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A method to predict position-dependent structural natural frequencies of machine tool

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ABSTRACT

Machine tool structure has a strong influence on the dynamic properties of the tool. The change of a machine tool's structure will cause variations in the dynamic parameters of the entire tool, such as its natural frequency, which will result in changes to the stability of the tool and poor machining quality. Thus, a study on the variations of machine tool dynamics is essential for high performance cutting. In this paper, using the mass change method, a basic mathematical model for predicting the natural frequency change resulting from structural change was presented followed by an experimental validation of the model. The mathematical model indicates that structural change will lead to the outward variation of the natural frequency, which is essentially related to the change of the squared mode shape values between the original position and the modified position of the moving component. With this natural frequency change rate prediction model, the natural frequency in the case of structural change can be easily predicted. The predicted results indicate that the positional change of different moving components has differing influences on the natural frequency of the machine tool.

1. Introduction

In the machining procedure, machining quality and productivity depend largely on the dynamics of the machine tool. Research investigating the dynamic properties of the tool center point (TCP) is especially important for the design and installation of the machine tool as well as the machining procedure. Considerable work has been undertaken concerning the dynamic properties of the whole machine tool system expressed by means of the frequency response function (FRF) at the tool center point. A common method is to predict chatter just from the dynamic properties of the spindle and tool system [1]. Many researchers have predicted the dynamic parameters of the spindle nose or the tool [2–4]. With the development of high performance cutting, machining the parts in the shortest time while respecting the physical constraints of the process such as torque, power, vibrations, tool wear and failure, surface quality, and tolerance, is demanded. Therefore, good dynamic behavior of the machine tool during high performance cutting is required. In general, the dynamic behavior of a machine tool can be greatly influenced by its structure, especially for heavy machine tools. Sulitka et al. [1] investigated the influence of the machine frame on the dynamic properties of the spindle and tool, and they demonstrated the importance of the machine tool structure on the dynamic properties evaluated at the tool tip. Kolar et al. [5] performed an experimental analysis of the machine frame impact on the dynamic properties evaluated at the spindle nose. The results show that the natural frequencies of the spindle nose were greatly influenced by the machine tool structure. Current studies have proven that the structure of a machine tool contributes significantly to the dynamic stiffness of the spindle nose; therefore, analysis of the dynamic properties of the entire machine tool structure is warranted.

As we know, a complete machine tool system is very complex, and some components, such as the headstock and worktable in most machine tools, do not stay in a fixed location during the machining process, and the dynamic properties always vary with the change in the machine tool structure [6]. Mousseigne et al. [7] studied the influence of different dynamic parameter shifts on the

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stability lobe curves and they found that the variation of the natural frequency has the greatest influence on the stability lobe curves, so that a precise estimation of the natural frequencies is well related to the certainty of the stability lobe curves. Because the position-dependent natural frequencies result in varying stability of machine tool [8], which has a great influence on the machine tool's productivity and the quality of the machined parts, it is necessary to predict the position-dependent natural frequencies of the machine tool.

The primary goals of the research in this paper are to propose a basic theory of natural frequency change rate prediction in the case of structural change, to validate this theory experimentally and to use it to predict the structural natural frequencies of a machine tool in the case of structural change. In the experimental validation process, the active excitation modal analysis (AEMA) [9] method was applied.

This paper is organized as follows: Section 2 proposes the basic mathematical models for calculating the natural frequency change rate both in the case of mass change and in the case of structural change. Then, both of these basic mathematical models are verified using simulation methods. Section 3 introduces the experimental verification of the mathematical model for calculating the natural frequency change rate in the case of structural change, and then the structural natural frequencies of a machine tool are predicted. In the experimental verification process, a newly proposed active excitation modal analysis method was applied. Finally, the work is summarized in Section 4.

2. Theoretical background

The basic theory of natural frequency change rate prediction in the case of structural change is based on the mass change method [10]. In this section, the basic theory of natural frequency change rate prediction in the case of mass change is proposed in Section 2.1. Based on the theory in Section 2.1, the basic theory of natural frequency change rate prediction in the case of structural change is presented in Section 2.2.

2.1. The basic theory of natural frequency change rate prediction in the case of mass change

2.1.1. The basic mathematical model of natural frequency change rate prediction in the case of mass change

In general, the structure of a whole machine tool system consists of fixed structure and moving components. In the case where the mass of the moving component is much less than that of the fixed structure, the dynamic stiffness of the moving component will be much larger than that of the fixed structure. Hence, the moving components can be regarded as rigid bodies, and they will vibrate with the vibration of the fixed structure. It can be thereby regarded that the moving component is attached onto the fixed structure [11], as is shown in Fig. 1.

In the case of no damping or proportional damping, the differential equation of motion of the fixed structure subjected to a force $\{F\}$ can be expressed as follows:

$$[M]\{\ddot{x}\} + C\{\dot{x}\} + K\{x\} = \{F\} \tag{1}$$

where $[M]$, $[C]$ and $[K]$ are the mass, damping and stiffness matrices of the fixed structure; $\{\ddot{x}\}$, $\{\dot{x}\}$, and $\{x\}$ are time functions organized in column vectors that characterize the evolution of the acceleration, velocity and displacement, respectively; and $\{F\}$ is a column force vector applied to the structure. The eigenvalue equation corresponding to Eq. (1) is given by

$$[M]\{\phi_0\}_r\omega_{0r}^2 = [K]\{\phi_0\}_r \tag{2}$$

where $\{\phi_0\}_r$ and ω_{0r} are the mode shape and the natural frequency of mode r of the fixed structure, respectively, and subscript '0' indicates the fixed structure.

When attaching a component on the fixed structure, such as installing a worktable or a headstock on a machine tool structure, there will be a variation in the mass distribution, as is shown in Fig. 1b.

As is shown above, the differential equation of motion of the modified structure can be expressed in the following form:

$$([M] + [\Delta m_i])\{\ddot{x}\} + C\{\dot{x}\} + K\{x\} = \{F\} \tag{3}$$

where $[\Delta m_i]$ is the mass matrix caused by the attached component, and

$$[\Delta m_i] = \begin{bmatrix} 0 & & & & \\ & \ddots & & & \\ & & m_i & & \\ & & & \ddots & \\ & & & & 0 \end{bmatrix},$$

in which m_i is equal to the m value, i.e., $m_i = m$. Subscript 'i' denotes the #i degree of freedom (DOF) of the fixed structure. Therefore, matrix $([M] + [\Delta m_i])$ reflects that the component of which the mass is m attaches to the fixed structure on the position of DOF #i. Similarly, the eigenvalue equation corresponding to Eq. (3) can be written in the following form:

$$([M] + [\Delta m_i])\{\phi_i\}_r\omega_{ir}^2 = [K]\{\phi_i\}_r \tag{4}$$

where $\{\phi_i\}_r$ and ω_{ir} are the mode shape and the natural frequency of mode r of the modified structure, respectively, when the component is attached to DOF #i. Subtracting Eqs. (2) and (4), the following expression can be obtained:

$$[M]\left(\{\phi_i\}_r\omega_{ir}^2 - \{\phi_0\}_r\omega_{0r}^2\right) + [\Delta m_i]\{\phi_i\}_r\omega_{ir}^2 = [K]\left(\{\phi_i\}_r - \{\phi_0\}_r\right) \tag{5}$$

Assuming that the mass of the component is so small when compared to that of the fixed structure that the mode shapes do not change significantly, i.e.,$\{\phi_0\}_r \approx \{\phi_i\}_r \approx \{\phi\}_r$, Eq. (5) can be written in the following form:

$$[M]\{\phi\}_r\omega_{ir}^2 + [\Delta m_i]\{\phi\}_r\omega_{ir}^2 = [M]\{\phi\}_r\omega_{0r}^2 \tag{6}$$

Multiplying Eq. (6) by the transport of the modal shape vector of

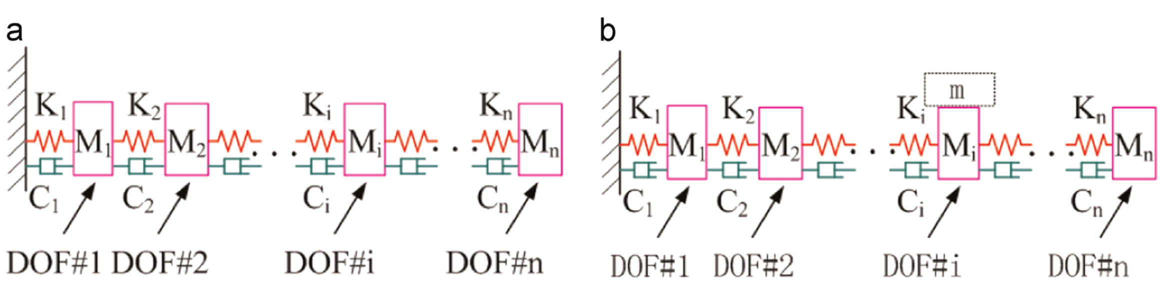

Fig. 1. (a) Fixed structure and (b) modified structure, with a component attached onto the fixed structure.