

The Integrated Optimization of High-Speed NC Milling Machine Based on the Assembly of Modal Force Matrix

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Abstract: A relative displacement between cutter and workpiece formed by Incremental linear superposition, which is created by all joints of the whole structure system of high-speed NC milling machine, influences the machining precision. In this study, connection relations of major components in machine tool are analyzed. The corresponding minimum displacement of vibration mode between the principal axis and workpiece in low order mode of the whole machine is adopted as the evaluation indicator of dynamic performance advantages and disadvantages. The comprehensive analysis is conducted on natural frequency and vibration mode of machine tool through the assembly of modal force matrix, so as to search out the corresponding maximum modal frequency and weak structural links, to correct modal frequencies of main structural components for separation, to avoid substantial increase of the whole vibration amplitude caused by the fact that the driving frequency of the whole machine is close to the modal frequency of component and to achieve the goal of optimizing the whole structure. Compared with the whole finite element, the analytical accuracy error of this method is less than 2.5% and the analysis time can be reduced by 50%.

Keywords: Assembly of modal force matrix, high-speed NC milling machine, integrated optimization, Modal Frequency Separation (MFS)

INTRODUCTION

The integrated optimization design of the entire machine tool is the key to improve the machining accuracy. At present, various problems of machine tool in operation caused by the effect of dynamic force can be basically solved, systematic analysis can be carried out on machine tool with the application of kinetic model and dynamic optimization design can be completed in the design stage. Because NC milling machine is developing in the trend of overloading, high speed, high accuracy and reducing dead weight as much as possible, dynamic performance requirements are much higher than that of traditional machine tool. Thus, requirements on dynamic design and characteristic analysis are extremely crucial. The high-speed machining technique has such requirements on major components of machine tool: relatively high specific stiffness, good dynamic characteristics, sufficient static stiffness, good hot deformation characteristics, etc. Scholars both at home and abroad have made a large amount of studies on dynamic performance of high-speed and complex NC machine and obtained plentiful achievements. Chen (2000) and Zatarain (1998) derived the system dynamic equation of the whole machine tool

with modal synthesis method; (Huang, 2005) conducted finite element pretreatment with HYPERMESH and carried out machine modal analysis by finite element calculation software ANSYS; (Zhang, 2001) proposed an evaluation methodology of dynamic characteristics of the whole machine based on the principle of Fuzzy Mathematics; (Mao, 2002) adopted Riccati transfer matrix method and finite element method based on distributed mass and conducted fitting correction on kinetic model by parameter optimization; (Hu, 2003) established a finite element model of internal grinding machine, found out weak links of machine tool through modal force method and optimized them. In order to improve the performance of the whole machine, finite element mechanical model of NC milling machine appears quite complex, bandwidths of freedom degree and stiffness of accurate model are extraordinarily large. Because of ceaseless structural modifications and optimizations and repeated performance iterative analysis, there might be eigenvalue problems of large degree of freedom in dynamic finite element analysis, leading to excessive waste of time in high frequency calculation on computer. Simplified calculation and improvement of calculation accuracy is key problems faced by finite element analysis of high-speed NC milling machine.

In fact, processing quality is ultimately dependent on the relative vibration displacement between workpiece and cutter instead of the absolute displacement of each component. At the inherent frequency of a certain mode, despite the large absolute dynamic displacement of main components, the relative dynamic displacement would be quite small and the machine tool would have good property for processing if vibration phases are consistent. According to this idea, the substructure modal technology is introduced, which breaks down a complex structure into several major substructures and conducts finite element analysis on each substructure separately. Because the matrix bandwidth of each component is much smaller than that of the whole machine, computation load of analysis is far smaller than that of the overall finite element analysis, even though the sum of freedom degrees of all components equals the freedom degree of the whole machine. And naturally, the efficiency is much higher. Then establish the system dynamic equation in line with interface connecting conditions between substructures and assemble with modal force matrix, so as to conduct comprehensive analysis on inherent frequency and vibration mode of machine tool, to calculate directly relative dynamic displacements between workpiece and cutter at all inherent frequencies, to find out the maximum modal frequency that is corresponding to relative displacement and to search out the weak link of a certain structure of the whole NC milling machine. After that, reduce their relative dynamic displacement by structural optimization and improvement to realize the dynamic optimization design of machine components. The order of secular equation is decided by the freedom degree of interface node. Quadratic coordinate transformation and assembly of substructures is unnecessary if there is no matrix inversion, so that calculation load can be largely reduced and the eigenvalue problem and the dynamic response problem of the whole system can be resolved by solving this equation. From the perspective of dynamics, it has to improve dynamic characteristics of machine tool in order to improve its processing performance. Thus, the amount of machine vibration can be controlled within permitted scope and there will be no cutting self-excited vibration when a machine tool is operating within rated power range (momentary overload is permitted). It can be known from the actual operation test of original NC milling machine that the first order mode is likely to be excited. Therefore, for the first order mode, the minimum relative displacement of vibration mode between the principal axis (cutter) and work piece can be used as the evaluation function of dynamic performance advantages and disadvantages for choosing a design proposal of machine optimization.

DYNAMICS MODELLING AND ANALYSIS OF NC MILLING MACHINE

Establish finite element analysis model of the main components and assemble them, forming the junction surface model and the exciting force model. And then solve eigenvalues and eigenvectors of components with the finite element method. On this basis, establish a matrix of frequency response function of components. For a sinusoidal excitation with the frequency of ω , its response function can be expressed as the formula:

$$M \ddot{x}(t) + K \dot{x}(t) = f(t) \quad (1)$$

$$\text{So: } x(t) = \Phi [\Lambda - \omega^2 I]^{-1} \Phi^T f(t) \quad (2)$$

$$\text{or: } X = H(\omega) F \quad (3)$$

where,

M : Mass matrix

K : Stiffness matrix

$x(t) = X e^{j\omega t}$: Dynamic displacement vector of freedom degree

$f(t) = F e^{j\omega t}$: Force vector of freedom degree.

Only inherent frequencies and vibration modes of component of first l orders are calculated in actual dynamic analysis. Therefore,

$$\Phi = [\Phi_1, \Phi_2, \dots, \Phi_l] \quad (4)$$

$$\Lambda = \text{diag} [\omega_1^2, \omega_2^2, \dots, \omega_n^2]$$

Φ_i : Component eigenvector of the i^{th} order;

ω_i : Inherent frequency of component of the i^{th} order;

$H(\omega) = \Phi [\Lambda - \omega^2 I]^{-1} \Phi^T$ is called the frequency response function matrix of component. In order to eliminate modal truncation errors as much as possible, the modifier formula $H(\omega)$ can be derived according to static residual flexibility:

$$H(\omega) = \left[(\Lambda - \omega^2 I)^{-1} - \Lambda^{-1} \right] \Phi^T + [K]^{-1} \quad (5)$$

Then assemble the modal force matrix of the whole machine in accordance with the junction surface model and the modal secular equation and calculate the inherent frequency of the machine by the modal force matrix. On the junction surface between components, firstly separate sub-matrixes of frequency response function related to the junction surface from the frequency response function matrix of component, so:

$$\begin{bmatrix} X_i^N \\ X_j^N \end{bmatrix} = \begin{bmatrix} H_{ii}^N(\omega) & H_{ij}^N(\omega) & \dots \\ H_{ji}^N(\omega) & H_{jj}^N(\omega) & \dots \\ \dots & \dots & \dots \end{bmatrix} \begin{bmatrix} F_i^N \\ F_j^N \\ \dots \end{bmatrix} \quad (6)$$

Where,

- N : Serial number of component
- I, j : Serial number of junction surface
- X_i^N : Displacement amplitude vector of component N on junction surface i
- F_i^N : Acting force amplitude vector of component N on junction surface i
- $H_{ii}^N(\omega)$: Self-response sub-matrix of frequency response function of component N on junction surface i
- $H_{ij}^N(\omega)$: Cross response sub-matrix of frequency response function of component N on junction surface i, j
- F^{12} : The butted joint force of interface of substructures 1 and 2

The eigenvalues can be solved from $|H(\omega)| = 0$. $H_i(\omega)$ can be acquired by simple matrix multiplication and then get the modal force equation according to relation, displacement and contact stiffness of acting force on junction surface between components:

$$\bar{H}(\omega) \bar{F}(\omega) = 0 \quad (7)$$

$\bar{H}(\omega)$: Modal force matrix

$\bar{F}(\omega)$: Modal force vector. Thus $\det |H(\omega)| = 0$

INTEGRATED OPTIMIZATION DESIGN OF THE WHOLE STRUCTURE OF NC MILLING MACHINE

Modal force transfer matrix of NC milling machine:

According to conference case, the research prototype of this study is a floor-type gantry mold NC milling machine of a certain model. Figure 1 is the virtual design model of the mold NC milling machine and Fig. 2 is the schematic diagram model of the NC milling machine.

According to modal force equation, the whole modal force matrix of Fig. 2 can be acquired:

$$\bar{H}(\omega) = \begin{bmatrix} H_1(\omega) + K_1^{-1} & H_{12}(\omega) & \dots & H_m(\omega) & H_1(\omega)H_m(\omega) \\ H_{12}(\omega)^T & H_{22}(\omega) + K_2^{-1} & \dots & H_{2m}(\omega) & H_2(\omega)H_{2m}(\omega) \\ \dots & \dots & \ddots & \dots & \dots \\ H_m(\omega)^T & H_{2m}(\omega)^T & \dots & H_m(\omega) + H_m^{-1} & H_m(\omega)H_{mm}(\omega) \end{bmatrix} = [\bar{H}_j(\omega), \bar{H}_m(\omega)] \quad (8)$$

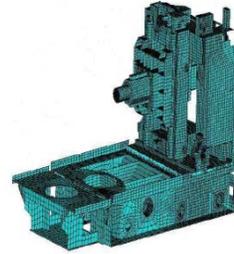


Fig. 1: Virtual design model of CNC

$$\bar{F} = (F_1, F_2, \dots, F_m; F_t, F_h)^T = (F'_j, F'_{th})^T \quad (9)$$

In the above formula, K contains damping of junction surface. $H_{ii}(\omega)H_{ii}^{N1} + H_{ii}^{N2}$ stands for the sum of frequency response function sub-matrices of two components N1 and N2 on junction surface i. $H_{ij}(\omega)$ represents cross response sub-matrix of frequency response function of the same component on junction surface i, j. t and w represent serial numbers of cutter and workpiece respectively. m is the total number of junction surfaces of the whole machine. \bar{F}_i stands for the force vector on junction surface i and \bar{F}_{th} stands for the acting force vector between cutter and workpiece. It can be got by combining formula (4) and (8) that:

$$\bar{H}_j(\omega)\bar{F}_j = -\bar{H}_{th}(\omega)\bar{F}_{th} \quad (10)$$

Cutting load and inherent frequency required by the design will be substituted into formula (10) as the excited frequency. And then substitute \bar{F}_j into formula (3) after being solved so as to get vibration mode and amplitude of each junction point of the whole NC milling machine at this excited frequency.

Modal frequency separation and structural optimization of NC milling machine:

In structural optimization design of NC milling machine, it is able to find out easily an optimal design proposal of a structural single piece by static and dynamic design of major components. The whole structure is composed of multiple structural single pieces. Good dynamic characteristics of single piece do not mean good dynamic characteristics of the whole machine. There are a variety of proposals for component optimization design and it is easy to get the optimal one. And there will be more design proposals of the whole machine after arrangement and collocation of multiple components. It is unrealistic to calculate, compare and

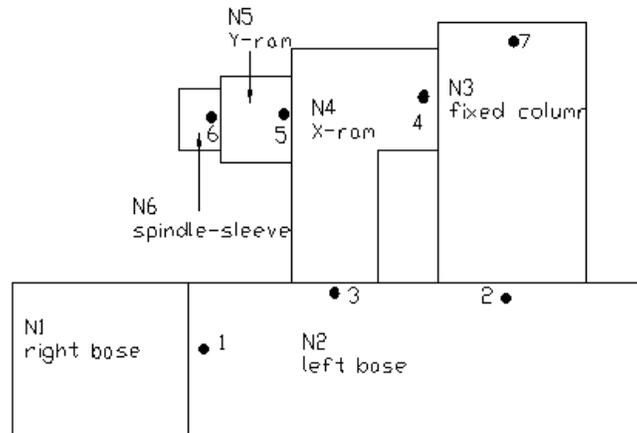


Fig. 2: System schematic of CNC

Table 1: Five-order modal frequency of main parts before optimization

Frequency	Right base	Left base	Spindle-sleeve	X-ram	Y-ram	Fixed-column
1 st order	114*	242*	123*	172	322	295
2 nd order	209*	266	134	215*	434	397
3 rd order	353*	367*	276	302	586	489
4 th order	412	305	343	461	645	587
5 th order	573	402	451	622	858	675

#,*: Represent similar modal frequencies of the same order to parts

analyse the design proposal of each combination and even more impossible to carry out the dynamic design of the whole structure of NV milling machine with methods like orthogonal design, because this would lead to a substantial increase in design time and cost. The modal frequency correction method means that modal frequencies of each major component in first few orders separate within a certain range so as to achieve the final goal of improving the dynamic performance of the whole machine. If components share the same or approximate frequencies and the driving frequency of the structure is exactly close to the modal frequency of component, the structural vibration amplitude will increase exponentially once the machine is energized. For this reason, in structural design of complex machine, the modal frequency correction must be conducted on each assembled component so that the mechanical structure can be equipped with good dynamic performance. Specific steps are like this: calculate key modal frequencies of major structural components in first few orders and then correct the modal frequency distribution of each major assembled component within the inherent frequency range of the whole machine. In dynamic structure design of machine tool, the modal frequency of the whole machine has to be taken as the criterion of component modal frequency correction in order to guarantee the improvement of dynamic characteristics of the whole structure.

Modal frequencies of major components of NC milling machine in first five orders are as shown in Table 1. It can be seen from Table 1 that: modal frequencies of lathe bed, upright post and crossbeam are close in the first order; modal frequencies of lathe bed and upright post are close in the second order.

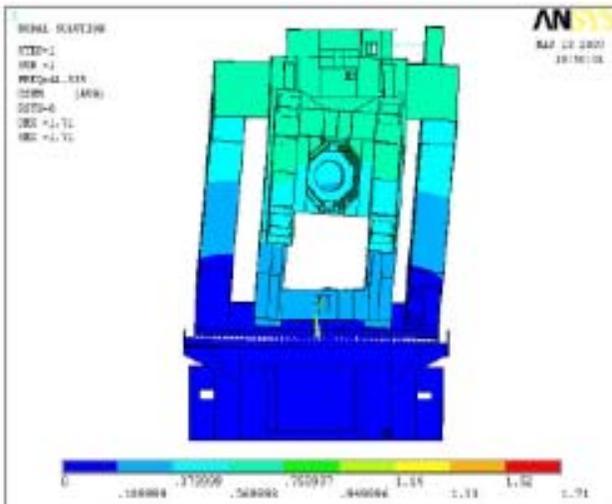
Modal frequencies of lathe bed and upright post are close in the third order; If components share approximate frequencies and the driving frequency of structure is exactly close to the modal frequency of component, the vibration amplitude of the whole structure will increase once the NC milling machine is energized. Hence, it is able to realize the dynamic performance optimization of the whole NC milling machine by modal frequency correction and separation of major components. Specifically, first calculate key modal frequencies of major components in first few orders and then correct the modal frequency distribution of each major component so as to achieve the ultimate goal of improving the whole dynamic performance. Modal frequencies of each component in the first five orders after optimization are as shown in Table 2.

With modal frequency correction, modal frequency of each optimized component can be separated, avoiding the fact that the vibration amplitude of the whole structure might increase when the NC milling machine is energized. It is able to enhance the anti-

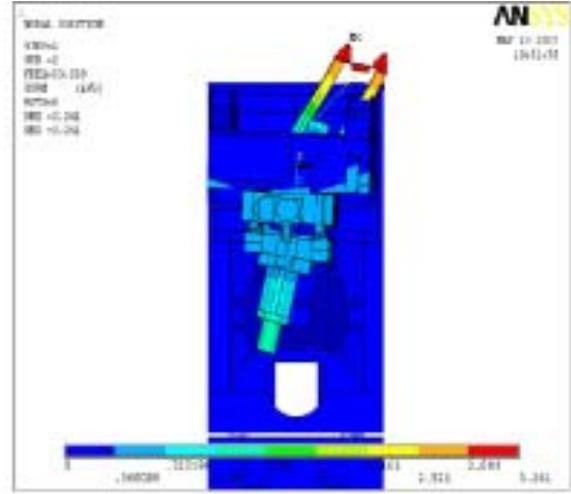
Table 2: Five-order modal frequency of main parts after optimization

Frequency	Right base		Left base		Spindle-sleeve		X-ram		Y-ram		Fixed-column	
	FEA	MFS	FEA	MFS	FEA	MFS	FEA	MFS	FEA	MFS	FEA	MFS
1 st order	124	120	262	267	143	145	172	176	322	329	295	301
2 nd order	235	249	321	317	193	197	215	221	434	440	397	403
3 rd order	355	362	387	377	206	209	302	306	586	590	489	500
4 th order	433	440	456	463	376	382	461	470	645	651	587	591
5 th order	571	565	597	608	467	477	622	630	858	862	675	682

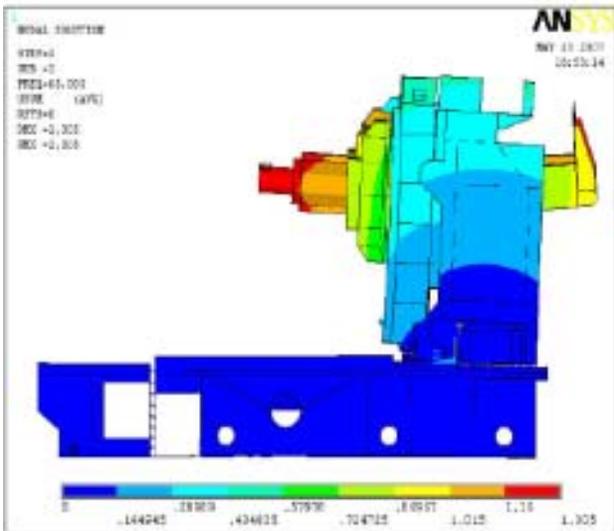
Structures of fixed upright post, X ram and Y ram are relatively reasonable before optimization. Modal frequency will not change much even if there is an appropriate improvement. So there is no change for them. The frequency can be separated from that of other components by improving structures of the left base, the right base, the main spindle quill, the supporting frame and the fixed upright post



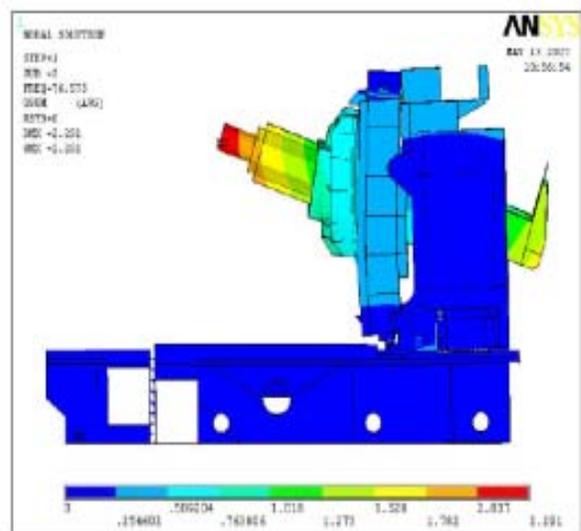
(a) 1st order vibration mode



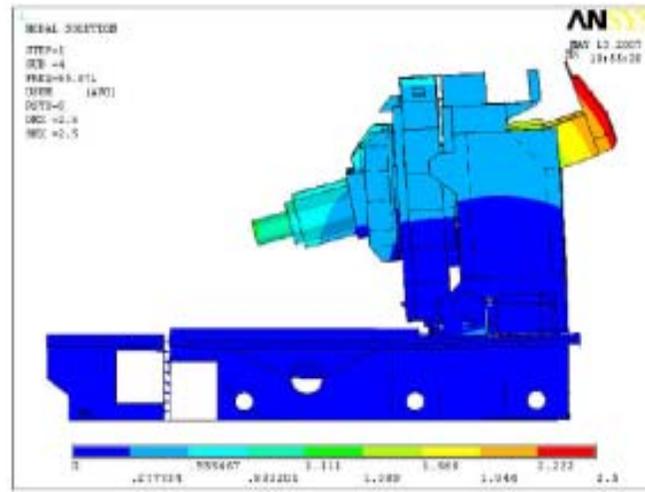
(b) 2nd order vibration mode



(c) 3rd order vibration mode



(d) 4th order vibration mode



(e) 5th order vibration mode

Fig. 3: First-five-order vibration mode

Table 3: Comparison of inherent frequency, relative displacement of vibration and response displacement before and after optimization

Machine model	First natural Frequency (Hz)	Relative displacement of vibration modes between spindle and workpiece (μm)	displacement of spindle-side (μm)	Second natural Frequency (Hz)	Relative displacement of vibration modes between spindle and workpiece (μm)	displacement of spindle-side (μm)
Before optimization	56	7.2	5.1	112	7.3	5.9
After optimization	68	6.1	4.3	134	5.8	5.0

vibration ability of large component by increasing its inherent frequency and improving its major vibration mode and accordingly to achieve the goal of avoiding low-order sympathetic vibration, reducing deformation and improving machining precision.

The Fig. 3 is the whole schematic diagram of the original machine tool in first five orders after improvement and optimization on weakness.

OPTIMIZATION OF THE WHOLE MACHINE AND EXPERIMENTAL ANALYSIS

Carry out the vibration excitation test on the machine tool so as to excite the vibration easily happened spontaneously. At each vibrating point of the high-speed mold NC milling machine, detect and measure response parameters with an accelerated vibration pickup and identify modal parameters of the high-speed mold NC milling machine. Conduct analysis on modal and inherent frequencies of the high-speed NV milling machine, shown in Table 3.

Measured data of inherent frequency of the whole machine are consistent with the optimization results.

After the optimization, inherent frequencies of the whole machine in the first and the second order are increased (respectively by 21 and 20%). Relative

displacement of vibration mode of principal axis and workpiece, which are corresponding to the frequency and vibration mode in the first order, is also reduced by more than 15%. The response displacement of the principal axis end in low-order inherent frequencies (the first and the second order) is reduced by 17% compared to that before the optimization.

CONCLUSION

- In the dynamic analysis of complex NC milling machine, with the method of modal force matrix assembly described in this study, the analysis speed is remarkably faster than that of the finite element method (shorten more than half of the time) and the error of the analysis accuracy is no more than 2.5% compared to the finite element method. It can be used as a method of verifying the finite element analysis as well as a substitution for the finite element in dynamic analysis. The more complex the design object, the more tremendous the stiffness matrix bandwidth of the whole finite element model. Thus, design modifications and optimization iterations will become more and advantages of this method will be more significant.

- The aim of modal force matrix and modal frequency correction is to avoid sympathetic vibration, to reduce the relative displacement of cutter and workpiece at sympathetic vibration, to find out weak structures of machine tool for optimization and to separate modal frequencies of main structural components, so that the dynamic performance of the whole machine can be improved. The rapid development of machine tool requires enhancement of the whole stiffness, so it is of vital importance to increase the inherent frequency of the whole machine appropriately. Attentions should be paid to wall thickness selection of large structure and reasonable installation of ribbed slab. Do not weaken the structural stiffness blindly in order to reduce weight.

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