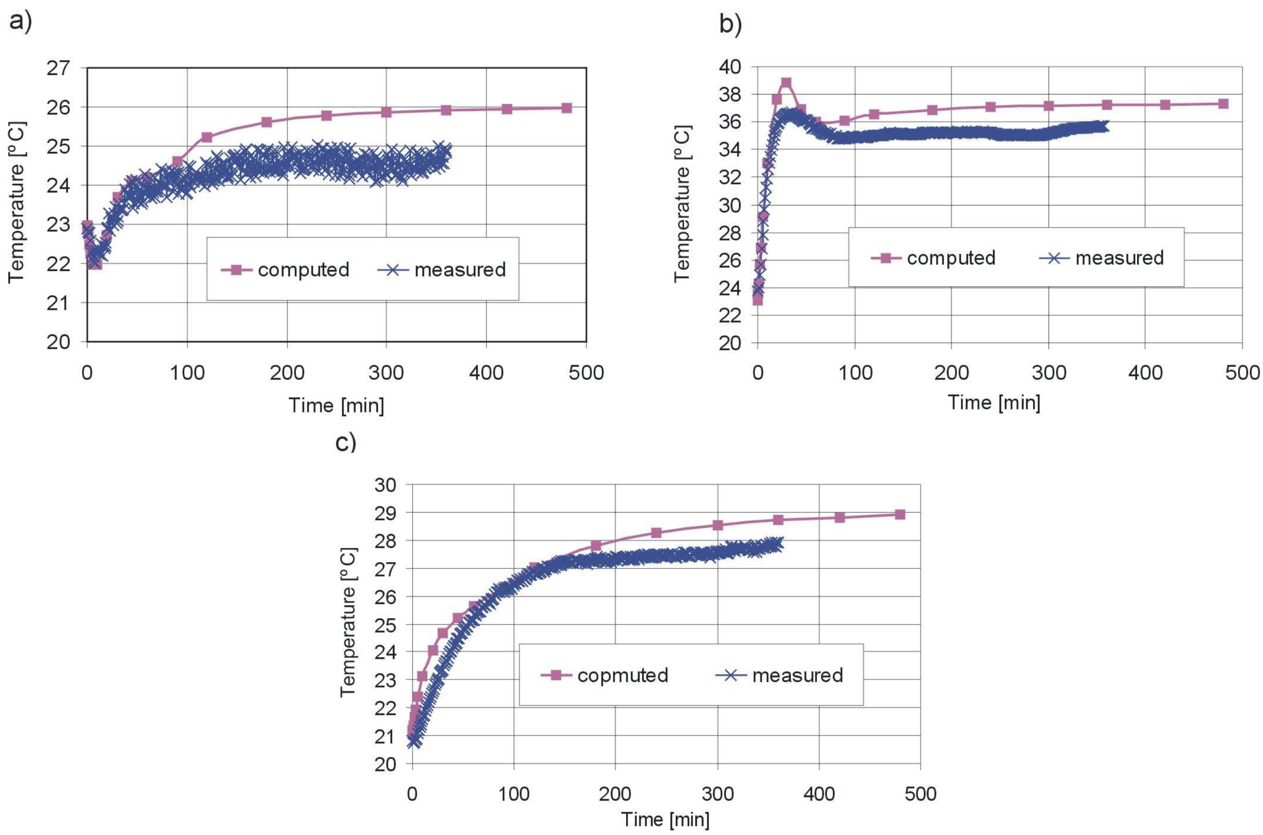


Fig. 8. Verification of headstock temperatures in transient state for:
a) head containing front spindle bearings, b) sleeve for cooling front bearings, c) point on a housing far from coolers

The computation of such non-monotonic behaviour of temperature in point of front bearings housing centre and in point of cooling sleeve centre was possible thanks to precise modelling, mainly thermal-elastic and hydrodynamic phenomena in bearings, as well as fluid cooling.

Despite the discrepancy of calculated and measured transient temperatures are slightly higher than for thermal state treated as stabilised, it can be stated that the elaborated headstock model is able to ensure proper character of temperature run in time not only for monotonic runs (Fig. 8a, 8c), but also for more complex runs (Fig. 8b).

5. CONCLUSIONS

Aspects discussed in this paper are a small part of computational models which should be applied to fully represent thermal phenomena, which accompany the operation of headstocks. Often very simplified models are used or modelling of some phenomena is omitted, assuming that e.g. they have little impact on final results. It is sometimes very risky, which is proven by the results shown in Fig. 8, where it is presented that the structure heating-up curve does not have to have a monotonic character in all points. Proper representation of so much different thermal characteristics in various areas of the same object is the best verification of proper taking into account present phenomena, whether commonly applied simplifications were really insignificant. Based on conducted computational analyses it can be stated that it is definitely important that modelling of bearings as heat sources should include main conjugations of heat losses with thermal deformations of a spindle, housing, distance rings and with temperatures. Force balance should also be monitored in preloaded bearing sets. Additionally, modelling of heat transmissions in cooling system for various flow speeds with heat carrying is very important, similarly as modelling of complex force convection functions on external walls of headstocks.

ACKNOWLEDGEMENT

This work was supported by Doosan Infracore, Korea. The authors would like to thank the advanced development team of Doosan for providing valuable insight and advice.

REFERENCES

[1] BUCHMAN, K., JUNGNICKEL, G., 1978, Die Wärmeubertragung an Be-und Verarbeitungmaschinen, PWR Wroclaw, ITMiA, 20, 279, (in polish).
[2] HARRIS T. A, KOTZALAS M. N., 2006, *Rolling Bearing Analysis*, Fifth Edition, Taylor&Francis Group.
[3] WINIARSKI, Z., KOWAL Z., BŁAŻEJEWSKI A.., 2008, Decreasing of thermal errors in a lathe by forced cooling of ball screws and headstock, Journal of Machine Engineering, 8/4, 122-130.
[4] JĘDRZEJEWSKI J., MODRZYCKI W., KOWAL Z., KWAŚNY W., WINIARSKI Z., 2007, *Precise Modelling of HSC Machine Tool Thermal Behaviour*, Journal of Achievements in Materials and Manufacturing Engineering, 24/1, September, 245-251.
[5] JĘDRZEJEWSKI J., KOWAL Z., KWAŚNY W., MODRZYCKI W., 2004, *Hybrid model of high speed machining centre headstock*, Annals of the CIRP, 53/1. 285-288.
[6] KOWAL Z., WINIARSKI Z., 1990, *Computer system for analysis and optimisation of thermal deformations in machining centres*, Przegląd Mechaniczny, 49, 21/22, 42-46, (in polish).
[7] MAEDA O., CAO Y., ALTINTAS Y., 2005, *Expert spindle design system*, International Journal of Machine Tools & Manufacture, 45, 537–548.
[8] WINIARSKI Z., 2006, *Thermal analyses in designing electrospindle of modern machine tools.* Mechanik, 03, SIMP, Warszawa, 230-234, (in polish).
[9] WINIARSKI Z., KOWAL Z., KWAŚNY W., HA J.,Y., 2010, *Thermal behaviour modelling of a spindle with a complex drive system*, Inżynieria Maszyn, 15/1, 114-127, (in polish).