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A numerical approach for investigating thermal contact conductance

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ABSTRACT

Thermal resistances occurring at contacting interfaces have key importance for accurately predicting the thermal behavior of machine components. Although many efforts have been made to develop experimentally validated tools, which quantify contact heat transfer without needing to resort to experiments, this has not yet been satisfactorily achieved. The goal of this paper is to introduce a novel approach for predicting contact heat transfer, which involves micro-scale numerical simulation of the contact mechanics and the thermal behavior of three-dimensional bodies, modeled after topographically measured technical surfaces. As is shown, this approach has significant advantages in comparison to previous work on this subject. The simulation tool is described in detail and validated against experiments conducted by means of a transient measurement method involving high-speed infrared thermography. The results show good agreement between measurement and simulation.

1. Introduction

Thermal management and thermo-energetic design of technical applications has become a necessary tool for improving productivity, efficiency and precision of machining and manufacturing processes.

For instance, a necessary prerequisite for a precise machining process is a temperature field in steady-state and, thus, stable thermal expansion rates within the manufacturing machine, as well as the work piece. Increasing the productivity of a manufacturing process can, for instance, involve reducing the machine's warm up time, or reducing the actual production time, which leads to a higher thermal load to machine and work piece. Both effects result in non-homogeneous temperature fields and thus, differing thermal expansions within the machine. Thus, reducing the productivity of a machining process, while still maintaining a highly precise output, can result in conflicting goals.

In order to resolve this conflict, a thermo–energetic model is required, which allows prediction and, in turn, correction of the machine's thermo–elastic behavior, especially under transient manufacturing conditions. This approach is dependent on the underlying thermal model of the machine, including heat sources and boundary conditions being as precise as possible. A extensive overview and discussion of this subject matter is given by Ref. [1].

Within this field of research, the presented paper focuses on one particularly essential problem to the accuracy of the thermal model, which is the prediction of heat transfer at contacting interfaces. At these interfaces, thermal resistances occur due to surface roughness, which results in a significant discrepancy between the interface's nominal cross-sectional area and the actual surface area within contact (real surface area). This leads to constricted heat flows and a measurable temperature drop over the contact region, which is illustrated in Fig. 1 for a stationary heat conduction problem.

The constriction of heat flow can be quantified by means of the contact heat transfer coefficient h_c, which, analogous to convectional problems, is defined through the ratio between heat flux and the observed temperature drop across the contact region ΔT_c (Fig. 1). A_{nom} represents the nominal area of the unconstricted heat flow cross-section:

$$h_c = \frac{\Delta T_c}{\dot{Q} \big/ A_{nom}} \tag{1}$$

The contact heat transfer coefficient can be determined experimentally by measuring the temperature drop, as well as the temperature gradient of the unconstricted flow in each specimen. This is either done by a stationary experiment, which is described extensively by Ref. [2], or through a transient method described by

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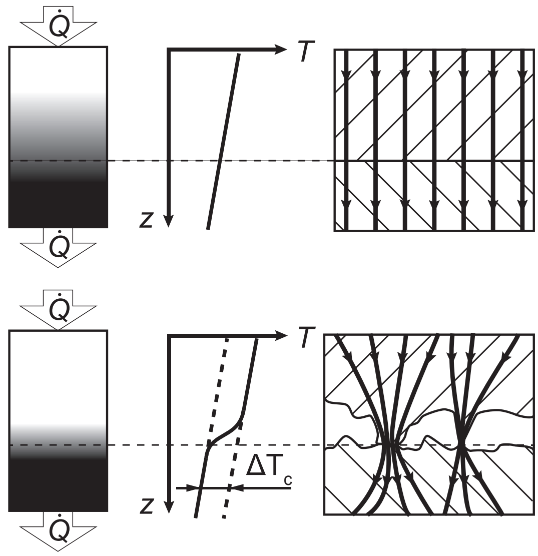

Fig. 1. Illustration of heat transfer and temperature distribution for a stationary case. Top: Conductive heat flow for a perfect contact (fully filled solid body). Bottom: conductive heat flow through a real contacting interface with significant surface roughnesses.

Refs. [3,4].

For an accurate thermo-energetic design, however, these coefficients need to be predicted for each contacting interface and implemented into a thermal model through transitional conditions, without relying on actual temperature measurements under laboratory conditions. Ref. [5] shows, how contact heat transfer coefficients impact the accuracy of thermal models for linear guidings, ball screws, ball bearings and mounting elements. Ref [6] discusses, how thermal contact resistance effects the temperature fields within the pultrusion process.

Some assumptions can be made, in order to simplify the heat transfer problem. Heat conductance is the dominant heat transfer mechanism for most contact heat transfer problems at atmospheric conditions. This is due to the low thermal conductance of gases, compared to solid materials, and no convection as a result of the small cavity sizes, i.e. small Rayleigh-numbers. Furthermore, if temperature ranges are significantly low, as typical for machine components, thermal radiation also is negligible [3,7]. Due to the minor influence of convection and radiation, the thermal modeling presented in this paper assumes contact heat transfer to be entirely due to conduction at the solid contacting spots (cf. Fig. 1, bottom). Based on these assumptions, the contact heat transfer coefficient is governed by the geometrical properties of the contacting surfaces and the thermal conductivity of the materials. The surface geometries are influenced by the mechanical properties of the contact (material hardness, contact pressure, elasticity).

This has been extensively covered by other research and has produced numerous analytical correlations, which calculate the contact heat transfer coefficient as a function of parameters, which are chosen to represent these contact properties as accurately as possible. A comprehensive overview of analytical work addressing contact heat transfer is given by Ref. [8]. Although the degree of complexity or the physical effects included in the model vary for these different correlations, the basic conceptual approach applied is mostly similar.

Most commonly cited correlations include Ref. [9–11], which thus, being representative of analytical work, shall be investigated further. These correlations use the surface contact pressure p and the micro–hardness of the material H as mechanical parameters, the harmonic heat conductivity of the joint material k as the thermal parameter, as well as the standard deviation of the surface heights σ and the average mean slope of the profile heights (m) as geometrical parameters. The correlation by Ref. [10], for instance, is given by:

$$h_c = 1.13 \cdot k \frac{m}{\sigma} \left(\frac{p}{p+H} \right)^{0.94} \tag{2}$$

Since the accurate parameterization of surface geometries is paramount for the quantification of the contact heat transfer coefficient, a comprehensive definition of the commonly used surface parameters is included in Appendix A.

The mentioned correlations employ simplifications regarding the geometrical structure, the mechanical deformation, as well as for solving the multidimensional temperature differential equation for the two deformed surfaces within contact. The advantage of this approach is, that it is comparatively easy to use for calculating contact heat transfer coefficients. However, as will be shown in the following chapter, certain deficiencies occur, due to improper simplifications, as well as to the main concept of using parametric roughness values to specify the thermal behavior of complex surface structures.

2. Deficiencies of analytical approaches

Analytical correlations make use of parametric values, which are aimed to characterize complex surface geometries. In addition, the surfaces are typically simplified to having an isotropic geometry and Gaussian height distribution. The latter simplifications reduce the number of variables, which characterize a surface, significantly. Most commonly, analytical approaches use two parameters, the mean slope m and the surface roughness σ, which are supposed to fully define a surface.

However, these simplifications are not permissible for the vast amount of different surface geometries, produced through different manufacturing processes. As an example for this, Fig. 2 shows an isotropic surface with Gaussian height distribution in comparison to a topographical measurement of a milled surface, in which predominant directions of the surface asperities can be observed. Thus, the analytical result to the contact heat transfer coefficient is prone to errors, especially for surface structures diverging from isotropy and Gaussian height distribution.

Moreover, the analytical models assume only one rough surface being in contact with a smooth and rigid plane, instead of independently modeling two surfaces with different parameter values and deformation behavior (see Fig. 3). This rough surface is deemed representative of the contacting instance by being a combination of the two single surfaces. The representative surface's roughness parameters are determined by the root-mean-squares of the surface parameters of the corresponding two surfaces in contact (Eq. (3)).

$$\begin{aligned} \sigma &= \sqrt{\sigma_1^2 + \sigma_2^2} \\ m &= \sqrt{m_1^2 + m_2^2} \end{aligned} \tag{3}$$

In order to obtain an analytical solution to the temperature-field over the contact area, Ref. [9–11] solve the stationary two-dimensional temperature-differential equation by employing so-called flux tubes, or adiabatic heat cylinders, which have shown to over-estimate the heat transfer significantly. This is due to neglecting the thermal resistance, which arises through the decrease in the cross-sectional area of the heat flow above and below the contacting spots of the interfaces, see Fig. 4 [12].