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Study for the Identification of Dominant Frequencies and Sensitive Structure on Machine Tools Using Modal Decoupling and Structural Sensitivity Analysis

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Abstract

Background The research on machine tool vibration has significant impact to improve the processing quality of the processed parts. The vibration response on the surface quality is quite different based on different modes of vibration.

Purpose Current study proposed a new approach of modes decoupling based on Operational Deflection Shape (ODS) and structural sensitivity analysis, which used to identify the structural vibrations on surface topography in manufacturing process. Method According to the modes decoupling based on the ODS method, the dominant vibration frequencies of machine tool are identified in a wide range of frequency band. Furthermore, the modal mass distribution matrix analysis is used to determine the sensitive structures that cause greater machining errors. A milling process is used to conduct experiments.

Results The experimental and theoretical results indicate that (i) the contribution of dominant vibration frequency to the vibration of machine tool is larger than weak modes vibration and (ii) the sensitive structure has higher vibration energy than the insensitive structures.

Conclusion The multimode vibration at dominant frequencies of the sensitive structures (MVDFSS) directly can determine the surface topography of the products.

Keywords Vibration response · Operational Deflection Shape · Mass distribution matrix · Sensitive structures

Introduction

The surface quality of a machining operation plays a significant role on the characterization of product performance. Moreover, it is also an important index to reflect the machining accuracy. In the literature, researchers have explored a series of factors which can influence the surface generation, including cutting conditions [1], material reduction swellability [2], material indentation crack [3], lattice orientation [4], tool position and its orientation [5], tool wear and vibration of machining tool [6, 7].

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The influences for the surface creation of workpieces relevant to the vibration from machine tool mainly focus on two aspects. The first aspect is the machine tool cutter with high-frequency vibration (usually reached upon 10,000 Hz), which may influence the surface manufacturing roughness. The second aspect is the machine tool structure with low-frequency vibration. Actually, several researchers have considered the surface creation of the machined parts which are affected by cutting tool vibration. For example, Wang et al. [8] studied the impact of the surface roughness from the high-frequency vibration of tool nose on the cutting force direction. Arizmendi et al. [9] established a prediction model to deal with the surface topography that occurred in the peripheral milling machining process, which considered the tool vibrations from cutting process. Cheung and Lee [10] designed a simulation system for the analysis of surface roughness generation that considered the impact of relative vibration between tool and workpiece. Zhang and To [11] studied the multimodal high-frequency vibration from the surface shaping effects in ultra-precision diamond turning (UPDT).

Structural vibrations can play a significant role in influencing the machining surface when the first-order natural frequency of the machine tool becomes three times greater than the designed operational frequency [12]. Chen et al. [13] proposed that the multimodal frequency vibration in machine tool plays a key role in surface generation. Abuthakeer et al. [14] studied the impact of spindle vibration on surface roughness through variance analysis. Zhang et al. [15] presented a calculation model to optimize the surface creation effects on spindle vibration. Zhang and To [16] found that the coupled-tilting spindle vibration is a predominant factor which can influence surface generation. Zhang and To [17] identified that the spindle vibration can produce rule-based or irregular patterns on the surface. In most of these studies, the tool is the given vibration. However, vibration can also be induced in the other structures of machine tools including worktable, bed, and vibration in slide that can also affect surface generation at different levels. When the machine structure is vibrated for machining, the dynamic characters of the sensitive structure of machine tools are significant to determine the machining accuracy. Cheng et al. [18] determined the critical geometric errors of five-axis machine tool using the theory of multi-body system and the method of global sensitivity analysis. Chen et al. [19] conducted the volumetric error analysis in cutting point to determine the key defective components. Chen and Wen [20] carried out sensitivity analysis of spindle natural frequencies based on the structural parameters to identify the key parameters which can influence the modal parameters of machine tool. Wu and Wang [21] optimized the plan of the machine tool bed designing through local sensitivity analysis. However, the vibration characteristics of machine tool are different in different modes of vibration. In stability analysis, the regenerative chatter mostly appeared at the structural dominant frequency of the machine tool [22]. Tansel et al. [23] estimated the main mode vibrational frequency through index-based reasoning (IBR) method and identified that the dominant vibrational frequency is changed with the chatter occurring. Dombovari et al. [24] obtained the dominant vibrational frequency by calculating Fourier coefficients of the vibration velocity. Dhupia et al. [25] identified that the structural frequencies of vibration of the machines were less important compared with the vibration occurred in spindle, tool and tool holder, which used to ensure consistent dynamic behavior regardless of reconfiguration.

In general, multiple modes of vibration are excited in actual work pieces at the same time when they are processed on machine tools. Moreover, there are different effects in the machine tools in various vibration modes.

In the literature, different methods have been used for vibration analysis of machine tools. For example, operating deflection shape (ODS) is an important technique used in vibration analysis to identify the dynamical modal parameters related to the machine structure. Zhi et al. used the technique of ODS in non-destructive testing [27]. Richardson et al. presented an estimating method for the modal parameters of beam structure based on the combination with ODS and frequency response function (ODS FRF) [28]. Bae et al. considered the normalize effects from variable excitation force using a new form of coefficient scale based on the ODS FRF [29]. Liu et al. used ODS to detect the interface unsticking in reinforced concrete structures [30]. ODS is widely applied in vibration analysis of engineering due to its simple and intuitive, and the visualization and animation function with ODS which can clearly display and analyze the running deformation of machine or structure.

Therefore, in the current research, a new method is used to investigate the influence of structural vibrations on surface generation of work pieces through modal decoupling with operational deflection shape (ODS) and structural sensitivity analysis on the basis of theory above. The proposed method is significant to identify the modal mass distribution matrix and feed back different structural sensitivities. The rest of this paper is organized as follows: the next section indicates the sensitivity analysis theory of modal decoupling of ODS and explains how to identify the dominant vibrational frequencies and the sensitive structures of the machine tool. In the following section, a milling experiment setup is designed which is used to validate the effect of MVDFSS on surface creation. The main contributions of this research are shown in the final section.

Theoretical Basis and Problem Description

In the literature, there are different methods used to determine the modal parameters of vibration structures which are explained in this section.

Experimental Model Analysis

Experimental model analysis (EMA) is a method which has been mostly used in the literature (give some reference) to determine the parameters of machine tools which are vibrating by hammer. The EMA played a major role in accurately measuring of time domain signal for the excitation and vibration response to machine tool structure.

The excitation in EMA that used a shaker is a broadband and high-energy response, and the modal parameters are identified by the FRFs from the static-state shaker experiment [26]. There can be no doubt that great differences of the modal parameters occurred on operational state and static state. In actual operation, some natural frequencies of the structures can drift and even some can disappear, and the excitation generated by inertia impact between structures and cutting force is unmeasured and time variable. Therefore, the

time-varying modal parameters for the mechanical structure can directly determine the surface quality of the products, and it needs more consideration. Therefore, a new modal decoupling method based on the ODS is proposed in the current study to identify the dominant frequency vibration. Moreover, structural sensitivity analysis is applied to identify the sensitive structures and the impact of the structure vibration on surface creation is verified by experiments.

Operational Deflection Shape (ODS) Method

The proposed method of modal decompiling of ODS is presented in this section. The ODS method is significant and in this method deflection shapes of structures can be described at different frequencies. A correlation between the modal analysis method and ODS method can be expressed as a linear weighting of multiple modal shapes through modal decoupling method. Consider a structure is under vibration which has certain displacement, velocity, acceleration, mass and so on, the differential kinematic equation is illustrated in the following equation:

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

where $[M]$, $[C]$ and $[K]$ are denoted as the mass matrices, damping matrices, and stiffness matrices of the system, whereas $\{\ddot{x}\}$, $\{\dot{x}\}$ and $\{x\}$ represent the vectors of acceleration, velocity and displacement, respectively. Furthermore, $\{f\}$ illustrates the vector of exciting force, which is composed of the cutting force and the internal inertia impact force between adjacent structures on the machine tool. Fourier transform is applied to Eq. (1) to obtain the exciting force vector and vibration vector which are presented in the following equations, respectively:

$$\{X(jw)\} = [H(jw)]\{F(jw)\}, \tag{2}$$

$$[H(jw)] = \sum_{r=1}^{n} \left\{ \left[\frac{a_r\{\psi\}_r\{\psi\}_r^T}{jw - \lambda_r} + \frac{a_r^*\{\psi\}_r^*\{\psi\}_r^{*T}}{jw - \lambda_r} \right] \right\}, \tag{3}$$

where a_r, λ_r and $\{\psi\}_r$ are the modal scale factor, system poles and modal shape vector, respectively. T and $*$ represent the transpose and conjugate vector of the modal shape vector. According to Eqs. (2) and (3), the ODS will emerge as the response of some special exciting forces which is indicated in the following equation:

to distinguish the weight for each modal shape by modal decoupling method, and the following equation is the means of images after decoupling:

$$\{ODS(f_0)\} = S(\propto_1 \{\phi_1\} + \propto_2 \{\phi_2\} + \cdots \propto_i \{\phi_i\} + \cdots + \propto_n \{\phi_n\}) + Rest, \tag{5}$$

where $\{ODS(f_0)\}$ is the ODS of the structures at vibrational frequency of f_0 while ϕ_i is the modal shape vector at ith mode, $\propto_i$ represents the decoupling coefficient (%) of the different modal shape vector of ith mode, and S represents the coefficient scale to make the decoupling coefficients normalized. "Rest" is the margin vector outside the modal shapes group.

The decoupling coefficient $\propto_i$ denotes the contribution of the relevant frequency mode to the ODS. Theory shows that the stronger vibration emerges in machine tool, the value of the decoupling coefficient is greater, and the maximum value of the decoupling coefficient is indicated as the dominant vibration to the structure. Current study identifies the natural frequency and each modal shape to the structure of machine tools based on the hammer experiment. Furthermore, the ODS based on each modal shape in single acceleration time and deceleration time for machine structures is obtained from the empty moving of machine tools in the current study. The decomposition coefficient is obtained from the modal decoupling method based on a finite set which contained each modal shape. In addition, software LMS is used to finish all the functions above based on experiments as follows.

Structural Sensitivity Analysis

In vibrating structures, there are several moveable joints which exist in the structures. The level of vibration is different in different types of structures due to different value of their stiffness and has different levels of vibration energy. Sensitivity analysis of these structures is significant for structural stability and it is presented in this section. The considered method of sensitivity analysis chose the modal mass distribution matrix and because of that it can reflect different vibration energies to different structures. The considered method can identify the sensitivity of structures from

$$\{ODS(jw)\} = \left\{ \sum_{r=1}^{n} \left\{ \left[\frac{a_r\{\psi\}_r\{\psi\}_r^T}{jw - \lambda_r} + \frac{a_r^*\{\psi\}_r^*\{\psi\}_r^{*T}}{jw - \lambda_r} \right] \right\} \right\} \{F(jw)\}. \tag{4}$$

Equation (4) is the description of the ODS in frequency domain, which means that the ODS of the system at a specific frequency can be decoupled to a linear weighting of multiple modal shapes. However, current study needs

the contrast of different structures' vibration energy. The modal mass of ith mode can be given by

$$m_i = \{\varphi\}_i^T[M]\{\varphi\}_i, \tag{6}$$

where $[M]$ denotes the effective mass matrix of the system, and $\{\varphi\}_i = \{\varphi_{1i}\, \varphi_{2i} \cdots \varphi_{ni}\}^T$, and $\{\varphi\}_i^T$ are the modal shape vector and its transpose vector of ith mode, respectively. The calculated method for $[M]$ and $\{\varphi\}_i$ has been integrated to software such as ANSYS and LMS, and so on. In addition, n is the numbers of the test nodes and i is the numbers of degree of freedom (DOF). Consider a diagonal matrix $[M]$ as indicated in the following equation:

$$[M] = \begin{bmatrix} M_1 & \cdots & 0 \\ \vdots & \ddots & \vdots \\ 0 & \cdots & M_n \end{bmatrix}. \tag{7}$$

Equations (6) and (7) are considered simultaneously to get a relation as presented below:

$$m_i = \{\varphi_{1i}\, \varphi_{2i} \cdots \varphi_{ni}\} \begin{bmatrix} M_1 & \cdots & 0 \\ \vdots & \ddots & \vdots \\ 0 & \cdots & M_n \end{bmatrix} \{\varphi_{1i}\, \varphi_{2i} \cdots \varphi_{ni}\}^T = \sum_{k=1}^{n} M_k \varphi_{ki}^2. \tag{8}$$

It can be seen from Eq. (8) that the modal mass of ith mode is the sum of n items. Each item can be regarded as the kinetic energy at the corresponding DOF. If these items are used to construct a vector, it can be regarded as the spatial distribution of modal mass called as modal mass distribution vector, as indicated in Eq. (9):

$$\{m_i'\} = \{M_1 \varphi_{1i}^2\, M_2 \varphi_{2i}^2 \cdots M_n \varphi_{ni}^2\}^T = \sum_{i=1}^{n} m_i = \sum_{i=1}^{n} \left(\sum_{k=1}^{n} (M_i \varphi_{ki}) \right)^2, \tag{9}$$

where $\{m_i'\}$ is the modal mass distribution vector of ith mode. The modal mass distribution matrix consists of the modal mass distribution vectors of all modes, which is illustrated in the following equation:

$$[m'] = [\{m_1'\}\{m_1'\} \cdots \{m_1'\}]_{n \times n}, \tag{10}$$

where $[m']$ is the modal mass distribution matrix, which contains $m_1', m_2' \cdots m_n'$. Since the modal shape of the machine tool structures is an existing form of an indigenous property, it is not related to the excitation point.

It can be seen from Eq. (9) that the expressions of modal mass distribution matrix is similar to the kinetic energy expression which is given in Eq. (8). If the mass distribution matrix of some modes is mainly concentrated in local DOFs, the corresponding component can be considered as the sensitive structure in these modes.

Experimental Design and Data Analysis

The experimental validation of the proposed ODS method is presented in this section. In the considered experiments, the machine tool is SMTCL TC500 which has small size for its worktable, 0–500 Hz is selected as the frequency band for analysis which covers the frequency band of interest and the influence of structural vibration on the surface creation.

To validate the multimode vibration at dominant frequencies of sensitive structure (MVDFSS) method, first, the EMA used machine tool to identify the natural frequencies between 0 and 500 Hz based on hammer experiment. Later, the ODS in different structures directly obtained from the time domain signal source based on inertia impact of machine tool experiment, and the implementation can use the ODS modular integrated in the software of LMS. Then the value of the dominant frequencies of the machine tool is got from the modal decoupling of ODS based on the response signals of the structures in the considered machining process. Once determining the machine tool's dominant frequencies, structural sensitivity analysis is implemented to identify the sensitive structures of the machine tool at dominant frequencies. At the end, an experiment on milling operation is carried out to verify that the MVDFSS can directly determine the surface topography of the products. All tests in the considered experiments are conducted on model number TC500 machining center.

Impact Testing Setup

In the experimentation, an impact testing on the TC500 machine is conducted at first. The experimental machine tool, impact point and measuring point model are shown in Fig. 1; following types of equipment are used for experiments:

1. 21 measuring points: the manifest points as shown in Fig. 2b are arranged on different structures to obtain the modal shapes accurately. In general, there are some constrains of the choice of the measuring points. First, it is assured that the number of the measuring points is at least the same (or fewer) as the numbers of modes of the identified structure. Later, the choice of the measuring points is combined with the finite element analysis, and different options of combination are used in comparison. The choice of 21 measuring points has good efficiency between lots experiments tested in the lab.
2. The acceleration sensor type: PCB-356-A15, and the hammer type: PCB 086C03.
3. The sampling frequency: 1024 Hz.

In experiments, the natural frequencies (multi-order natural frequency) from TC500 machine tool are set as 0–500 Hz which are reckoned up EMA in current experiments. The algorithm integrated in LMS is used in the experimental modal analysis modular. Table 1 shows the natural frequency results obtained from TC500 machine structure between 0 and 500 Hz [31].

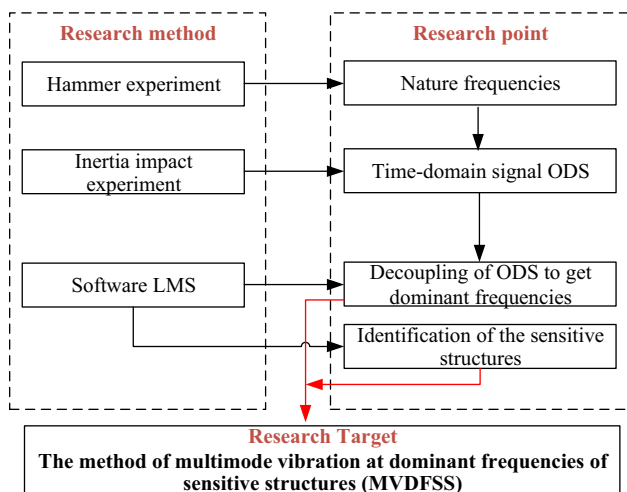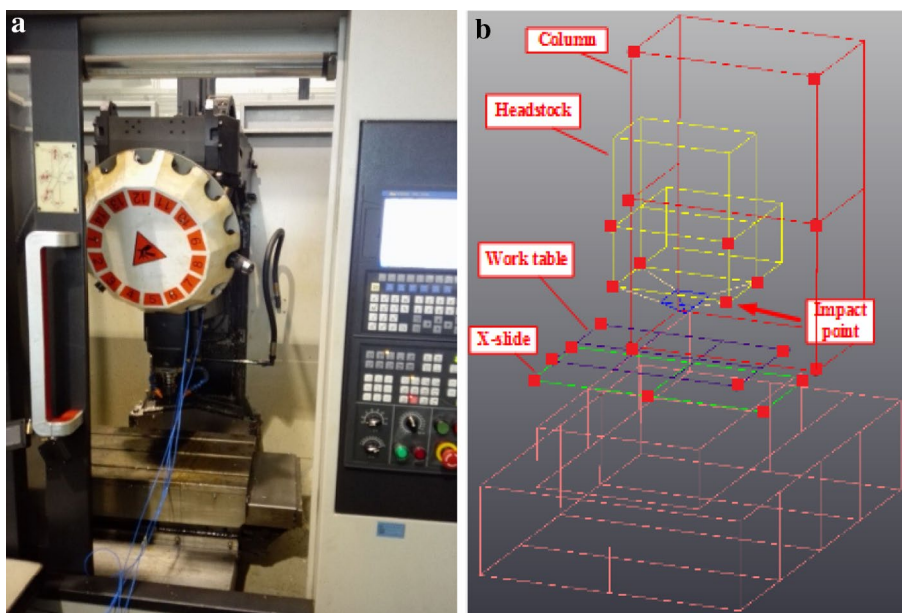**Fig. 1** Impact testing system

Fig. 2 The flowchart of research target

Once the ODS analysis is performed, the modal assurance criterion (MAC) is used to calculate the quality of the estimated modal parameters. MAC is defined as the squared correlation coefficient between two modal shape vectors, which can evaluate the accuracy of estimated modal parameter indirectly. If two vectors are different in the physical modal shape, the value of MAC is 0. Oppositely, the value is 1. Figure 3 shows the MAC values of modal shapes at different natural frequencies in Table 1.

It can be seen from Table 1 that the modal parameters estimated by EMA are well accepted.

Modal Decoupling of ODS in Machining Process

In this section, a milling experiment on the TC500 machine is used to get the response signals of the machine tool structures in machining process, which is shown in Fig. 4a. To better reflect the impact of structural vibration on the surface generation, a curve trajectory and flank milling operations are performed which is shown in Fig. 4b. The spindle speed and feed rate are set to 2200 rpm and 1500 mm/min, respectively. The transverse cutting depth is set to 0.1 mm per cycle.

The time domain response signals in single path processing are presented in Fig. 5. It can be seen from Fig. 5 that the response is variable along the path in machining process. Based on the theory of modal decoupling, the ODS of the system at a specific frequency can be decoupled into a linear weighting of multiple modal shapes. The cross power spectrums are applied to implement the modal decoupling of ODS. In the above experiment, the first eight modes are presented in Table 1 which contained one modal shape group. The ODS at different frequencies, 0–500 Hz, can be decoupled to the linear combination of the eight modes.

At first, the ODS at each frequency in 0–500 Hz is calculated through LMS peak picking method and the frequency increment is 1 Hz. Later, the modal decoupling method is
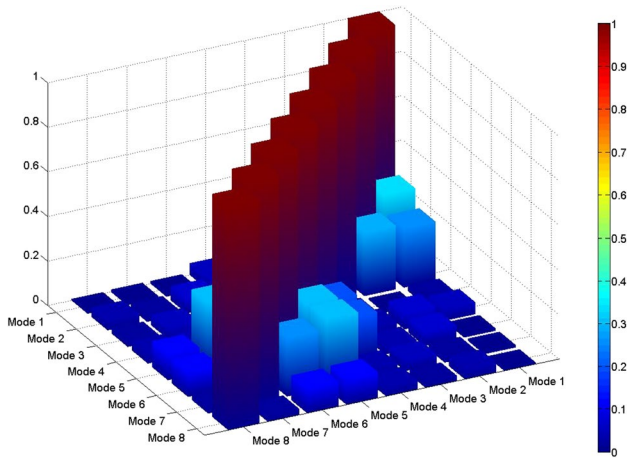
Table 1 Natural frequencies of the whole structure of the TC500 machine (0–500 Hz)

Mode	1	2	3	4	5	6	7	8
Freq. (Hz)	54.1	125.1	151.6	225.9	289.5	315.7	330.9	381.6

Fig. 3 MAC values of different modal shapes

performed to obtain the decomposition coefficient for each frequency which is composed of eight modes. Each mode corresponds to a decomposition coefficient at each frequency of the ODS. The results of 2D decomposition coefficient distribution of the first eight modes are illustrated in Fig. 6. According to the calculation from LMS peak, each of the frequency in 0–500 Hz can decompose into eight modes which are described as Fig. 5. For example, the decoupling of ODS in frequency $f_0 = 200$ is described as follows in contrast to Fig. 5:

$$ODS(f_0 = 200) = S(0.4\{\phi_1 = 54.1\}$$
$$+ 0.6\{\phi_2 = 125.1\} + 0.8\{\phi_3 = 151.6\}$$
$$+ 1\{\phi_4 = 225.9\} + 0.5\{\phi_5 = 289.5\}$$
$$+ 1\{\phi_6 = 315.6\} + 0.2\{\phi_7 = 330.9\}$$
$$+ 1\{\phi_8 = 381.6\}).$$

The proportion of the decomposition coefficient over 50% of the first eight modes is presented in Fig. 7. It can be seen

Fig. 4 The photograph of milling experiment on TC500 machine and the work piece
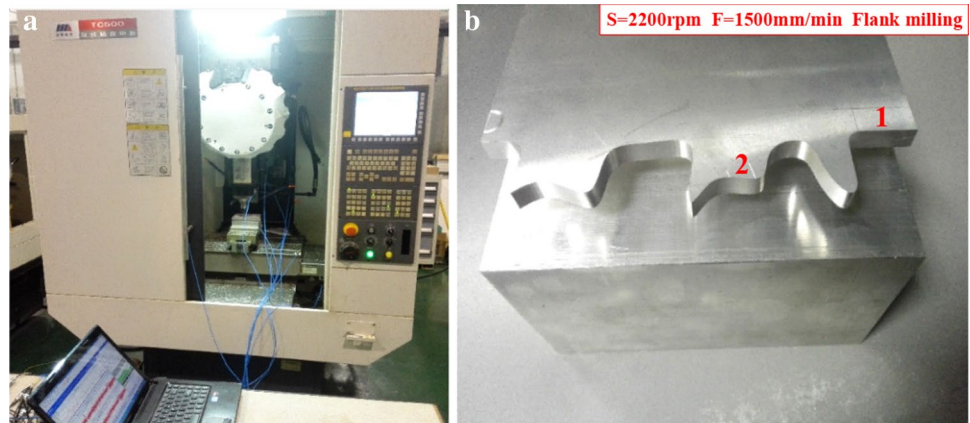

Fig. 5 The time domain response signals in single path processing
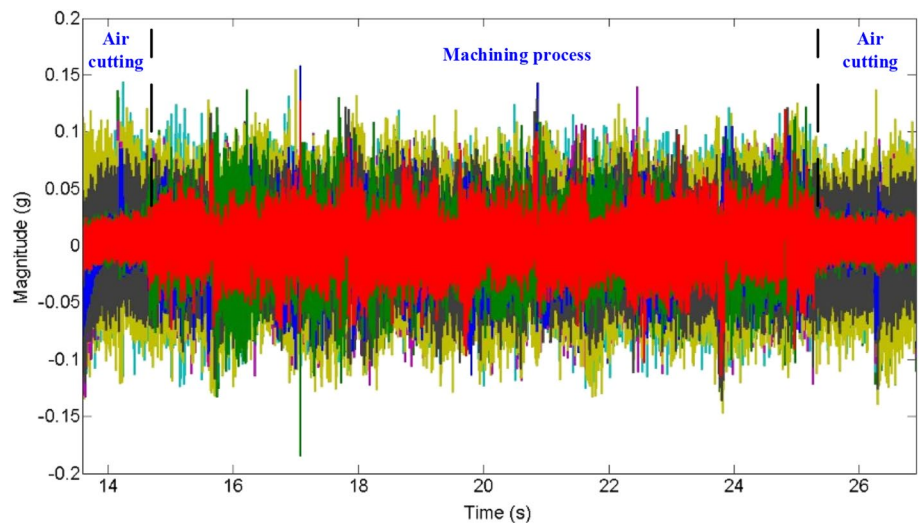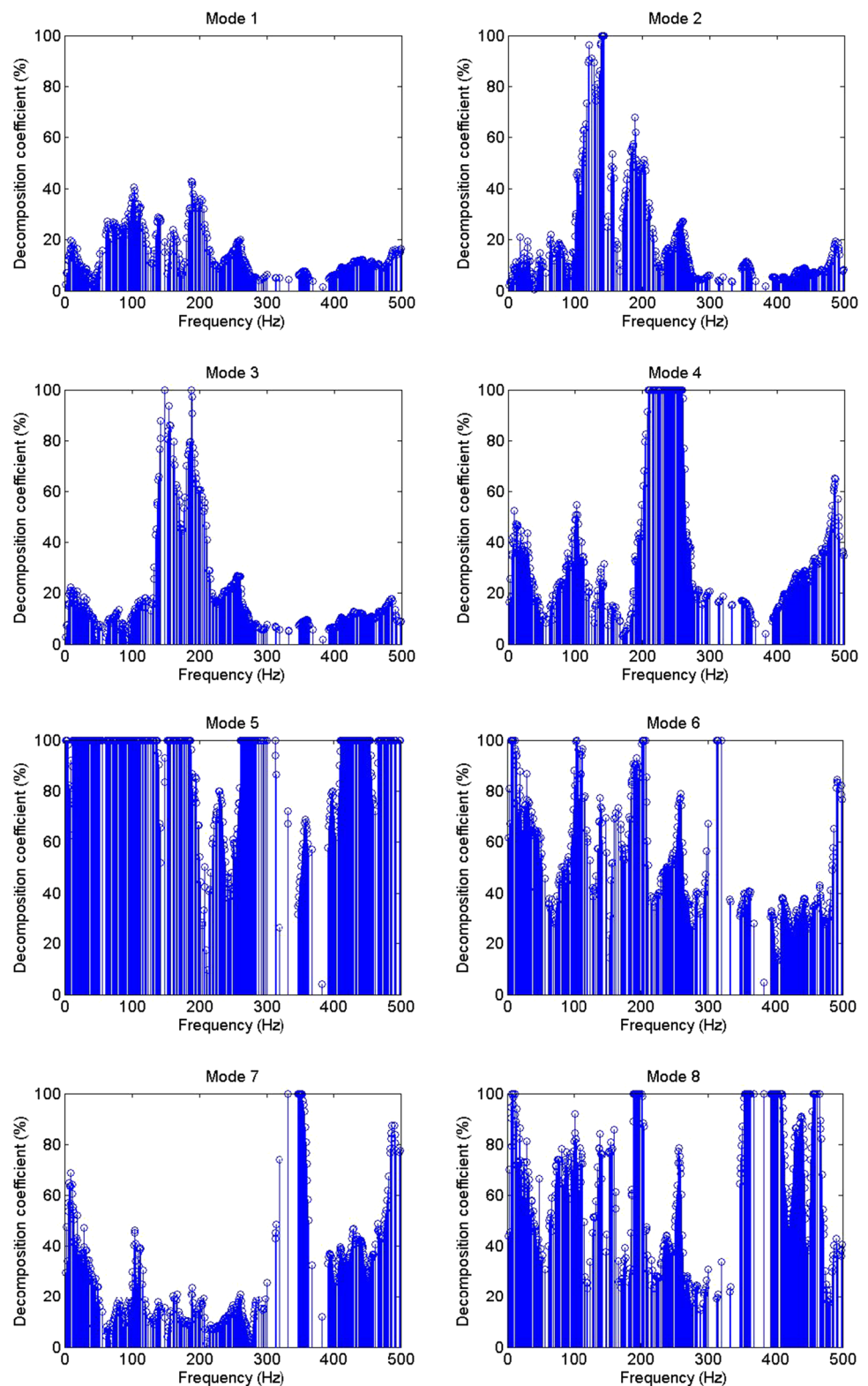
Fig. 6 The 2D decomposition coefficient distribution of the first eight modes

from Fig. 7 that the modes that accounted for a larger proportion of the decomposition coefficient over 0.5 are mode 5 (accounted for 37%), mode 6 (accounted for 18%), and mode 8 (accounted for 25%). These results indicated that maybe the three modes have an important effect on the surface creation in actual processing, and the next section will research the effect of structural vibration on surface creation by experiment.

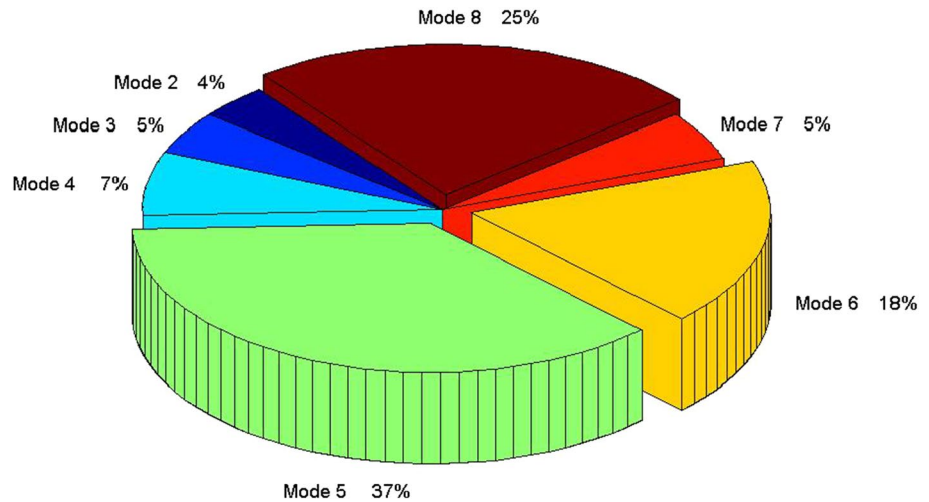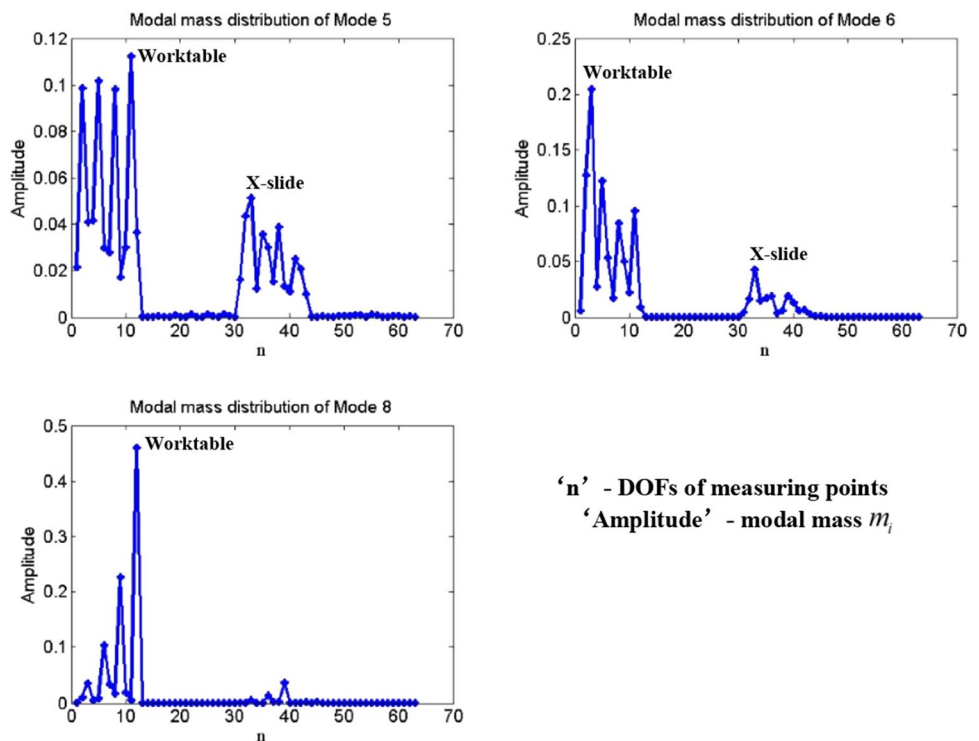**Fig. 7** The proportion of the decomposition coefficient over 50%

Fig. 8 The modal mass distribution at the dominant frequencies

'n' - DOFs of measuring points
'Amplitude' - modal mass m_i

The Effect of Structural Vibration Sensitivity Analysis

Based on the theory of structural sensitivity analysis and according to the results of modal decoupling of ODS, structural sensitivity analysis is carried out here to identify the sensitive structures at dominant frequencies. Figure 8 shows the modal mass distribution at dominant frequencies, where 'n' represents the corresponding DOFs of different measuring points. 'Amplitude' represents the modal mass m_i in Eq. (9).

There are 21 measuring points in the previous impact testing, as indicated in Fig. 1; each point is composed of

three DOFs (X, Y, Z). The no. 1 to no. 12 DOFs correspond to the worktable, no. 13 to no. 30 DOFs correspond to the column, no. 31 to no. 45 DOFs correspond to the X-slide, no. 46 to no. 63 DOFs correspond to the headstock. The modal mass distribution of considered test experiments at the dominant frequencies is presented in Fig. 8. It can be seen from Fig. 8 that the worktable and X-slide have high amplitude. These results indicate that the sensitive structures of the TC500 machine are the worktable and X-slide. It can be concluded from these results that these two parts of the machine structures can play a decisive role in surface forming in actual processing.

Fig. 9 The filtered surface topography map of the work piece at position 2
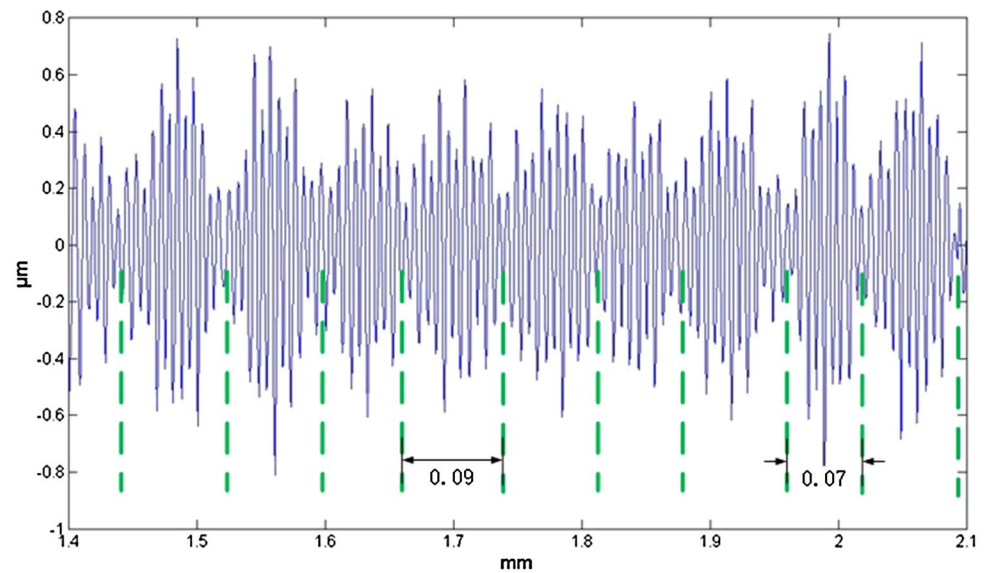
Validation of the MVDFSS Method

In this section, a milling experiment on the TC500 machine is used to validate the proposed method that the MVDFSS directly determined the surface generation of the products which is shown in Fig. 4a. Curve trajectory and flank milling are adopted to better reflect the impact of structural vibration on the surface generation which is shown as Fig. 4b. The spindle speed is set to 2200 rpm and the feed rate is set to 1500 mm/min, respectively. The transverse cutting depth is set to 0.1 mm per cycle.

Figure 9 shows the local filtered surface topography map of the work piece at position 2. It can be seen from Fig. 9 that there are some regular 'small peaks' in Fig. 9 (separated by the green dotted line), and the width of these 'small peaks' is between 0.07 and 0.09 mm, respectively. In milling machining operation, the relationship between the structural vibrational frequency and the width 'small peak' and the feed rate is $\bar{f} = \frac{F_{\text{feed}}}{60\bar{d}}$. Therefore, the calculated structural vibrational frequency reflected by 'small peak' is between 277.8 and 357.1 Hz, respectively. The frequency related to the small peak is calculated as follows:

$$\bar{f_1} = \frac{F_{\text{feed}}}{60\bar{d}} = \frac{1500 \text{ mm/min}}{60 \times 0.09 \text{ mm}} = 277.78 \text{ Hz},$$

$$\bar{f_2} = \frac{F_{\text{feed}}}{60\bar{d}} = \frac{1500 \text{ mm/min}}{60 \times 0.07 \text{ mm}} = 351.7 \text{ Hz}.$$

These results indicate indirectly that these small peaks are generated by the low-frequency vibration from the cutting tool. In particular, the frequency band is consistent with the dominant vibrational frequency.

To validate the correctness of the proposed MVDFSS method, the dominant frequency vibration with all modes of vibration and the response between the sensitive structures and the insensitive structures need to be compared. Figure 10 shows the FRF comparison from different structures of TC500 machine (0–500 Hz). It can be seen that the vibration states of different structures at the same frequency are different. Moreover, these vibrations have higher amplitude at the dominant vibrational frequency than at weak modes, and the sensitive structures at the main mode vibrational frequency (mode 5, 6, 8) have higher amplitude than the insensitive structures. It is important to note that mode 7 has the highest amplitude in all eight modes, but mode 7 is not the dominant vibrational frequency; it can only be explained that the dominant frequency vibration is crucial, instead of the larger amplitude–frequency vibration. Therefore, the MVDFSS method is more effective than the FRF method.

Conclusions

Vibration in the machine tools is significant for improving the quality of machined products. Therefore, the current study proposed a new method of modal decoupling of ODS and structural sensitivity analysis to identify the structural vibration on surface creation in milling process. The dominant frequency vibration and the sensitive structures of the machine tool are identified through modal decoupling of ODS and structural sensitive analysis, respectively. The proposed method is validated by the experiment as results. Following are the highlights and conclusions of this paper:

Fig. 10 The FRF comparison of different structures of TC500 machine (0–500 Hz)
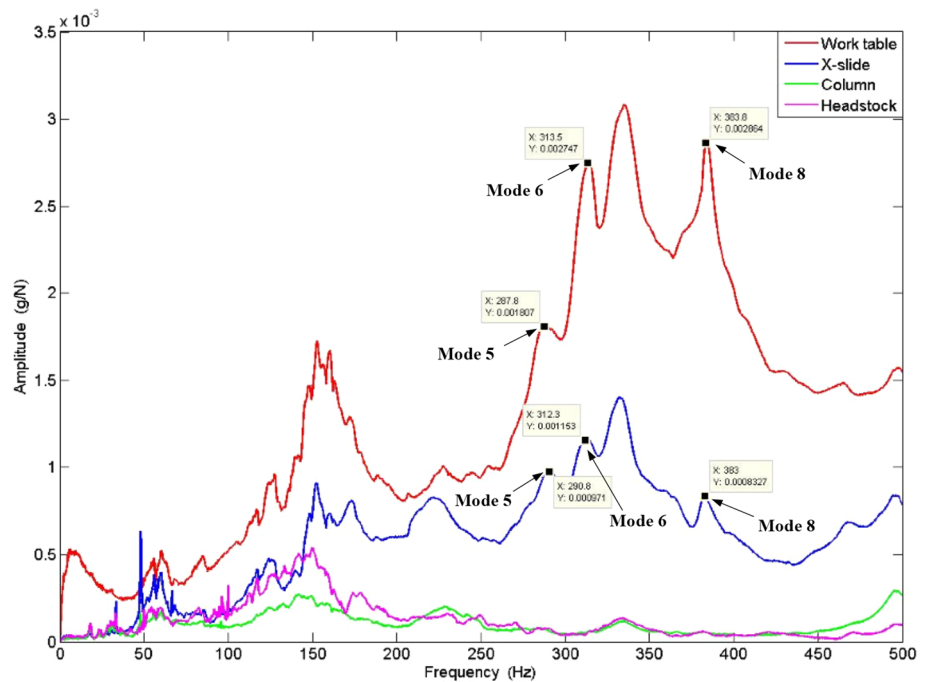
1. The MVDFSS plays a decisive role in the surface creation in milling machining operation.
2. The dedication of the main mode vibrational frequency to the vibration of machine tool is larger than weak modes, instead of the larger amplitude–frequency.
3. The sensitivity is different for different structures or the same structure at different modes. The sensitive structures have higher vibration energy than the insensitive structures.
4. For a linear system, the ODS of the machine tool structures at a specific frequency can be decoupled to a linear combination of multiple modal shapes through modal decoupling method.

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