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Model based prediction approach for internal machine tool heat sources on the level of subsystems

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Abstract

Modern machine tools are highly sophisticated mechatronic systems. Each subsystem's energy efficiency is important regarding thermal effects of the machine: Losses in the subsystems are mainly heat sources, causing temperature gradients and thermal elongation. Knowledge about the internal heat sources is therefore mandatory for high precision machining, as well as for the design of compensation strategies. This paper presents a modeling approach to estimate the heat release of machine tool subsystems and predict boundary conditions for thermal models. The simulation results are verified by measurements on an internal cooling system of a lathe.

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Keywords: Machine tool; Energetic models; Spindle cooling; Heat source prediction; Temperature measurement

1. Introduction

1.1. Motivation

Precision and process stability are key success factors in manufacturing industry. The challenge thereby is not only to achieve a certain machining quality, but also to maintain these properties under various disturbances. A major source for disturbances are thermal effects, causing up to 80% of all geometric errors on workpieces [1] . The formation of these errors can be summed up as follows: Driven by different heat sources, temperature gradients are formed, leading to thermal elongations and therefore to displacements at the tool center point (TCP) [2, 3] . For the sake of simplicity, for heat sources as for heat sinks the term heat source is used, since heat sinks are negative signed heat sources. Heat sources can have different underlying effects and causations. A general categorization is given with internal and external heat sources. While external heat sources represent the interaction with the environment, internal heat sources have their origin within the machine tool housing. Examples are ohmic losses, friction or decompression of process gas. Since a machine tool is an assembly of different components with certain inefficiencies, each of its components is a potential internal heat source. The machine tool components used depend on the type of the machine. Examples are main spindle drives, bearings, servo motors, lubricant pumps, compressed air consumers, control devices and many others. This list is neither mandatory nor complete for the big diversity of different types and sizes of machine tools existing. More important is that each machine has components to provide the process energy and axis movements, but also auxiliary components like temperature control and chip

c_p	specific heat capacity	[J/(kg·K)]
m	mass	[kg]
Q	thermal energy	[J]
P	electric power	[W]
p	pressure	[Pa]
T	torque	[Nm]
α	heat transfer coefficient (HTC)	[W/(K·m^2)]
β	coefficient of performance (COP)	[-]
η	efficiency	[-]
ρ	density	[kg/m^3]
ϑ	temperature	[K]
ω	rotational speed	[rad/s]

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removal in order to guarantee the required process and machine conditions. Investigations show that the auxiliary components have a substantial contribution to the total power demands of modern machine tools [4-6] .

Thus, heat loss of each machine tool component influences its precision, the process ability and process stability. If the influences can be predicted, two actions become accessible:

- First, the design of better machine concepts with respect to thermal effects, and
- Second, active compensation of thermal effects by prediction of heat losses.

In this paper an approach is described predicting heat sources as sum of the energy consumed as well as its interaction with the environment. Using basic conservation laws and the laws of physics, heat sources could be quantified in dependency of the consumed electricity and resources, as well as its interaction with the environment. The computational results using this approach are compared to measurements on the level of machine tool components. The approach therefore develops general energetic models of machine tool components.

1.2. State of Research

The demand of an energy efficient production is the driver for the development of energetic machine tool models. Since machine tools are known to consume significantly more energy during their use-phase than containing gray energy [7] , the operational phase has to be optimized. In order to face this challenge, different approaches are available. An overview of the state of art in energy flow modelling of manufacturing systems is given by Thiede et.al. [8] . The authors present further a holistic system definition, including single machines in a factory building. Li et.al. [9] present an empirical approach to predict the power consumption of a single machine in dependency of the process parameters. Using measurements on a lathe, the authors identify the necessary empirical parameters. A different approach is chosen by Braun et.al. [10] and Gontarz et.al [11] . They subdivide the machine tool into its components. The energetic behavior of each component is computed using component models based on physical laws. Gontarz et.al. further developed a simulation platform, including a component library, which allows computing the energetic behavior of machine tool concepts and the involved components in early development stages. Using physical laws the approach of Gontarz et.al. is able to compute, besides the power consumption on component level, heat losses.

For computing thermally induced TCP-displacements several models are developed. An overview is given in [1] . Ess [3] developed a computation routine based on Finite Element Method (FEM) for efficient modelling and computation of machine tool typical load cases. In [12] a computation approach Finite Differences Element Method (FDEM) is described for computing thermally induced TCP-errors efficient. In [3, 12] further methods for model order reduction are described reducing the output to the relevant TCP-displacements. Gebhardt et al. [13] developed thermal error reduction models based on differential equations and phenomenological system modelling approaches for five-axis machine tools. The models are able to compute axis correction movements for reducing thermal errors in real time.

Most equipment for energy measurements in manufacturing is measuring effective electrical power $P_{eff}(t)$. There are various commercially available measurement systems for power measurements as indicated by Kordonowy [14] . In research Behrendt et al. [15] and Avram et al. [16] use conventional single channel 3-phase metering systems in their measurement activities. For the analysis of all relevant subsystems a multichannel approach with synchronized multimeters are needed as used by [3] and [17] .

1.3. Research Gap

As shown in the literature review, the simulation and prediction of thermal deformations and TCP errors are known processes. Up to know the model approaches focus on the spindle and main drives. Auxiliary units, such as pumps or electronics are not included. In energetic modelling physical and modular approaches to compute the energy demand of single machine tool components have successfully been shown. The gap in modelling between energetic models and thermal models computing the TCP errors have not been investigated until now. In this work an approach is shown that uses the outputs of an energetic model for machine tool components as input parameters to compute the interaction of the components with its environment. The results can easily be implemented in existing thermal machine tool models and be used as boundary conditions for computing TCP errors.

2. Model Developing Procedure

To quantify the capability of energetic models in thermal simulations, as well as the identification of research gaps for a complete union of the two model domains, a model developing procedure is suggested in the following section. The procedure must assure the following outcomes:

- Prediction of heat losses by energetic models
- Determination of induced transient temperature distribution by extended energetic models
- Verification of the simulation results by measurements
- Identification of research demand based on the results gained during the verification and system modeling

The test system selected to apply the procedure is a lathe with a 15 kW. This particular machine tool is used, due to having an internal cooling system for the spindle that is not influenced by other components. The cooling system consists of a circulation pump and a compressor. Since this system configuration includes essential components to fulfill the machine main task, as well as auxiliary devices it is a representative part of a machine tool. The target boundary condition to be modelled is the coolant entry temperature at the spindle. To address the required outcomes of the procedure, a three step approach is chosen:

- First the system is modelled using extended energetic models
- Afterwards the system behavior is measured in the second step
- During the third step, the computation results are compared to the measurement data.

Based on the identified deviations, the capability of the postulated approach will be quantified and the required research will be identified.

3. Test Bench and Data Acquisition

For parameter identification and for model verification measurement data is required. Lathe under investigation is a *Schaublin 42L*. The main spindle of the machine tool is temperature controlled, while the coolant reservoir of the spindle cooling device is temperature controlled by a two level controller. Thus, the inlet temperature of the cooling fluid is fluctuating between 26.5°C and 31°C. The circulation pump is constantly conveying a constant coolant flow from the reservoir to the spindle, while compressor is switched on and off by the control of the reservoir temperature. Since the cooling device uses the ambient air for re-cooling, a fan is constantly operating, providing the required air flow. According to the data sheets, the power demand of the compressor is 1000 W, while the circulation pump requires 135 W. For model evaluation the relevant energetic energy flows, as well as the thermal state of the system over time are measured. The thermal state is the inlet temperature of the coolant in the reservoir. It is assumed to be equal to the inlet temperature.

3.1. Measurement set-up

In order to acquire all relevant data, the power input to the spindle amplifier, as well as the power to the cooling device are measured by synchronized multimeters as described by Gontarz et al. [18] . The ambient temperatures and the cooling fluid reservoir temperature are retrieved by thermocouples.

3.2. Data Acquisition

For model evaluation two operational stages are measured. During the first stage the machine tool is operating face turning of a 51CrV4 blank of 40 mm diameter over a length of 40 mm with cutting speed 100 m/min. In this stage, two internal heat sources have to be considered: Heat loss from the spindle and from the cooling device itself. In the second measurement stage the machine is not operating and the spindle is stopped. The cooling system is kept fully operational, so that heat loss from the cooling device itself has to be considered.

4. System Modeling

Fig. 1 shows the schematic of the test bench system and its boundary. The inputs to the system are required torque and rotational speed of the spindle. The outputs are demanded electrical power and heat flux from the cooler. The internal state of the model is the cooler temperature ϑ_t. The model describes the relationship between the input, output and ϑ_t. Since the process is defined by cutting speed and depth during the measurements, a conversion into spindle speed and torque is required. Hence, the cutting model of Kienzle et.al. [19] is used with the Kienzle parameters $k_{c1.1}$=1768 N/mm^2 and m_c=0.31. The values are identified with the material and tools used.

4.1. Spindle

The spindle transforms electrical energy to mechanical. Given the provided torque T and speed ω as operational point, the power demand can easily be predicted using the spindle's efficiency map. The efficiency map indicates the ratio between the mechanical power and the electrical power, and depends on the operational point. Following the sum of energy, the losses $\dot{Q}_s$ generated in the spindle are the difference between the electrical power consumed and the resulting mechanical power P_s, such that

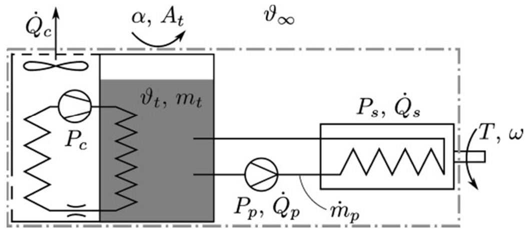

Fig. 1. Schematic of the system with all energy consumptions P, heat sources $\dot{Q}$ and mass flows $\dot{m}$ of the spindle (index s), the pump (index p) and the compressor (index c). The coolant tank is characterized by its temperature ϑ_t, mass m_t and surface A_t. The gray dot-dash line indicates the system boundary with the inputs torque T, rotational speed ω, ambient temperature ϑ_∞ and HTC α.

$$P_s = \frac{T \cdot \omega}{\eta(T,\omega)} \quad \text{and} \quad \dot{Q}_s = \left(\frac{1}{\eta(T,\omega)} - 1\right) \cdot T \cdot \omega . \tag{1}$$

This model implies that all losses are thermal losses and that a description of the efficiency η in dependency of the provided torque and rotational speed does exist.

4.2. Cooling System

Modeling the cooling system requires three steps:

- First the characterization of the circulation pump,
- Second the description of the compressor behavior and
- Third the modeling of the reservoir temperature dynamics.

Pump systems are characterized by a map, describing the relationship between mass flow $\dot{m}_p$, pressure p_p, and power demand P_p. Knowing this relationship, the power demand can be expressed as function of the mass flow, while the power losses $\dot{Q}_p$ are estimated as the difference between power supplied and fluid-dynamic power received:

$$\dot{Q}_p = P_p(\dot{m}_p) - \frac{\dot{m}_p}{\rho} \cdot p_p(\dot{m}_p) \tag{2}$$

For characterizing the compressor, the coefficient of performance (COP) is used. The heat flux of the compressor consists out of two parts:

- First the heat losses due to inefficiencies and
- second a heat flux induced by the working gas for the cooling cycle.

With the assumption, that all power P_c supplied to the compressor is transferred into heat, $\dot{Q}_c$ results in the following description:

$$\dot{Q}_c = (1 - \beta_c) \cdot P_c \tag{3}$$

Since for the COP β_c of a heat pump $\beta_c > 1$, the heat flux from the compressor to the system will be negative, e.g. heat energy is extracted from the system.
In order to describe the temperature dynamics of the reservoir temperature, the first law of thermodynamics is used [20] :

$$m_t \cdot c_p \cdot \frac{d\vartheta_t}{dt} = \dot{Q}_s + \dot{Q}_c + u \cdot \dot{Q}_p - \alpha \cdot A_t \cdot (\vartheta_t - \vartheta_\infty) \tag{4}$$

On the right hand side of (4), all heat losses of the components, as described above, are included. The constant parameter u indicates the fraction of pump losses going into the fluid and thereto into the reservoir. Heat exchange with the environment of the reservoir through convection is characterized by the additional term on the right hand side in (4). The cooling fluid flow in and out of the reservoir causes swirls. Therefore a uniform temperature distribution in the reservoir can be assumed.

4.3. Parameterization

Except of the heat transfer coefficient α and the pump loss distribution u all parameters used in (1)-(4) are given in datasheets [21, 22] . Using the measurement of the temperature development during the machine heat up, u can be estimated as 35% by a least squares approach. If the machine tool is powered up and no process is active, the reservoir temperature is only influenced by the pump losses, hydraulic friction, the convection at outer tank surface and the compressor heat flux to the ambient air. This leads to the periodical heat up as shown later in Fig. 3. Given the temperature range of operation as [ϑ_{low}, ϑ_{high}] and the time between two cooling cycles as t_s, a simple relationship can be derived based on (4):

$$u \cdot \dot{Q}_p = \alpha \cdot A \cdot \left[\vartheta_{high} - \vartheta_\infty + \frac{\vartheta_{high} - \vartheta_{low}}{\exp\left(- \frac{\alpha \cdot A}{m_t \cdot c_p} \cdot t_s \right) - 1} \right] \tag{5}$$

Solving (5) numerically for u=0.35 value of α can be identified. Table 1 shows a list of all used parameters. Fig. 2 shows the efficiency map of the spindle. Since this work intends to compare simulation results with measurements, the initial condition for the models state – the reservoir temperature – is extracted from the measurement data as well.

5. Results

Time series of the measurement and simulation are shown in Fig. 3. The first 120 minutes are the warm-up phase before processing. Afterwards machining is performed for further 155 minutes. During the process, the compressor is switched on in short intervals of about 10 minutes, causing a high frequent reservoir temperature fluctuation varying between 26.5°C and 31°C. Afterwards the machine tool is operating without cutting. Oscillations of the same amplitude but with a period time of about 210 minutes take place. Since the circulation pump is the only heat source within the selected system in the relevant timeframe, pump losses and hydraulic friction raise the reservoir temperature. The same heat sources are the cause for temperature raise in the reservoir in the first 120 minutes in Fig. 3.

Verification of the model is achieved by comparing the simulation results to the measurement results discussed above. This comparison is also shown in Fig. 3. The results show that the cooling fluid in the reservoir can be computed with the model. Even in the transient behavior some differences occur. This effect can be seen in Fig. 3 around time=400 minutes.

6. Discussion and Outlook

Within this work, the prediction of boundary conditions for thermal machine tool models by an energetic approach has been discussed on an exemplary case of coolant temperature. The presented model has been kept as simple as possible, while enabling a quantitative and qualitative prediction of the studied boundary condition. The simplicity of the model allows the integration into existing complex thermal models for real-time TCP error compensation without enhancing the computational effort substantially. The model is based on datasheet values, except two machine tool specific parameters, the heat losses of the pump and the heat transfer coefficient to the environment. These parameters have to be determined experimentally. To dissolve this dependency on machine specific measurements, two approaches are possible:

- First, a refinement of the presented model, by more detailed modelling of components and their interactions, or
- Second, creation of a macro-element database enabling the prediction of the desired parameters based on empirical data.

Since with the first approach the simplicity of the model is lost – and there to the possibility of real-time compensation – the second approach is enforced. In order to realize this approach and enhance the

Table 1. Summary of the parameters used in the model.

Parameter		Value	
Compressor EER	β_c	3.14	W/W
Tank mass	m_t	13	kg
Specific heat capacity coolant	c_p	4’180	J/kg/K
Coolant density	ρ	1’000	kg/m^3
Exposed tank surface	A_t	0.25	m^2
HTC tank	α	15.5	W/m^2/K
Pump mass flow	$\dot{m}_p$	0.2	kg/s
Pump power	$P(\dot{m}_p)$	135	W
Pump pressure	$p(\dot{m}_p)$	70’000	Pa
Pump loss share	u	35	%

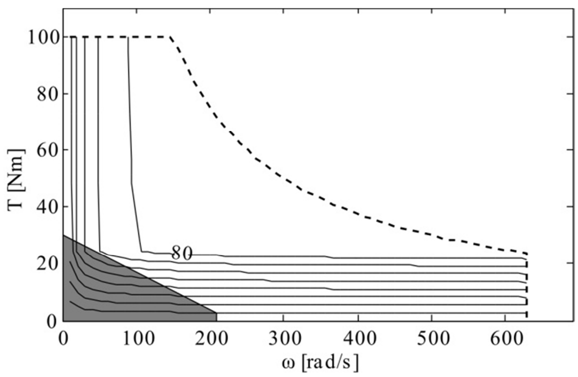

Fig. 2. Efficiency map of the main spindle. The gaps between single lines are 10%, while the 80% efficiency iso-line is indicated. The gray area indicates the operational points used during the measurement and the simulation.

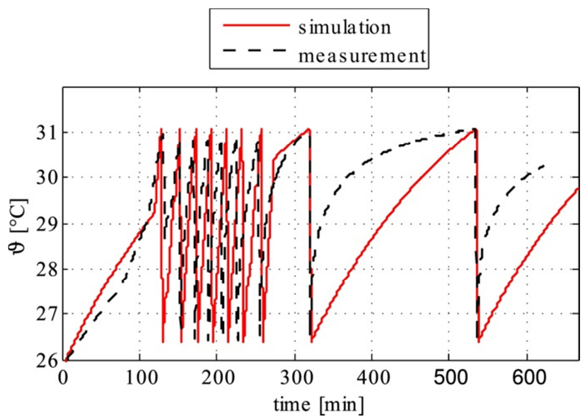

Fig. 3. Simulated and measured reservoir temperature ϑ over time. For simulation, the initial condition at time=0 is set according to the measured value.

capabilities of model based boundary condition prediction for thermal machine tools, additional research is required in the domains of:

- Sophisticated heat transfer coefficient prediction: Machine tools provide difficult conditions for a simple rule based setting of the HTC. While housings and complex geometries might disturb free convection, other effects such as sealing air could lead to forced convection.
- Generic description of heat release in machine tool components: Using a sum of energy and energetic models to predict the heat release of a component does not give any geometric information, e.g. where the heat sources are actually located. In this work, the overall pump losses have been multiplied with a factor representing the part of the losses going into the fluid. However, for each new system new measurements are required to identify the machine tool specific parameters. Developing a generic and configurable method to predict heat fluxes within the single components would overcome this limitation.

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