Study on dynamic characteristics analysis of CNC pipe thread lathe based on the energy of modal effective mass

Xu Guang-chen¹,², Sun Xing-wei¹
¹Mechanical Engineering College of Shenyang University of Technology, Shenyang, People’s Republic of China
²Mechanical and Power Engineering College of Yingkou Institute of Technology, Yingkou, People’s Republic of China
E-mail: kanu012@163.com

Abstract: Owing to the modal analysis of complex structure system exist in the problems such as the local mode cannot be identified and the overall modal vibration mode cannot be extracted, this study deduces a formula of modal effective parameters based on the kinetic energy and introduces to the simulation of mechanical characteristics of the SCK230 CNC pipe thread repair the lathe bed. The simulation shows that this method can identify the local mode of three directions effectively, and determines whether the numbers of solved modalities satisfy the requirement of precision. In order to improve the natural frequencies of the bed and analyse the influence of the opening of the bed on its dynamic performance, three optimisation schemes are proposed based on this, and verified the feasibility of the optimisation scheme. Through structural optimisation, improve the dynamic characteristic of the lathe bed and reduce the manufacturing cost, and the optimisation of the structure can be achieved without experimental modal analysis.

1 Introduction

With the rapid development of drilling industry, the demands of the drill string are increasing. At present, most of the drilling tools production enterprises in China can only rely on the transformation of old machine tools to produce drilling tools, it is imperative to independently research and develop special-purpose drill boring drill pipe for CNC pipe thread lathes [1]. In the complex system of the machine tool, the bed is a very important part, its layout and size parameters determine the dynamic performance of the machine tool. In the dynamic design of the machine tool, the unreasonable design of the bed body will lead to the inadequacy of the stiffness of the bed, produce various vibration and deformation, and then reduce the machining accuracy. Therefore, it is particularly important to analyse the dynamic characteristics of components during the design phase of the machine tool [2, 3].

There are lots of local modes in the finite element modal calculation of the structure, which make the results of the analysis very confused and difficult to distinguish, and the modal effective mass provides an effective method for the analysis of the ‘meaning’ of the vibration mode. The concept of modal effective mass was first proposed by Clough [4, 5]; Palanichamy [6] presented a modal effective mass derivation method based on vibration control equations and a derivation method of using the Craig–Bampton substructure method, and applies the modal effective mass to the carrier. Zhao [7] set up a random vibration analysis technique based on the effective modal mass optimisation method to obtain the system level random vibration response quickly and effectively.

In this paper, the vibration problems of SCK230 CNC pipe threading lathes are studied, and a modal effective mass formula based on kinetic energy calculation is deduced, through the energy distribution and modal effective mass analysis of the finite element model of the lathe bed, the weak link and the local mode of the structure are found, the analysis results can provide the theoretical basis and design guidance for the overall design of the structure.

2 Finite element model of pipe thread lathe bed

The establishments of the finite element model are the basis of the finite element analysis, and the choices of reasonable modelling method are the key to the establishment of an accurate model. The establishment of the model can be directly in the finite element analysis software to create a solid model, but also in the computer-aided design system to create a corresponding model, and then imported into the ANSYS environment. In this paper, the bed entity model is made using three-dimensional (3D) software. The 3D solid model of the CNC pipe thread lathe is shown in Fig. 1, and the 3D entity model of the bed is shown in Fig. 2. The bed material is HT250, the elastic modulus $E$ is 150 GPa, the density $\rho$ is 7340 kg/m$^3$, and Poisson's ratio $\nu$ is 0.27 [8].

Meshing is the key step of finite element analysis. The quality and efficiency of the finite element analysis are directly affected by the meshing. Therefore, in order to facilitate the meshing, the bed models need to be properly handled before meshing. Ignore all the characteristics of small models, including chamfer, fillet, and small holes, processing linearisation and planarisation for the small taper and the small curvature surface in the model. In the process of meshing, using 10-node tetrahedron structure of the solid 187 unit and the free division of the grid, the bed finite element model of pipe thread repair lathe is shown in Fig. 3. The whole model had 39,121 nodes and 22,699 units.

Fig. 1 Model of the lathe

Fig. 2 Entity model of the bed
3 Calculation of modal effective mass based on kinetic energy

The modal effective mass is a dynamic characteristic of the structure, and its size directly reflects the dynamic mass of each mode in the fundamental excitation of the structure. The large modal effective mass is easily stimulated by the basic excitation, and it is the main contributor to the response of the system. However, the small modal effective mass is not easy to be excited by the basic excitation. In addition, the calculation of modal effective mass also determines whether a certain mode is the main direction of the mode and the size of modal participation factors.

3.1 Calculation of kinetic energy

In the finite element structure system, when a unit \( s \) vibrates in the \( i \)th mode, the maximum kinetic energy \( T_{i} \) generated can be expressed as [9]

\[
T_{i} = \frac{1}{2} \omega_{i}^{2} [A^{(i)}]^T [M] [A^{(i)}] \tag{1}
\]

where \([A^{(i)}]\) is the amplitude matrix of the unit \( s \) in the \( i \)th mode; \([M]\) the inertia matrix of the unit \( s \); \( \omega_{i} \) the natural frequency of the \( i \)th mode.

The total energies of the whole structure in the vibration are the sum of all the units energies. Therefore, the maximum kinetic energy \( T_{r} \) generated by the vibration of the \( r \)th mode can be expressed as

\[
T_{r} = \sum_{i=1}^{N} T_{i} \tag{2}
\]

where \( N \) is the number of modal participation factors.

The ratio of the kinetic energy of the unit \( s \) to the whole structure is called the kinetic energy distribution rate of the unit \( s \) in the \( r \)th mode. The units with a high rate of kinetic energy distribution should reduce its mass.

3.2 Modal mass

The motion equation of an undamped multi-degree-of-freedom system is [10–12]

\[
[M][\ddot{x}] + [K][x] = [F] \tag{3}
\]

where \([M]\) is the mass matrix; \([K]\) the stiffness matrix; \([\ddot{x}]\) the acceleration vector; \([x]\) the displacement vector; \([F]\) the activation function.

The eigenvalues and eigenvectors are obtained by solving the homogeneous equation (3). The eigenvectors are the vibrational modes of the structure.

The modal mass of the \( i \)th mode of unit \( s \) is expressed as

\[
M_{i} = [A^{(i)}]^T [M] [A^{(i)}] \tag{4}
\]

Thus, the kinetic energy formula (1) is brought into formula (4) to derive the modal mass formula based on kinetic energy is

\[
M_{i} = \frac{2T_{i}}{\omega_{i}^{2}} \tag{5}
\]

3.3 Effective modal mass

Define a coefficient vector \( \{L\} \) as [13–15]:

\[
\{L\} = [A^{(i)}]^T [M] [\{r\}] \tag{6}
\]

where \( \{r\} \) is the displacement vector used to represent the displacement of each structural freedom due to the static displacement of the unit bearing.

The modal participation factor and modal effective mass of the \( i \)th mode are

\[
\Gamma_{i} = \frac{\{L\}_{i}}{M_{i}} = \frac{\{L\}_{i} \omega_{i}^{2}}{(2T_{i}/\omega_{i}^{2})} = \frac{\{L\}_{i} \omega_{i}^{2}}{2T_{i}} \tag{7}
\]

\[
m_{\text{eff},i} = \frac{\{L\}_{i}^{2}}{M_{i}} = \frac{\{L\}_{i}^{2} \omega_{i}^{2}}{(2T_{i}/\omega_{i}^{2})} = \frac{\{L\}_{i}^{2} \omega_{i}^{2}}{2T_{i}} \tag{8}
\]

The sum of the effective mass of the first \( N \) modes is the cumulative modal effective mass. In the modal analysis, only a limited number of modes are extracted for the reason of the calculation, and the effective mass of the modal can judge whether the number of the extracted modes satisfies the problem solving accuracy, and the effective masses of the accumulated modal of the finite mode are at least 90% of the total mass.

4 Modal analysis of the bed

The modal of the structure may contain a large number of local modes in vibration, and it is one of the most important problems to identify the modal identification of the structure. Through modal analysis of the structure, the modal effective parameters are introduced, combined with modal shape and energy analysis, the whole and local modes can be identified effectively, and the contribution of each order mode in the corresponding direction is determined.

4.1 Analysis of the natural frequency and energy

The machine bed is divided into meshes, and a fixed constraint is applied on the bottom to carry out modal analysis, and the first six-order information of the bed are extracted, as shown in Table 1.

| Order | Natural frequency, Hz | Kinetic energy, J | Vibration mode description |
|-------|-----------------------|-------------------|---------------------------|
| 1     | 302.82                | 1,810,031         | XY plane swings along the Z-direction |
| 2     | 350.35                | 2,422,950         | XY plane twists           |
| 3     | 489.65                | 4,732,537         | XY plane swings           |
| 4     | 520.45                | 5,346,813         | XY plane swings along the Y-direction |
| 5     | 686.10                | 9,291,778         | the top of bed swings along the Z-direction |
| 6     | 693.01                | 9,480,112         | the top of bed twists along the Z-direction |

This is an open access article published by the IET under the Creative Commons Attribution License (http://creativecommons.org/licenses/by/3.0/)
Fig. 4 Kinetic energy distribution of first six orders
(a) First-order kinetic energy distribution, (b) Second-order kinetic energy distribution, (c) Third-order kinetic energy distribution, (d) Fourth-order kinetic energy distribution, (e) Fifth-order kinetic energy distribution, (f) Sixth-order kinetic energy distribution

Table 2 First six-order modal effective mass

| Order | \( m_{\text{eff},x} \), N s^2/m | \( m_{\text{eff},y} \), N s^2/m | \( m_{\text{eff},z} \), N s^2/m | \( m_{\text{eff},y} \), N s^2/m | \( m_{\text{eff},z} \), N s^2/m |
|-------|-------------------------------|-------------------------------|-------------------------------|-------------------------------|-------------------------------|
| 1     | \( 0.32 \times 10^{-5} \)     | \( 0.31 \times 10^{-5} \)     | \( 0.14 \times 10^{-3} \)     | \( 0.16 \times 10^{-3} \)     | \( 0.086 \)                   |
| 2     | \( 0.04 \)                     | \( 0.037 \)                   | \( 0.27 \times 10^{-7} \)     | \( 0.25 \times 10^{-7} \)     | \( 0.02 \)                   |
| 3     | \( 0.12 \times 10^{-3} \)     | \( 0.13 \times 10^{-3} \)     | \( 0.15 \times 10^{-2} \)     | \( 0.14 \times 10^{-2} \)     | \( 0.028 \)                  |
| 4     | \( 8.73 \)                     | \( 8.86 \)                    | \( 0.77 \times 10^{-6} \)     | \( 0.77 \times 10^{-6} \)     | \( 0.81 \times 10^{-7} \)    |
| 5     | \( 0.16 \times 10^{-6} \)     | \( 0.16 \times 10^{-6} \)     | \( 0.11 \)                     | \( 0.15 \)                     | \( 2.04 \)                   |
| 6     | \( 0.12 \)                     | 0.14                          | \( 0.99 \times 10^{-4} \)     | \( 0.98 \times 10^{-4} \)     | \( 0.76 \times 10^{-3} \)    |

where \( n \) is the spindle speed; \( z \) the number of teeth; \( c \) the number of channels for collecting signals. The maximum excitation frequency of spindle rotation can be calculated as \( f_{\text{MAX}} \approx 44.3 \text{ Hz} \).

From Table 1, the lowest natural frequency of the bed is known to be higher than the maximum excitation frequency of the machine tool, so it can avoid the resonance of the bed structure during the machine tool work. The frequency range of the first to third order of the bed is 302.82–489.65 Hz, and the vibration pattern shows the whole vibration mode. From the beginning of the fourth order, the frequency is >520.45 Hz. The vibration pattern shows the local vibration pattern at the top of the bed. It needs to optimise the local structure of the bed and turn the local mode into the whole vibration mode.

The first six-order kinetic energy distribution diagram of the lathe bed is obtained through the post-processing module of the finite element software Ansys, as shown in Fig. 4. It can be seen from Fig. 4 that the kinetic energy distribution rate at the bottom of the bed is higher and the quality should be reduced.

4.2 Analysis of modal effective mass

According to the finite element model of the bed, the effective masses of the first six translational directions are calculated through the finite element software Ansys, as shown in Table 2. The \( m_{\text{eff},x} \), \( m_{\text{eff},y} \), and \( m_{\text{eff},z} \) in the table are the effective modal mass of the bed in the direction of \( X \), \( Y \), and \( Z \) respectively, and \( m_{\text{eff},y} \) and \( m_{\text{eff},z} \) are the theoretical modal effective mass of the bed in the direction of \( X \), \( Y \), and \( Z \) respectively.

It is known from Table 2 that the order of the maximum effective mass of the first six translational directions are the fourth order of \( X \)-direction, the fifth order of \( Y \)-direction, and the fifth order of the \( Z \)-direction, which is in agreement with the analysis results of the vibration pattern. In addition, from Table 2, it is known that the error of modal effective mass which is derived from the kinetic energy and derived from the finite element calculation is small, and the result is correct.

Fig. 5 is the cumulative modal effective mass fraction in three directions. As shown in Fig. 5, the cumulative modal effective mass fraction of the three directions of the first six-order modes are different, and the cumulative modal effective mass fraction of the first six orders of \( X \) and \( Z \) reaches 90%, while the effective mass fraction of the cumulative mode in the \( Y \)-direction is small, and more modes are needed to be extracted for subsequent calculation.

5 Optimisation of bed structure

In the dynamic design stage of machine tool, through the modal analysis of the lathe bed, the improvement plan is put forward for the weak part of the structure, and the rationality of the improved scheme is verified.

A lot of square holes are opened in the inside of the bed, which are used to clean sands and reduce the bed weight. Three optimisation programmes of bed square hole structure are put forward in this paper, as shown in Fig. 6.

The modal analysis of three kinds of bed body optimisation schemes is carried out. Fig. 7 is the first four-order natural
frequency distribution diagram of the bed optimisation scheme, and Table 3 is the kinetic energy information table of the bed body optimisation schemes.

It can be seen from Fig. 7 that the natural frequency of the bed structure is significantly improved after optimisation. From Table 3, we can see that the maximum values of kinetic energy of the improved three optimisation schemes have dropped significantly, and the most significant decline in the third scheme. To sum up, adopting the scheme 3 structure is more conducive to improving the dynamic performance of the lathe bed and optimising the structure of the machine tool.

6 Conclusion

i. A formula of modal effective mass based on kinetic energy calculation is derived, and its application in the simulation of the mechanical characteristics of a CNC pipe thread lathe bed is studied. Through the effective mass of the structure, the influence of each order mode on the motion of the rigid body can be judged effectively, and the improved design of the structure can be carried out selectively.

ii. Modal effective mass is a modal characteristic parameter of all system vibration. Based on the modal information obtained from modal analysis, combining the energy distribution and the effective modal parameters to determine whether the number of modes extracted in the simulation process meets the requirements of the simulation precision, and effectively identifies the overall mode and local mode. Therefore, the research in this paper can be widely used, especially the modal identification of more complex structures.
The optimisation scheme of the bed structure is put forward, the natural frequency of each order has been raised, and the weight of the bed has been reduced by 4.8%. Structural optimisation has greatly improved the static and dynamic performance of the bed, and achieved the desired results.

### 7 Acknowledgments
The Project is sponsored by the National Natural Science Foundation (no. 50475170) and Liao ning Bai Qian Wan Talents Program (Liao ning Bai Qian Wan Program [2015]47).

### 8 References

1. Ke, W., Jun, G., Xingwei, S.: ‘Optimal design of bed structure for cyclone screw milling machine based on FEM’, *Mod. Mach. Tool Autom. Manuf. Tech.*, 2016, 8, (8), pp. 134–136

2. Jie, L., Haijun, W., Ke, W., et al.: ‘The dynamic optimal design of bed structure for CNC screw milling machine’, *Mod. Mach. Tool Autom. Manuf. Tech.*, 2011, 46, (3), pp. 83–85

3. Xuegang, Z., Feng, Z., Xiaobo, M., et al.: ‘Research on machine tool dynamic performance optimization based on modal testing’, *Mach. Tool Hydraul.*, 2014, 41, (23), pp. 24–27

4. Clough, R.W., Penzien, J.: ‘Dynamics of structures’ (McGraw-Hill Press, New York City, NY, USA, 1975)

5. Jibao, Q., Zhengping, Z., Bo, H.: ‘A new method for solving dynamic problem of basic excitation structure’. 2007 National Symp. on Structural Dynamics, 2007, pp. 46–60

6. Palanichamy, S.: ‘Modal effective mass for launch vehicles’. AIAA-95-1252-CP, 1995

7. Zhao, L., Zhi-Hai, W.: ‘Random vibration analysis of airborne radar based on optimization of effective modal mass’, *Electromech. Eng.*, 2015, 31, (1), pp. 1–6

8. Guangchen, X., Zeyu, W., Nannan, Z.: ‘The application of finite element technique in machine dynamic design based on energy balance’, *Machinery*, 2015, 53, (2), pp. 8–10

9. Rao, S.S.: ‘Mechanical vibrations’ (Prentice Hall Inc., Upper Saddle River, NJ, USA, 2011, 5th edn.)

10. Chaohui, Z.: ‘Analysis of engineering application examples of ANSYS 12.0 structure analysis’ (China Machine Press, Beijing, China, 2009)

11. Guangzhou, X., Ping, R., Fu, L.: ‘Application of modal effective parameters in mechanics characteristic simulation for satellite payload’, *Comput. Aided Eng.*, 2012, 21, (1), pp. 65–69

12. Lin, S.: ‘Modal mass computation based on ANSYS finite element analysis’, *Missiles Space Veh.*, 2011, 31, (1), pp. 55–57

13. Zhaoxiang, D., Xingquan, L., Jingcheng, Z.: ‘Selection of substructure normal modes based on effective modal mass’, *Aeromet. Eng.*, 2013, 35, (6), pp. 543–547

14. Shi, J.-F., Wu, Q.-W., Guo, W.-C.: ‘Application of modal effective mass for finite element model construction’, *J. Syst. Simul.*, 2013, 25, (5), pp. 995–998

15. Wining, M., Klein, M.: ‘Modal selection by means of effective masses and effective modal forces an application example’. Proc. Conf. on spacecraft Structures, Materials & Mechanical Testing, The Netherlands, March 1996, pp. 751–759