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Shrink fit tool holder connection stiffness/damping modeling for frequency response prediction in milling

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Abstract

In this paper we present a finite element modeling approach to determine the stiffness and damping behavior between the tool and holder in thermal shrink fit connections. The continuous contact stiffness/damping profile between the holder and portion of the tool inside the holder is approximated by defining coordinates along the interface contact length and assigning position-dependent stiffness and equivalent viscous damping values between the tool and holder. These values are incorporated into the third generation receptance coupling substructure analysis (RCSA) method, which is used to predict the tool point frequency response for milling applications. Once the holder and inserted tool section are connected using the finite element analysis-based stiffness and damping values, this subassembly is then rigidly coupled to the (measured) spindle–holder base and (modeled) tool. Experimental validation is provided.

Keywords: Receptance; High-speed machining; Finite element

1. Introduction

Discrete part production by milling remains an important manufacturing capability. However, there are many potential obstacles to producing quality parts at low cost in a timely manner. One particular limitation that has received considerable attention in the literature is chatter, or unstable machining; a second is surface location error, or an error in the part dimension caused by dynamic deflections of the tool (and potentially the part/fixture) during stable cutting. In both cases, a primary factor affecting the process performance is the system frequency response function, or FRF.

The system FRF, often dominated by the flexibility of the tool–holder–spindle assembly as reflected at the tool's free end, can be obtained using impact testing, where an instrumented hammer is used to excite the tool at its free end (i.e., the tool point) and the resulting vibration is measured using an appropriate transducer, typically a low-mass accelerometer. However, due to the large number of spindle, holder, and tool combinations that may be available in a particular production facility, the required testing time can be significant. Further, the measured response is often strongly dependent on the tool overhang length. Therefore, a model which is able to predict the tool point response based on minimum input data is the preferred alternative [1–13].

The purpose of this paper is to build on the previous work of Schmitz et al. [1–8], which describes tool point FRF, or receptance, prediction using the receptance coupling substructure analysis (RCSA) method. In these previous studies, two- and three-component models of the machine–spindle–holder–tool assembly were defined. In the two-component model, the machine–spindle–holder displacement-to-force receptance was recorded at the free end of the holder using impact testing, while the tool was modeled analytically. The tool and machine–spindle–holder substructure receptances were then coupled through translational and rotational springs and dampers, where their values were determined through a nonlinear least squares fitting procedure. In this initial work the displacement-to-moment, rotation-to-force, and

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rotation-to-moment receptances at the free end of the holder were assumed zero (i.e., perfectly rigid).

In the second-generation three-component model, the machine–spindle–holder substructure was separated into two parts: (1) the machine, spindle, holder taper, and holder flange (or spindle–holder base subassembly); and (2) the remaining portion of the holder from the flange to the free end (the extended holder subassembly). The rotation-to-force/moment and displacement-to-moment receptances for the free end of the spindle–holder base subassembly were determined using displacement-to-force measurements and finite-difference computations. The experimental procedure involved recording direct and cross displacement-to-force measurements of a simple geometry 'standard' holder clamped in the spindle and calculating the receptances at the free end of the assembly by finite differences [5,6]. The portion of the standard holder beyond the flange was then removed in simulation using an inverse receptance coupling approach to identify the four spindle–holder base subassembly receptances (i.e., displacement/rotation-to-force/moment). These receptances were then coupled to models of the actual holder and tool. Again, to account for finite stiffness and energy dissipation (i.e., damping) in the tool–holder connection, the tool was coupled to the holder using translational and rotational springs and dampers, assembled in the matrix K (Eq. (1)), where k_{yf} is the displacement-to-force stiffness, $k_{\theta m}$ the rotation-to-moment stiffness, c_{yf} and $c_{\theta m}$ are the corresponding viscous damping terms, and ω is the circular frequency (rad/s). See Fig. 1. The portion of the holder with the tool inserted was treated using a composite modulus and mass in the event that the holder and tool materials were different, such as a steel holder and carbide tool.

$$K = \begin{bmatrix} k_{yf} + \mathrm{i}\omega c_{yf} & 0 \\ 0 & k_{\theta m} + \mathrm{i}\omega c_{\theta m} \end{bmatrix}. \quad (1)$$

In this work, we extend the three-component model to include multiple connections between the tool and holder along the interference contact within the holder (rather than at the end of the holder as before). This is shown schematically in Fig. 2, where multiple complex stiffness matrices, K_i, describe the connection parameters at each location. We believe this to be a preferred solution because the stiffness/damping is now located at the appropriate locations, rather than artificially at the junction between the portions of the tool inside and

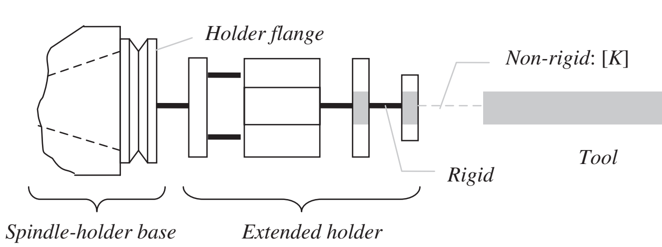

Fig. 1. Second generation RCSA model—the finite tool-holder stiffness/damping was represented by the empirical stiffness matrix, K, which was used to couple the overhung portion of the tool to the rest of the assembly. All other connections were rigid.

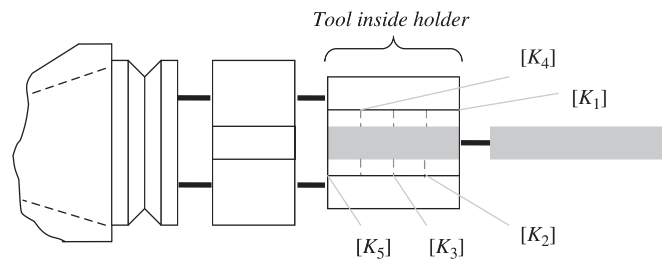

Fig. 2. Third generation RCSA model—the finite stiffness/damping between the tool and holder is represented by multiple K matrices determined from finite-element simulation.

outside the holder. We note, however, that the coordinate-based stiffness/damping analysis is an approximation of the continuous contact stiffness/damping profile between the holder and portion of the tool inside the holder.

In this new model the fully populated K matrix is defined as shown in Eq. (2), which now accounts for the displacement imposed by moment and the rotation caused by force through the nonzero off diagonal terms. Finite-element models are developed to determine the position-dependent stiffness and equivalent viscous damping values for a thermal shrink fit connection between the tool and holder, which represents the preferred interface for many high-speed milling applications. Using these values, the tool point FRF is predicted a priori and compared to measurements for a number of cases. No fitting parameters are applied in this analysis.

$$K = \begin{bmatrix} k_{yf} + \mathrm{i}\omega c_{yf} & k_{ym} + \mathrm{i}\omega c_{ym} \\ k_{\theta f} + \mathrm{i}\omega c_{yf} & k_{\theta m} + \mathrm{i}\omega c_{ym} \end{bmatrix}, \quad (2)$$

The paper is organized as follows. First, the RCSA approach for joining the portions of the tool and holder, which comprise the shrink fit connection is described. Second, the RCSA equation for the entire tool–holder–spindle assembly is provided. Third, finite-element modeling for the shrink fit connection is detailed. Fourth, experimental validation is provided. Finally, conclusions are presented.

2. Multiple point coupling for the tool–holder connection

To demonstrate the coupling between the concentric inner tool and outer holder components, the case of $n = 3$ connection coordinates, located at the ends of the contact length and at the mid-point, is now presented. The portions of the tool and holder in shrink fit contact are treated as free–free beams. For $n = 3$, a total of six component coordinates is obtained—three each on the internal tool and external holder (see Fig. 3) [14]. The component (i.e., tool and holder) displacement/rotations can be written as

$$\begin{aligned} u_1 &= R_{11}q_1 + R_{12}q_2 + R_{13}q_3, \quad u_2 = R_{21}q_1 + R_{22}q_2 + R_{23}q_3 \\ u_3 &= R_{31}q_1 + R_{32}q_2 + R_{33}q_3, \quad u_4 = R_{44}q_4 + R_{45}q_5 + R_{46}q_6, \\ u_5 &= R_{54}q_4 + R_{55}q_5 + R_{56}q_6, \quad u_6 = R_{64}q_4 + R_{65}q_5 + R_{66}q_6, \end{aligned} \quad (3)$$