THERMO-MECHANICAL ANALYSIS OF A MACHINE TOOL WITH HYDROSTATIC BEARINGS

EDUARD STACH, JAN SMOLIK, MATEJ SULITKA, TOMAS LAZAK, IVAN DIVIS, JIRI FALTA

Czech Technical University in Prague, Faculty of Mechanical Engineering, Department of Production Machines and Equipment (RCMT), Prague, Czech Republic

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e.stach@rcmt.cvut.cz

Thermal behaviour is a key factor affecting the operating status of high performance hydrostatic (HS) bearings of rotational axes of machine tools (MT). This paper introduces a novel model developed for transient thermo-mechanical analysis of hydrostatic guideways with a surrounding MT structure. In contrast to common approaches to HS bearing design, which only consider the static load carrying capacity, the new model enables detailed prediction of the bearing thermal stability under various operating conditions. An analytical description of the HS bearing heat generation coupled with a MT finite element (FE) model enables calculation of heat transfer, conduction and especially thermal deformations of the entire MT structure affecting the working accuracy of the machine tool.

KEYWORDS:

Hydrostatic, machine, tool, thermal, behaviour, speed, lathe

1 INTRODUCTION

Hydrostatic (HS) guideways are considered one of the fundamental types of sliding guideways in the machine tool (MT) industry. HS guideways are used for their low friction, high precision, load capacity, stiffness and damping capability. HS guideways usually consist of multiple HS cells (a guiding surface with a facing HS pocket connected to an oil flow regulator and oil supply), which are generally able to bear loads in the direction normal to the guiding surface.

During HS guideway development, a distinction is made between HS guideways operating under low feed or peripheral speeds (quasi static) and high-speed HS bearings which tend to generate a large amount of heat. When designing quasi static HS guideways, attention is generally focused on careful inspection of all possible load cases and their influence on the load carrying capacity, stiffness and energy demands of each HS cell, as contact between the two main parts of the HS cell should be prevented during operation. When HS bearings are subjected to higher speeds, the heat losses cause thermal deformations of the machine tool's structural parts, affecting the load carrying capacity and parallelism of the guideways, which may thus end up in contact, resulting in accidental destruction (loss of geometry due to abrasion).

The properties of hydrostatic guideways can be described in conventional ways [Porsch 1969], [Perovic 1977], [Schlotterbeck 1964], [Dittrich 1967], [Effengerger 1970], [Weck 2006], [Perovic 2012], where the influence of the flexible structure of the machine tool is neglected. On the other hand, it is possible to use modern unconventional approaches which enable the coupling of the nonlinear force deflection behaviour of the HS guideways with the properties of the machine tool's flexible structure. Unconventional approaches allow for a relatively accurate description of the oil flow in the HS cell itself and the simulation of thermo-mechanical interaction of the HS cell with the structure of the machine.

1.1 Conventional approach to description of hydrostatic guideways

One of the well-known modifications of the Hagen-Poiseuille equation describes the flow through a thin slot of width b_s and height h, assuming that the slot width is significantly greater than the slot height [Mathieu 1863]:

$$Q = \frac{1}{12}\left(\frac{\Delta p}{l}\right)\frac{b_s h^3}{\mu} \tag{1}$$

where l denotes the slot length analogous to the length of the capillary tube in the Hagen-Poiseuille equation. Equation (1)(1) was used by D. D. Fuller to describe a hydrostatic bearing in 1947 [Fuller 1947] and has since been the fundamental equation used to describe the oil flow rate and throttling gap height of hydrostatic cells.

Reynolds provided a more comprehensive description of flow through a thin slot [Reynolds 1884], [Reynolds 1886], [Reynolds 1902]. Reynolds' work was based on a simplified form of the Navier-Stokes equation [Navier 1823], [Stokes 1849] in the form:

$$\frac{\partial p}{\partial x} = \frac{\partial^2 u}{\partial y^2}, \quad \frac{\partial p}{\partial z} = \frac{\partial^2 w}{\partial z^2} \tag{2}$$

combined with the continuity equation:

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} + \frac{\partial w}{\partial z} = 0 \tag{3}$$

By adjusting and applying the boundary conditions, we get the original form of the Reynolds equation, the differential pressure distribution equation considering the velocity of the guide surfaces and their contour $h_{(x)}$ [Hersey 1966]:

$$\frac{\partial}{\partial x}\left(h^3\frac{\partial p}{\partial x}\right) + \frac{\partial}{\partial z}\left(h^3\frac{\partial p}{\partial z}\right) = 6\mu\left[(U_0 + U_1)\frac{\partial h}{\partial x} + 2V_1\right] \tag{4}$$

[Sharma 1995], [Sharma 2010] and [Zuo 2013] describe the oil flow through a throttling gap using the Reynolds equation, where the solution is approximated by the Galerkin method. [Sharma 1995] describes a comparative analysis of the properties of a radial hydrostatic bearing consisting of four and six hydrostatic cells and in [Sharma 2010], an analysis of the properties of a unidirectional radial-axial hydrostatic bearing is presented. In [Zuo 2013], the Reynolds equation is used for a comparative analysis of the properties of a bidirectional radial-axial hydrostatic bearing with constant flow control and a self-compensating design.

In [Hanek 2012], the flow in the entire hydrostatic cell is solved using Navier-Stokes equations in 2D, and the finite element method (FEM) is used for the approximate solution. The results of the work in [Hanek 2012] were successfully compared in [Hanek 2013] with an experiment and with a simplified description using hydraulic resistances.

This approach is relatively demanding, but it allows for a detailed description of a hydrostatic cell that is not planar. Similarly, to other conventional approaches, the disadvantage of this approach is that the interaction between the hydrostatic cells and the machine tool structure is not considered. In [Kane 2003], the author describes a radial-axial hydrostatic bearing solution of the same design as in [Zuo 2013] by simplifying it to 1D flow and solving it using the hydraulic resistance method, which is less demanding and more flexible in describing the properties of the entire machine.

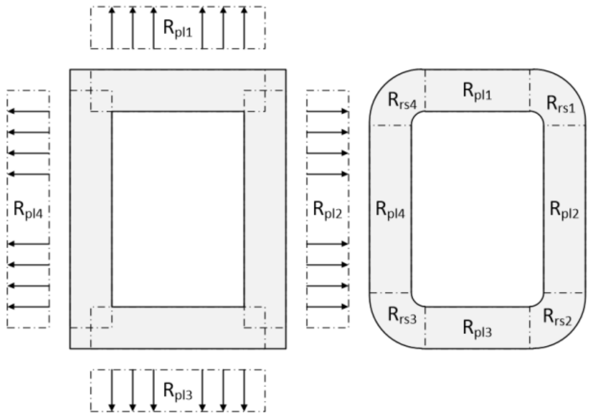

Figure 1. Division of HS pocket land into rectangular and inter circular segments

The method described e.g. by [Slocum 1992], [Kane 2003], [Weck 2006] and [Perović 2012] uses the analogy of a hydraulic circuit with an electrical circuit (electric voltage analogous to pressure p and electrical current to fluid flow). Based on the given analogy, the hydraulic resistance is defined as:

$$R = \frac{\Delta p}{Q} \quad (5)$$

The electrohydraulic analogy method was used in [Slocum 1992] to describe the flow of oil through a hydrostatic pocket of known geometry. A rectangular hydrostatic pocket, as seen in Fig. 1, consists of four straight slots (right) and sometimes four round corner slots (left). Flow through a straight slot is solved using the reduced Navier-Stokes (N-S) equation assuming fully developed laminar flow. The resulting relationship, which corresponds to equation (1), is adjusted to the following form using equation (5):

$$R_s = \frac{12\mu l}{b_s h^3} \quad (6)$$

where R_s denotes the hydraulic resistance of the slot. The hydraulic resistance of the corner slots of the hydrostatic pocket can be solved using the Navier-Stokes equation in polar form. The total hydraulic resistance of the hydrostatic pocket R_k is given by the sum of the reciprocals of the partial resistances.

[Slocum 1992] and [Weck 2006] described the hydraulic resistance of basic types of flow regulators when solving for the flow through a hydrostatic pocket. By solving the circuit, in more complex cases using Kirchhoff's laws, it is possible to determine the flow through the hydrostatic pocket and the appropriate regulator, or through the entire guideway. Fig. 2 shows an example of such a network of hydraulic resistances of HS pockets and radial-axial bearing flow regulators.

The advantage of the electro-hydraulic analogy combined with the solution of reduced (N-S) equations is the use of the hydraulic resistances of the sub-elements of the hydrostatic system, which makes it possible to describe the individual parts of the hydrostatic pocket and the circuit separately as discrete modules. This approach makes it possible to modify known solutions quickly and efficiently according to the current requirements of the geometry of the pockets or the arrangement of the hydrostatic guideway. On the other hand, this approach neglects the interaction of the hydrostatic cells and the machine tool structure.

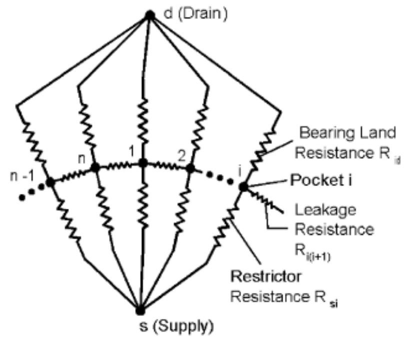

Figure 2. Network of hydraulic resistances [Kane 2003]

1.2 Unconventional approach to description of hydrostatic guideways

The most advanced solutions presented in the literature combine a numerical solution of fluid flow in CFD and structural (or thermo-mechanical) analysis using the finite element method.

[Lihua 2012] studied the static stiffness of the hydrostatic cell of a hydrostatic spindle. CFD analysis of the temperature and pressure field was performed in Fluent software by solving the Navier-Stokes equations. The results of the solution in Fluent were used as loads for the structural analysis in the Ansys FEM solver. Due to the pressure and temperature field, there was a significant decrease of almost 80% in the static stiffness of the hydrostatic cell, mainly due to its structural deformations. Another interesting conclusion is the 2% difference between the conventional cell flow calculation and the isothermal CFD calculation.

[Su 2014] described a hydrostatic spindle model that is solved using the finite volume method and the finite element method. The flow and heat generated, as well as the heat transfer between the oil and the guide surfaces, were analysed in the Fluent program, which was interfaced with the Ansys for structural analysis. This calculation procedure does not allow for the inclusion of the viscous friction loss inside the hydrostatic pocket cavities in the calculation.

In [Kim 2006], the structural properties of a centreless grinding machine and the effect of concrete filling in the casting bed were studied. In the FEM model of the structure of the entire machine, the linear stiffnesses of the moving screws, rolling and hydrostatic guides are replaced by a matrix element. The paper evaluates the deformations of the structure, including the tilting of the guiding surfaces of the hydrostatic cells. The paper does not discuss the influence of tilting on the operational parameters of the HS guideways.

In [Chen 2011] and [Chen 2013], a study of the influence of temperature deformations of the hydrostatic spindle on the running accuracy using the thermo-mechanical FEM model was presented. The articles describe the power loss of the integrated servo motor and of hydrostatic guideways, which is further divided into "hydraulic power loss" and "frictional power loss" on the effective friction surface, which unfortunately is neither cited nor described. However, it is evident from the calculation that viscous friction in the pocket cavity was not considered. The increase in oil temperature, determined from the sum of power losses, was introduced to the guiding surfaces of the hydrostatic guideways by thermal convection. Hydrostatic cells were replaced by linear stiffness. The task is solved as a stationary one, therefore the change of HS guideway parameters and power losses due to temperature changes were not considered.

In [Liu 2015], the FEM temperature analysis of a sliding axis with hydrostatic guideways with a self-regulating design was described. This paper examined the influence of inlet oil temperature, ambient temperature and hydraulic losses on the thermal behaviour and thermal deformation of the sliding axis.

In [Yang 2015], the effect of the throttle gap height on the oil temperature and the structural deformation of a two-column vertical turning lathe, enabling the grinding of large-sized aspherical glasses using FEM, was presented. The paper described models of hydraulic and frictional losses in the hydrostatic cells of the entire machine. Friction losses in pocket cavities were not considered. The friction loss model was applied to the surface of the entire pocket, i.e. including the cavity. In [Sun 2015], the results of modal and static stiffness analysis at the tool centre point of the same machine as in [Yang 2015] were described. Hydrostatic cells were modelled by a linear spring with damping in the FEM model. The sub modelling procedure was not discussed.

1.3 Summary of the state of the art

The coupling of CFD and FEM, described in [Lihua 2012] and [Su 2014], does not allow for sufficient description of the heat sources in the HS cell due to problems with the modelling of the HS pocket with land and cavity in 3D, and does not solve the heat balance of the oil flowing through the HS system (the amount of heat generated in the HS guideway, the amount of heat leaving the MT domain with the hydraulic medium and the amount of heat transferred from the hydraulic medium to the machine structure). There are currently limited possibilities for integrating CFD simulation into the structural analysis of the entire machine and these possibilities are not suitable for complex modelling of entire machines or their larger parts. Thermo-mechanical finite element analysis, when coupled with analytical descriptions of HS guideway nonlinear properties, enables extensive tasks to be solved with accurate results. In [Kane 2003], the analytical solution of the Navier-Stokes equation of a radial-axial hydrostatic bearing, where the guiding surfaces have a conical shape, was described. In [Lin 2014], a high degree of agreement between CFD and the analytical solution of the flat hydrostatic cell oil flow was described. Similar results were described in [Lihua 2012] and [Hanek 2013]. The thermo-mechanical solutions using FEM described in [Kim 2006], [Chen 2011], [Chen 2013], [Liu 2015], [Yang 2015] and [Sun 2015] do not include a complete description of heat loss flows in HS guideways nor do they solve the heat balance of the hydraulic medium, include a description of the influence of the tilting of the guiding surfaces, or update the heat loss flows or the properties of HS cells during the calculation due to the development of temperature fields and structural deformations.

Solving the heat balance of the machine tool or a relevant section of the MT with the hydrostatic bearing and updating the properties of HS cells and heat loss flows during the solution appear to be key prerequisites for a relevant simulation of the operation of a machine tool with hydrostatic guideways.

2 Methods

New requirements are placed on modern hydrostatic guides by MT manufacturers, such as the ability to compensate for geometric errors caused by temperature deformations, gravitational forces during machine configuration changes, cutting forces or inertial effects. Other requirements include increasing movement speeds, low heat sources, low passive resistance at high speeds, low energy consumption and minimized requirements for production accuracy. In order to develop a comprehensive description of the interaction of hydrostatic guideways with the flexible structure of the machine tool, it is necessary to move beyond the more widespread conservative approach to modelling HS guideways as a completely isolated part of the machine tool. Hydrostatic guideways are affected by the flexible MT structure and, conversely, the machine structure is affected by the operation of the HS guideways, which in modern high-speed applications act as a significant source of heat. The development of a complex model of the HS guideways aims to describe the HS guideways' thermal behaviour and its influence on the temperature field and thermal deformations of the machine structure.

In general, the most suitable tool for the analysis of temperature and deformation fields in bodies is the finite element method (FEM). The accuracy of the analysis is only a matter of appropriate discretization and abstraction of the problem. FEM solvers enable calculations with various types of nonlinearities and elements containing mixed thermo-mechanical properties.

For a complex model of hydrostatic guideways, the temperature field in the investigated machine is obtained by solving the thermal FEM problem and then applied in the structural FEM problem, and thus a deformed shape is obtained. The internal mechanical state of each hydrostatic cell is exported and the characteristics and other partial descriptions of the HS guideway behaviour are updated, which are then introduced into the next iteration of the thermal and structural problem. The entire problem is solved using a pre-programmed sequence of thermal and mechanical FEM problems supplemented by updating the elementary characteristics and partial descriptions of the HS guideways. When a sufficiently short step is chosen, the behaviour of the monitored parameters of the discrete model approaches a real continuous event.

A comprehensive model of hydrostatic guideways consists of:

- A detailed finite element model of the MT with sub models of HS cells,
- A description of the force-deformation characteristics of hydrostatic cells,
- A description of the effect of tilting the guide surfaces of HS cells on the force-deformation characteristics (as described in [Stach 2016]),
- A description of the heat sources in the HS cells,
- A description of the heat transfer between the hydraulic medium and the structure inside the HS cell,
- A description of the heat transfer between the hydraulic medium and the structure at the outlet from the HS cell and the path to the outlet.

A complex model alone does not solve the requirements of machine tool manufacturers. However, it is an important tool that enables designers to meet the requirements for minimizing thermal deformations, heat sources, passive resistance and energy consumption and increasing movement speeds.

2.1 Conservative model of hydrostatic bearing and representation in FEA

The key component of a complex model of a machine tool with hydrostatic guides is a mathematical description of the most important properties of the hydrostatic cell (load capacity, stiffness) and a methodology for modelling its interaction with a flexible structure. The principle governing HS guideway function results in fundamental requirements for parallelism of the guide surfaces of the guideway and for high MT structure rigidity. In reality, the guiding surfaces are never perfectly parallel. These errors are most often caused by both manufacturing tolerances and gravitational forces, cutting forces, inertial effects and the flexibility of the structure. If the structure or the layout of the guiding surfaces is designed insufficiently, the HS guideway carrying capacity is reduced, and in the worst case, the guiding surfaces come into contact during operation and are destroyed. In [Mares 2012], the positive effect of an optimized structural design and the distribution of HS cells is described.

In order to describe the flow on the land of the HS pocket (Fig. 1), a partial rectangular part of the edge was first solved. Assuming a laminar flow of a Newtonian fluid between two parallel surfaces, and due to the very small ratio of the throttle gap height to the length of the land, a fully developed flow may be considered and the Navier-Stokes equation may be reduced to the form:

$$\frac{1}{\rho}\frac{\partial p}{\partial X} = \nu \frac{\partial^2 u}{\partial y^2} \quad (7)$$

where ρ denotes the density of the hydraulic medium, $\partial p/\partial X$ denotes the pressure gradient in the flow direction (Fig. 3), u denotes the fluid flow rate and ν denotes the kinematic viscosity of the hydraulic medium. The boundary conditions in this task do not consider the relative movement of the HS guideway guide surfaces. By integrating equation (7) and applying the boundary conditions $u=0$ at $y=0$ and $y=h$, the velocity profile of the flow in the throttling gap of the rectangular part of the edge of the HS pocket was obtained as a function of:

$$u = \frac{1}{2\rho\nu}\frac{\partial p}{\partial X}(y^2 - yh) \quad (8)$$

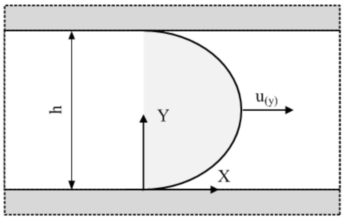

Figure 3. Fully developed flow between two parallel surfaces

The flow rate given by the parabolic velocity profile due to the pressure gradient is $\partial p/\partial X$, obtained by integration over the throttle gap:

$$Q_{pl} = \int u \, dS \quad (9)$$

$$Q_{pl} = \frac{1}{2\mu}\frac{\partial p}{\partial X}\int_0^{b_{pl}}\int_0^{h}(y^2 - yh)\, dy\, dz \quad (10)$$

$$Q_{pl} = -\frac{b_{pl}h^3}{12\mu}\frac{\partial p}{\partial X} \quad (11)$$

where b_{pl} denotes the length of the rectangular part of the land. Pressure can only be a function of position along the land. By integrating equation (11) over the width of the land in the X direction, we get:

$$\int_{p_t}^{0} dp = -\frac{12\mu Q_{pl}}{b_{pl}h^3}\int_0^{l} dx \quad (12)$$

$$p_t = \frac{12\mu l}{b_{pl}h^3} Q_{pl} \quad (13)$$

where l denotes the width (shorter side) of the rectangular part of the land and p_t denotes the pressure on the inner part of the land (pressure in the pocket cavity). Considering the electro-hydraulic analogy, equation (13) is adjusted to form the hydraulic resistance of a rectangular slot:

$$R_{pl} = \frac{12\mu l}{b_{pl}h^3} \quad (14)$$

Applying an electro-hydraulic analogy again, we can calculate the sum of the inverted values of the hydraulic resistance of all sub-rectangular parts of the land (e.g. Fig. 1):

$$R_k = \frac{1}{\sum_0^{i=4}\frac{1}{R_{pli}}} \quad (15)$$

If all lands of the pocket are rectangular and of equal width, we get:

$$R_k = \frac{12\mu l}{bh^3} \quad (16)$$

where b denotes the sum of the effective lengths of all the rectangular lands. Although it would be possible at this point to determine the load carrying capacity and, in the case of constant flow regulation, the stiffness of the hydrostatic cell from equation (16), in real applications, capillary and PM regulators are most often used, and the flow rate through the HS pocket is not only unknown, but also not constant. With known pressure at the hydraulic unit output, system flow, along with load carrying capacity and stiffness as functions of throttle gap height, are calculated.

Using the Hagen-Poiseuille equation, it is possible to describe the hydraulic resistance of the capillary regulator R_c. The hydraulic resistance of an entire branch of the hydrostatic system (HS cell with flow regulator) is described by the following relationship:

$$R_v = R_k + R_c \quad (17)$$

when the following is valid:

$$p_p = R_v Q_v \quad (18)$$

where p_p denotes the pressure at the hydraulic unit output (losses in the pipeline are neglected) and Q_v denotes the flow of the hydraulic medium through the solved branch of the hydrostatic system. From the flow continuity equation, it follows that:

$$\frac{R_v}{p_p} = \frac{R_k + R_c}{p_p} = \frac{R_k}{p_t} \quad (19)$$

By modifying equation (19), it is possible to obtain a relation for calculating the pressure in the hydrostatic pocket and the flow through the solved branch of the hydrostatic system:

$$p_t = \frac{R_k}{R_k + R_c} p_p \quad (20)$$

$$Q_v = Q_k = \frac{R_k + R_c}{p_p} \quad (21)$$

By substituting equation (20) into the relationship for calculating the total reaction force of the HS pocket, the relationship for calculating the reaction force of the hydrostatic cell was obtained:

$$F_r = A_{ef} p_t \qquad (22)$$

where A_{ef} denotes the effective area of the HS pocket. The nonlinear stiffness of the hydrostatic cell is expressed as:

$$K = \frac{\partial F_r}{\partial h} \qquad (23)$$

Hydrostatic cells are sliding guides that ensure the mutual movement of two components on a thin layer of oil film, with a thickness rarely exceeding 0.01 mm, which indicates that the correct functioning of these guides is significantly influenced by the compliance of the surrounding components and groups of the machine, which must also be analysed. A procedure describing the partial properties of the HS cell with relevant analytical relations and their implementation into a detailed finite element model of the entire machine was chosen. This procedure allows sufficient variability in the preparation of a computational model with a large number (i.e. tens) of different hydrostatic cells and their controllers. Each hydrostatic cell is replaced by its own sub model, which can be prepared automatically based on the input data.
However, after substituting into equation (22), it can be seen that the reaction force of the HS cell is a function of the third power of the throttling gap height. This means that the HS cell force-deformation and stiffness characteristics are non-linear and should not be represented by an ordinary linear spring-type element. A procedure was used that replaces the nonlinear force-deformation characteristic in FEM models of machine tools using COMBIN 39 elements and the Constraint Equation in the Ansys environment. These elements make it possible to define the nonlinear dependence of the displacement of the corresponding nodes on the load.

2.2 Heat sources in hydrostatic bearing

During fluid flow, friction against the surface of the channel occurs, causing the fluid to lose kinetic energy, which turns into heat. To maintain flow velocity, the drop in kinetic energy is compensated by potential energy, a drop in pressure at the end of the channel. This creates a pressure loss. Assuming a free flow of fluid from the hydrostatic pocket, the heat generated by pressure losses can be described by the following relation for the heat loss flow:

$$\Phi_t = p_t Q \qquad (24)$$

During the movement of the guide surfaces of the hydrostatic guideway, the work of frictional forces (tangential stresses from viscosity) causes energy dissipation. This dissipated energy (loss) changes irreversibly into heat and acts as the main source of heat in rotating hydrostatic guideways. The heat loss flow that arises due to energy dissipation on the land of the pocket (Fig. 4) is described by the relation [Rowe 2012]:

$$\Phi_l = \mu \frac{U^2}{h} S_{land} \qquad (25)$$

Energy dissipation also occurs in the hydrostatic pocket cavity, which, despite the relatively large depth, cannot normally be neglected due to the recirculation flow in the pocket cavity, which tends to generate a larger amount of heat. The heat loss flow in the hydrostatic pocket cavity due to energy dissipation is approximated by the relation [Shinkle 1965], [Slocum 1992], [Rowe 2012]:

$$\Phi_c = 4\mu \frac{U^2}{H} S_{cavity} \qquad (26)$$

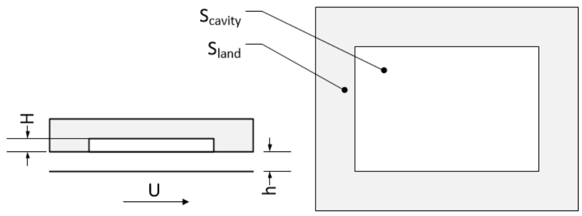

Figure 4. Schematics of the throttling gap height *h*, cavity depth *H* and the area of land S_{land} and area of cavity S_{cavity}

The total heat loss flow generated in the hydrostatic cell is described by the relation (see Fig. 5):

$$\Phi_g = \Phi_l + \Phi_c + \Phi_t \qquad (27)$$

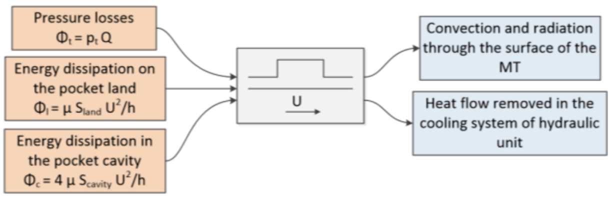

Figure 5. Heat generated in hydrostatic guideway

2.3 Heat transfer between hydraulic oil and machine tool structure

The dominant heat source in high-speed hydrostatic guideways is energy dissipation and, to a lesser extent, pressure loss. Due to the temperature difference between the hydraulic medium and the machine tool structure, the generated heat is transferred to the machine structure. Most of the lost power in the hydrostatic pocket is supplied by a drive of the given axis through the energy dissipation mechanism, and the rest is supplied by the hydraulic unit through the pressure loss mechanism. The ratio between the two is determined by the speed of movement of the guide surfaces. In the static state, heat will be generated only by pressure losses. But if we imagine the rotary axis of a machine tool with hydrostatic guideways (i.e. the table of a vertical turning lathe), then at constant revolutions all the power supplied by the drive (up to tens of kW in mid-sized and large tables) is changed into heat in the oil film layer. Part of the heat generated in the HS pockets transfers from the hydraulic fluid to the guide surfaces, another part transfers to the structure on the way to the tank and the rest is transferred by the fluid to the hydraulic unit, from which the waste heat must be removed by the cooling system.
In order to design a temperature-stable machine tool, it is necessary to achieve a state of equilibrium between the heat generated in the hydrostatic lines and the heat that transfers to the environment through the MT structure and the oil. CFD may be used for temperature analysis, but as follows from the literature review, its use for the analysis of groups of MT or entire machine tools would demand extensive computing power, and thus it is not used in practice. For this reason, the following simplification of the heat transfer mechanism between the hydraulic medium and the machine tool structure was proposed, which relies only on the FEM solver and is schematically shown in Fig. 6.
Assume that the flow of hydraulic medium Q at temperature T_0 enters the hydrostatic cell, where energy dissipation and pressure losses occur. The total heat loss flow ϕ_g is introduced

directly into the HS cell guide surfaces. Then heat transfers from the HS cell guide surfaces to the hydraulic medium. Let's assume that the temperatures of the guide surfaces and the hydraulic medium equalize at the temperature T_P. The heat flow can be expressed as:

$$\phi_s = c_p \rho Q (T_P - T_0) \qquad (28)$$

The density of the heat flux transferring from the HS cell guide surfaces to the hydraulic medium can be expressed as:

$$q_s = \frac{c_p \rho Q (T_P - T_0)}{S_t} \qquad (29)$$

where S_t denotes the total area of the hydrostatic cell. The flow of the hydraulic medium continues from the hydrostatic cell to the tank through the machine tool structure. Provided that the temperatures of the hydraulic medium and the wetted surface of the structure equalize at the temperature T_B due to the heat transfer between the medium and the wetted surfaces, it is then possible to express the heat flow as:

$$\phi_p = c_p \rho Q (T_P - T_B) \qquad (30)$$

The density of the heat flux passing between the heat exchange surface at the oil outflow of the structure and the hydraulic medium can be expressed as:

$$q_p = \frac{c_p \rho \dot{Q} (T_P - T_B)}{S_o} \qquad (31)$$

where S_o denotes the heat exchange surface at the oil outflow of the hydraulic medium through the structure. From the outlet port in the structure, the hydraulic medium flows through the lines to the hydraulic unit and to the cooler. In order to guarantee the temperature stability of the machine, the guide surfaces of the hydrostatic cell must not exceed a certain temperature (in practice, the temperature $T_{Pmax}= 50°C$ is often used, however this might be related to rapid degradation of hydraulic oil at higher temperatures) and that the oil in the cooling system is cooled to the initial temperature. The heat flow that must be removed from the hydraulic medium in the cooling system can be expressed as:

$$\phi_0 = c_p \rho Q (T_B - T_0) \qquad (32)$$

In order to include the heat flux density, expressed in equations (29) and (31), in the FEM task without complicated adjustments, the similarity with the relation for convective heat transfer was used. The heat flux density due to convection is given by the relation:

$$q_{konv} = h (T_s - T_r) \qquad (33)$$

where h denotes the heat transfer coefficient, T_s denotes the surface temperature and T_r the ambient temperature. Comparing relation (33) with relations (29) and (31) shows that we can formally consider the heat transfer coefficients on the surface of the HS cell as:

$$h_s = \frac{c_p \rho Q}{S_t} \qquad (34)$$

and T_r=T_0. The heat transfer coefficients on the heat exchange surface of the outflow of the hydraulic medium through the machine tool structure can be expressed as:

$$h_p = \frac{c_p \rho Q}{S_o} \qquad (35)$$

where T_r=T_B. The scheme of the thermal boundary conditions set up is shown in Fig. 7.

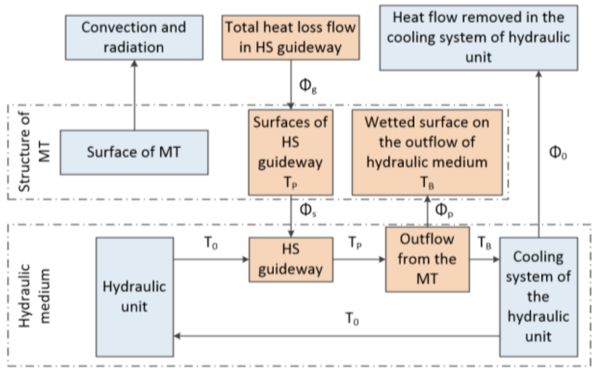

Figure 6. The proposed heat transfer mechanism between the hydraulic medium and the structure of the machine tool with hydrostatic guideways

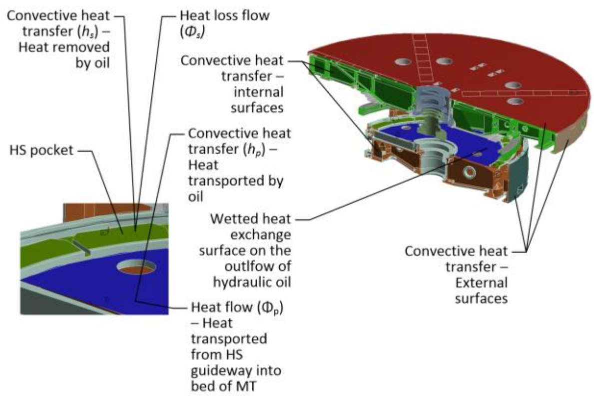

Figure 7. Diagram of heat transfer BC on a turntable as an example

The presented thermal model uses the proposed simplified heat transfer mechanism between the hydraulic medium and the machine tool structure to solve the given heat balance using FEM as a non-stationary problem in the given time interval and to provide the temperature field for structural analysis.

During the operation of hydrostatic guideways, the temperature fields that are generated contribute to the thermal deformations of the structure, which manifest as tilting and displacement of the guiding surfaces. It follows from the measurements of the table used as an example that the tilting of the guiding surfaces caused by temperature deformations can exceed the tilting of the guiding surfaces caused by force effects, such as gravitational forces, inertial effects or cutting forces. The preparation and solution of the thermal model are therefore important parts of the analysis of a machine with high-speed hydrostatic guideways.

2.4 Structure of the thermo-mechanical simulation

The goal of solving a thermo-mechanical problem is to simulate the operation of a machine tool with hydrostatic guideways. In the thermo-mechanical simulation, the discretization of the spectrum of movement speeds of the machine tool axes is used, which, after each time step of the temperature task and the subsequent structural calculation, allows for the updating of the parameters of the HS guideways and the thermal boundary conditions according to the current state of the structure and the heat sources in the HS guideways according to the operating conditions in the next step.

The complex simulation of a machine tool with hydrostatic guides provides three types of results: the temperature field, which is solved by the finite element method with known heat sources in HS guideways; the deformation field, which is solved by the finite element method after loading the temperature field and the characteristics of the hydrostatic cells; and the thermo-mechanical state of each HS cell, which is described analytically.

In chapter 2.1, a model describing the non-linear force deformation behaviour of the hydrostatic cell coupled with the flexible structure of the machine tool was presented. In the first part, matrices of nonlinear force deformation characteristics of individual hydrostatic cells were generated using the Matlab environment, which were then imported into the detailed FEM model of the MT as inputs of COMBIN39 elements, and this model was then used to submodel each HS cell. The effect of tilting the HS cell's guide surfaces on its hydraulic resistance and ultimately on its force deformation characteristics was modelled according to [Stach 2016]. The hydraulic resistance and characteristics of the hydrostatic pockets were updated between each time step of the analysis based on the deformation field from the previous step and the tilt evaluation from the displacement of the guide surfaces nodes. In chapter 2.2 and 2.3, the heat sources in the hydrostatic guideways and the simplified mechanism of heat loss transfer between the hydraulic medium and the structure of the machine tool were described. Temperature boundary conditions and heat sources are recalculated between each time step in the Matlab environment based on the current HS throttling gap heights, flow rates and MT operating conditions (mainly the speed of movement of each machine axis).

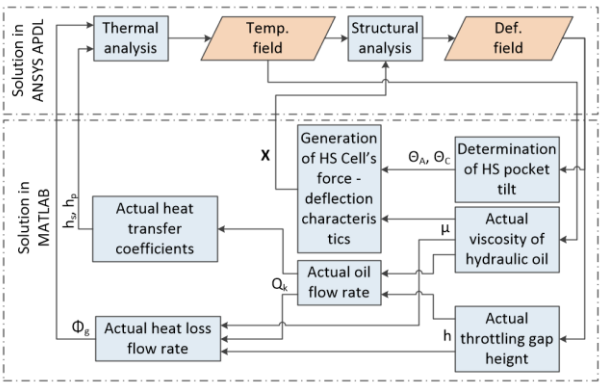

Figure 8. Structure of thermo-mechanical simulation

Data transfer between the computational part in the FEM solver and the Matlab environment does not take place directly. As can be seen from Fig. 8, the thermal and structural tasks are first performed in the FEM solver within one time step, then matrices are exported in ASCII text format and loaded after the script is run in the Matlab environment. After updating the HS cell characteristics, heat sources and temperature boundary conditions are written into matrices and exported in text format. Before starting the FEM calculation of the next time step, the new parameters are imported by the FEM solver. The whole process, controlled by Matlab script, runs automatically.

3. Experimental verification

In order to apply the described complex thermo-mechanical simulation of the behaviour of a structure with hydrostatic guideways on a real structure, a complex simulation model was built using the example of a table of a vertical turning lathe (VTL). In this case, the rotary table is a separate part of the machine tool structure, for which the solution of the heat balance of hydrostatic guideways is a current issue, and at the same time, the structure of the rotary table is too complex to be able to effectively use the calculation procedures described in the literature. For these reasons, a fully hydrostatic rotary table for workpieces of up to 4.7 m in diameter and of up to 100 t was chosen for the application of the developed complex simulation.

In order to experimentally verify the proposed simulation, the worktable was equipped with temperature and Eddy current sensors monitoring the axial HS guiding surface and the throttling gap height. Pressure and oil flow sensors were added to measure the HS hydraulic system properties.

The measurement and simulation followed the same discretized spectrum of table rotational speeds with no load (e.g. workpiece weight, cutting and clamping forces).

4. Results and discussion

A comparison of the results of the calculation and measurement of the development of the temperature in the axial hydrostatic guideway is shown in Fig. 9. The calculation results demonstrate a higher temperature gradient at the start of the rotation and then stabilization at almost 2°C lower than the measured results. At the end of the sequence the calculation results once again demonstrate a higher decreasing temperature gradient at the stop of the table rotation.

The difference in temperature gradients could be caused by the simulation procedure. The calculation of the heat loss fluxes and heat flow coefficients is based on the oil flow of the last time step while the heat sources are calculated based on the revolution speed of the next step. When the revolution speed increases, the model operates with high heat sources but with low oil flow, resulting in higher heat flux into the structure. In contrast, when the revolution speed decreases, the model operates with lower heat sources and with higher oil flow, resulting in higher heat flow outside the machine tool structure with the oil. Another effect that could affect the lower measured temperature gradient is the location of the temperature sensor, which was placed under the sliding plastic layer of the guiding surface and not directly in the HS pocket, resulting in lower sensitivity.

A comparison of the results of the calculation and measurement of the development of the throttling gap height of the axial hydrostatic guideway is shown in Fig. 10. The throttling gap height at the centre of the HS pocket stabilizes after the start of the HS system at a height of 80 µm where it is set through the whole sequence with only a slight deviation. In the second half of the sequence, the measured value starts to pick up a drift towards a height of 90 µm. The drift is in correlation with the change in temperature and may be caused by a deviation in sensor sensitivity, which has a non-linear temperature dependence. The measured values also demonstrate the difference between the throttling gap height on the outer and inner sides of the axial guideway, resulting in tilt of up to 50% of the central throttling gap, which is caused by thermal deformations. During ordinary operation the tilt would be evened out by the weight of the workpiece and the clamping forces to some extent.

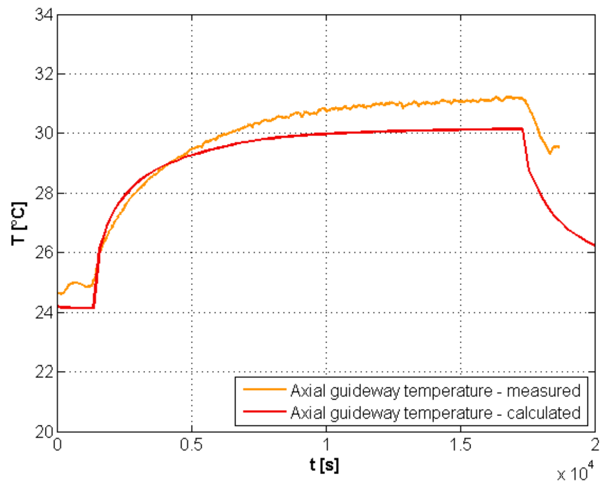

Figure 9. Comparison of the results of calculation and measurement of the development of the temperature of the axial hydrostatic guideway

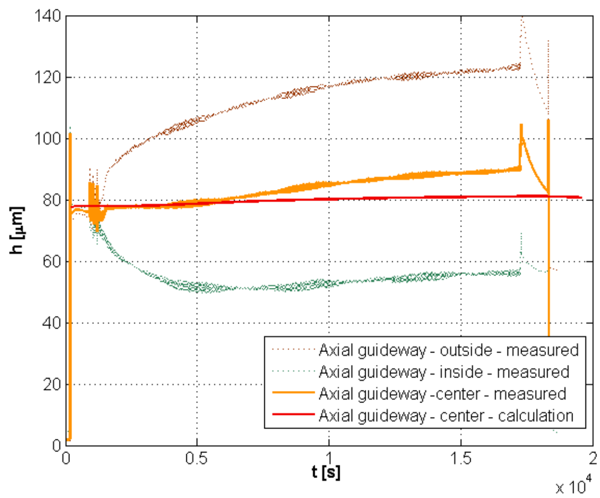

Figure 10. Comparison of the results of calculation and measurement of the development of the throttling gap height of the axial hydrostatic guideway

5. Conclusion

New materials and the ever-increasing demand for higher precision create new demands on machine tools. New concepts require much higher temperature and dimensional stability as well as high reliability and a long lifespan. Bearings and guides are a decisive factor, as they affect the machine tool parameters. One option is application of hydrostatic (HS) bearings and guides, which provide high damping, stiffness, reliability and durability. However, their design is more challenging for machine tool manufacturers.

Currently, hydrostatic guideways are designed using both conservative and unconventional approaches. Conservative approaches consider the hydrostatic guides as an isolated part of the MT structure, while on the other hand, unconventional approaches allow modelling of the interaction between the HS guideways and the MT structure either in greatly simplified form (solutions using FEM) or very detailed form that extends to only a small domain that does not cover the whole MT structure due to the computational complexity (a connected solution using CFD and FEM). The aim of this paper was to build a complex transient simulation allowing accurate representation of the operation of hydrostatic guideways within the entire machine tool structure. This goal was fulfilled by a coupled solution of the temperature and structural analysis using FEM with an analytical description of the thermo-mechanical state of HS cells.

In the proposed transient thermo-mechanical simulation, the discretization of the spectrum of movement speeds of machine tool groups is used, which, after each time step of the thermal analysis and the subsequent structural calculation, enables the HS cell parameters and thermal boundary conditions to be updated according to the current state of the structure and the heat source in the HS guideway according to operating conditions in the next step.

A complex simulation of a machine tool with hydrostatic guideways provides three types of results in relation to one another: the temperature field, which is solved by the finite element method for the described heat sources in HS guideways; the deformation field, which is solved by the finite element method after loading the temperature field and the characteristics of the hydrostatic cells; and the thermo-mechanical state of each HS cell, which is described analytically in the Matlab environment and allows heat sources and force-deformation characteristics of hydrostatic guideways to be solved and temperature boundary conditions to be updated.

During the operation of hydrostatic guideways, the generated temperature fields contribute to the thermal deformations of the structure, which manifest themselves as tilting and displacement of the guiding surfaces. The preparation and solution of the thermal model are therefore important parts of the complex transient model of a machine with high-speed hydrostatic guideways.

The thermal model uses a simplified mechanism of heat transfer between the hydraulic medium and the machine tool structure to solve the specified heat balance using FEM as a non-stationary problem at a specified time interval and to provide a temperature field for structural analysis.

The coupling of the detailed FEM model of the flexible structure of the machine tool with the model of the nonlinear force-deformation characteristics and the analytical description of the influence of the tilting of the HS pocket enables analysis of the MT hydrostatic guideways even in cases with more significant tilting of the guiding surfaces.

The proposed complex transient analysis procedure was applied and verified by measurements on a medium-sized table of a vertical turning lathe. For rotary tables, especially for turning machines, the maximum rotation speed is very important from the point of view of machining productivity. Its increase is often limited by the temperature stability of the MT structure. For the application of modern high-speed HS guideways, the described procedure of complex transient analysis is a helpful tool for verifying the thermal stability of the entire machine, but also for the appropriate dimensioning of the hydrostatic system and hydraulic unit, which is often very demanding in terms of energy.

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CONTACTS:

Ing. Eduard Stach, Ph.D.
Czech Technical University in Prague, Faculty of Mechanical Engineering, Department of Production Machines and Equipment (RCMT)
Horska 3, Prague 2, 128 00, Czech Republic
+420 221 990 969, e.stach@rcmt.cvut.cz,
https://rcmt.cvut.cz/

Ing. Jan Smolik, Ph.D.
Czech Technical University in Prague, Faculty of Mechanical Engineering, Department of Production Machines and Equipment (RCMT)
Horska 3, Prague 2, 128 00, Czech Republic
+420 221 990 918, j.smolik@rcmt.cvut.cz,
https://rcmt.cvut.cz/

Ing. Matej Sulitka, Ph.D.
Czech Technical University in Prague, Faculty of Mechanical Engineering, Department of Production Machines and Equipment (RCMT)
Horska 3, Prague 2, 128 00, Czech Republic
+420 221 990 944, m.sulitka@rcmt.cvut.cz,
https://rcmt.cvut.cz/

Ing. Tomas Lazak
Czech Technical University in Prague, Faculty of Mechanical Engineering, Department of Production Machines and Equipment (RCMT)
Horska 3, Prague 2, 128 00, Czech Republic
+420 221 990 968, t.lazak@rcmt.cvut.cz,
https://rcmt.cvut.cz/

Ing. Ivan Divis
Czech Technical University in Prague, Faculty of Mechanical Engineering, Department of Production Machines and Equipment (RCMT)
Horska 3, Prague 2, 128 00, Czech Republic
+420 221 990 941, i.divis@rcmt.cvut.cz,
https://rcmt.cvut.cz/

Ing. Jiri Falta, Ph.D.
Czech Technical University in Prague, Faculty of Mechanical Engineering, Department of Production Machines and Equipment (RCMT)
Horska 3, Prague 2, 128 00, Czech Republic
+420 221 990 966, i.divis@rcmt.cvut.cz,
https://rcmt.cvut.cz/