ISSN: 2414-1895 DOI: 10.6919/ICJE.202002_6(2).0016

Simulation research on milling vibration of large thin-wall axisymmetric part

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Abstract

Aiming at the poor rigidity of large thin-wall axisymmetric part and the problems of vibration in milling pro-cess, the mechanical model of large-scale thin-walled revolving part milling is established by using thin-shell theory. The constrained mode of simply supported at both ends of a large-scale thin-walled revolving part with a diameter of 800mm is calculated in this paper. The finite element analysis method is used to analyze the large thin-walled revolving bodies without internal support, six rigid internal supports and surface contact in-ternal support conditions, and their modal analysis and frequency response analysis results are obtained. The results show that the machining rigidity of the surface contact can reach 20294N/mm, which satisfies the pro-cessing rigidity requirement, avoids the severe vibration range of the machine tool-tool, and the amplitude is only 0.0475mm when the excitation is f=620Hz. The above research conclusions provide a theoretical basis for the design of fixtures for surface contact internal support.

Keywords

Large thin-wall axisymmetric part; Milling vibration; Finite element analysis; Auxiliary fixture ; Machining.

1. Introduction

In the aerospace, marine, wind power, petroleum and other equipment manufacturing fields, there are a large number of large thin-wall axisymmetric parts. Large thin-wall axisymmetric parts refer to the parts with space axisymmetric and ratio of thickness and radius less than 1/20. Due to the small wall thickness, during the milling process of such parts, the fluctuation of the cutting parameters cause a sharp change in the cutting force. The non-linear force of the milling cutter and the low rigidity of the part formed a milling cutter-workpiece weak rigid system. The natural frequency range of the machine tool system is about 200Hz ~ 500Hz [1], so the workpiece is easy to cause vibration during the milling process. Severe vibration will cause de-formation of the workpiece, which will lead to reduced geometric accuracy of the part processing and severe tool wear. Therefore, suppressing milling vibration can not only improve production efficiency, but also meet the higher manufacturing requirements of large thin-wall axisymmetric parts.

Many scholars have achieved fruitful results in studying the Suppresses milling vibration of large thin-wall axisymmetric parts. In terms of theoretical research, Songran Dou et al. [2-6] respectively proposed a calculation model for the thin-wall milling process, of which van Dijk verified that the simulation results of the calculation model are very close to the experimental results. Mukai Wang [7] combined the Rayleigh-Litz method to derive the constraint mode calculation formulas of the thin-wall axisymmetric thin-wall axisymmetric body under various boundary conditions and the frequency response calculation formula using the modal superposition method. Zebin Zhou [8]

ISSN: 2414-1895 DOI: 10.6919/ICJE.202002_6(2).0016

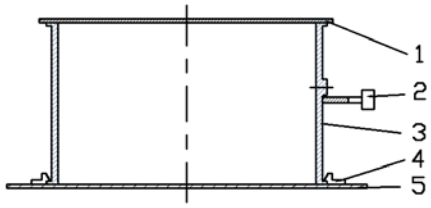
proposed a method for analyzing and determining the topology optimization model of fixture-assisted support dynamics, established a prediction model of the effect of residual stress on machining deformation during cutting, and obtained and verified the effect of cutting parameters on the residual stress distribution. Yu Wang and Zhong Luo [9] carried out a response characteristic analysis of a thin-walled cylindrical shell member subjected to radial harmonic excitation or radial impact excitation under a clamped-free constraint condition under forced vibration. Many scholars have conducted in-depth research on algorithms for actively controlling vibration, and have verified that experiments under the applied algorithm can effectively suppress vibration [10-13]. In terms of process suppression vibration, many scholars [14-16] have established a finite element simulation model of large thin-wall axisymmetric body cutting and milling and carried out numerical simulation to optimize the milling vibration of the casing from the perspective of the process plan. Jundong Shi et al. [17-21] believed that improving the residual stress of the blanks and appropriate cutting path can obtain a more stable milling force, thereby achieving the purpose of suppressing vibration, and verified its feasibility through experiments. In terms of active control and suppression of vibration, adding a vibrator to the workpiece to cancel the vibration in the fixture [22, 23], Liu, K.J. simulation verified that the two-degree-of-freedom structure model suppresses vibration significantly in sheet metal processing. Sallese, L., et al. [24] and Brecher, C. [25] designed an intelligent movable fixture with detection and active control. Tests on processing thin plate parts show that the fixture does effectively suppress vibration. Some researchers have designed a distributed disturbance active vibration control (AVC) system[26, 27]. In terms of passive control to suppress the vibration of thin plates, it is generally used to add passive components (springs and dampers) that can absorb vibrations in the fixture. Traditional fixtures commonly used in engineering applications are passive restraints. Many researchers have optimized designs based on the structure of traditional fixtures [28-30]. Yu Jin and Wang Qiqi studied the influence of the axial position of the support and the initial clamping force on the suppression of milling vibration, and provided data guidance for the design of the fixture. Wang Congmei [31] and Yu Xianyong [32] both designed a thin-walled tooling fixture. Yu Xianyong proposed a method for improving local machining stiffness using an adaptive airbag. Experiments show that this method can effectively reduce the deformation and control of the tool Machining accuracy of thin-walled parts.

Although there are many research results in suppressing milling vibration, it is still a traditional fixture in engineering applications. Large thin-wall axisymmetric parts are of high quality, which is difficult to clamp and disassemble. If the fixture is designed by actively controlling the vibration, the development of the control system and the structure design of the fixture are difficult. So it meets the manufacturing requirements, in the case of passive vibration suppression is the best choice. In order to suppress vibration during milling, this paper analyzes the vibration mechanism during milling of large thin-wall axisymmetric body, and simulates the fixture-workpiece system with different internal support forms. Based on the simulation results, a design scheme of fixture structure to suppress milling vibration of large thin-wall axisymmetric body parts is proposed.

2. Establishment of mechanical models

The machining features of large thin-wall axisymmetric body outer wall are generally realized by milling with a five-axis machining center. In the case of simple and no internal support fixture, the base of the slewing body is fixed to the machining center workbench. As the processing center rotates, the pressure plate presses the flange on the bottom of the slewing body to limit the radial displacement, while relying on friction to lim-it the circumferential displacement. The processing diagram is shown in Fig. 1.

ISSN: 2414-1895 DOI: 10.6919/ICJE.202002_6(2).0016



(1. axial limit plate 2. milling cutter 3. large thin-wall axisymmetric part 4.platen 5.base.)

Fig. 1 Schematic diagram of milling without simple internal support fixture

The interaction force between high-speed rotating milling cutter and large thin-wall axisymmetric body is a dynamic cutting force. Radial and circumferential vibrations will be generated in the force receiving area of the axisymmetric body. In the radial direction, the outer wall is momentarily deformed and displaced to the inside. As the milling cutter follows the trajectory of the cutter, the rigidity of the part will produce an instantaneous restoring force, and it will be displaced to the radial direction for a distance. There-fore, a significant vibration phenomenon is formed in the radial direction. Due to the loss of rigidity in the tangential direction, the instantaneous recovery force is extremely weak, and the phenomenon of tangential vibration is not significant. The effect of the cutting force on the rotating body is shown in Fig. 2.

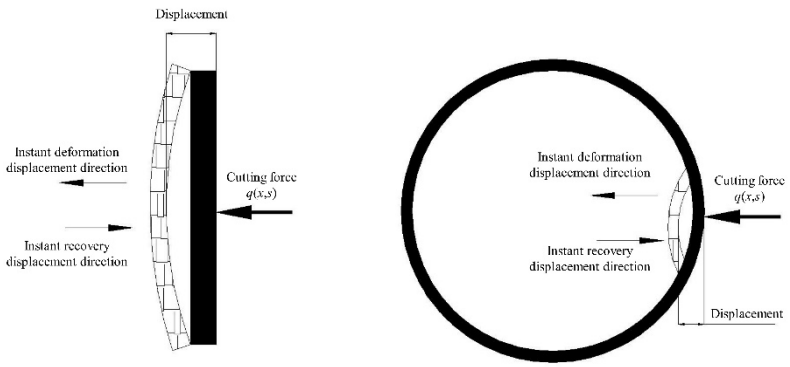


Fig. 2 The effect of cutting force on the part

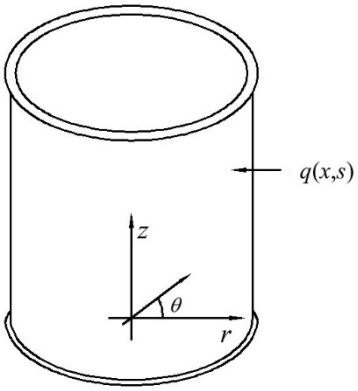


Fig. 3 Large thin-wall axisymmetric part and corresponding coordinate system

Establishing a coordinate system for the large thin-wall axisymmetric body is shown in Fig. 3. When the thickness of the rotating body is less than 1/20 of the radius, the thin shell theory can be used. Its motion equation is as follows:

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$$\begin{cases} R^2\frac{\partial^2 u}{\partial x^2}+\frac{(1-v)}{2}\frac{\partial^2 u}{\partial\theta^2}-\frac{\rho R^2(1-v^2)}{E}\frac{\partial^2 u}{\partial t^2}+\frac{R(1+v)}{2}\frac{\partial^2 v}{\partial x\partial\theta}+Rv\frac{\partial w}{\partial x}+ \\ \left(\frac{(1-v)}{2}\frac{\partial^2 u}{\partial\theta^2}-R^3\frac{\partial^3 w}{\partial x^3}+\frac{R(1-v)}{2}\frac{\partial^3 w}{\partial x\partial\theta^2}\right)=-\frac{(1-v^2)}{Eh}q_x(x,\theta)e^{j\omega t} \\ \frac{R(1+v)}{2}\frac{\partial^2 v}{\partial x\partial\theta}+\frac{R^2(1+v)}{2}\frac{\partial^2 v}{\partial x^2}+\frac{\partial^2 v}{\partial\theta^2}-\frac{\rho R^2(1-v^2)}{E}\frac{\partial^2 v}{\partial t^2}+\frac{\partial w}{\partial\theta}+ \\ \xi\left(\frac{3R^2(1-v)}{2}\frac{\partial^2 v}{\partial x^2}-\frac{3R^2(3-v)}{2}\frac{\partial^3 w}{\partial x^2\partial\theta}\right)=-\frac{1-v^2}{Eh}q_\theta(x,\theta)e^{j\omega t} \\ Rv\frac{\partial u}{\partial x}+\frac{\partial v}{\partial\theta}+w+\xi\nabla^4+\frac{\rho R^2\left(1-v^2\right)}{E}\frac{\partial^2 w}{\partial t^2}+ \\ \xi\left(\frac{R(1-v)}{2}\frac{\partial^3 v}{\partial x\partial\theta^2}-R^3\frac{\partial^3 u}{\partial x^3}-\frac{R^2(3-v)}{2}\frac{\partial^3 v}{\partial x^2\partial\theta}+w+2\frac{\partial^2 w}{\partial\theta^2}\right)=\frac{\left(1-v^2\right)}{Eh}q_r(x,\theta)e^{j\omega t} \end{cases} \tag{1}$$

Where R is the radius of the large thin-walled axisymmetric body; h is the thickness of the slewing body; E is Young's modulus; v is Poisson's ratio; ρ is the density of the axisymmetric body; $\xi=h^2/(12R^2)$; $\nabla^4=\nabla^2\nabla^2$ is the modified Laplace calculation; $\nabla^2=R^2\frac{\partial}{\partial x}+\frac{\partial}{\partial\theta}$; $u(x,\theta,t)$, $v(x,\theta,t)$ and $w(x,\theta,t)$ are respectively displacements in the axial, tangential, and radial directions.

The restraint of the two ends of the axisymmetric body is simply supported, and its boundary condition is:

$$\begin{cases} u_0=u_L=0 \\ v_0=v_L=0 \\ w_0=w_L=0 \\ M_x=0 \end{cases} \tag{2}$$

Where M_x is the moment in the x plane, which is defined as:

$$M_x=\frac{Eh^3}{12(1-v^2)}\left(-\frac{\partial^2 w}{\partial x^2}+\frac{v}{R^2}\frac{\partial v}{\partial\theta^2}-\frac{v}{R^2}\frac{\partial^2 w}{\partial\theta^2}+\frac{1}{R}\frac{\partial u}{\partial x}\right) \tag{3}$$

The frequency determinant is:

$$\begin{vmatrix} \Omega^2-H_1 & \frac{n\lambda}{2}(1+v) & v\lambda \\ \frac{n\lambda}{2}(1+v) & \Omega^2-H_2 & n \\ -v\lambda & n & \Omega^2-H_3 \end{vmatrix} \tag{4}$$

Among them, $\lambda=m\pi R/L$; n is the number of circumferential modes; m is the number of axial modes; the frequency parameter Ω is $\Omega=\omega\left(\frac{\rho\left(1-v^2\right)R^2}{E}\right)^{\frac{1}{2}}$; $H_1=\lambda^2+\frac{h^2}{2}(1-v)$; $H_2=\frac{\lambda^2}{2}(1-v)+n^2$; $H_3=1$.

The following is a constraint modal calculation for a 600mm × 800mm × 10mm large thin-walled axisymmetric body with simple support at both ends (such as the parameters in Table 1). The calculation results are shown in Table 2. The results show that the natural frequency in the middle of the slewing body is smaller than the two ends. The natural frequency of the slewing body is in the range of 120 Hz to 800 Hz, and the lowest frequency is 130 Hz.

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Table 1 Parameters of large thin-wall axisymmetric part

Parameter	Value
Length/mm	600
Diameter/mm	800
Thickness/mm	10
Young's modulus/GPa	199.9
Poisson's ratio	0.3
Density/(g/cm3)	8.24

Table 2 Natural frequency of the large thin-wall axisymmetric part with simple support at both ends

Natural frequency/Hz		Number of axial modes				
		1	2	3	4	5
Number of circular modes	0	441.8	502.1	561.4	712.5	817.1
	1	310.4	424.1	425.9	536.7	715.9
	2	153.3	275.9	286.6	363.0	594.0
	3	131.4	173.2	192.5	249.5	305.5
	4	167.7	364.6	319.3	330.8	321.4
	5	276.3	363.0	385.5	432.0	439.4
	6	356.3	460.3	482.0	517.5	613.3
	7	460.6	521.8	537.8	590.0	646.5
	8	497.7	540.5	617.4	680.2	707.8

Because the excitation frequency of the machine-tool system is often between 200 and 500 Hz, the modal frequency being solved is in the region of severe vibration, so severe vibration occurs during processing. In order to suppress milling vibration, a more economical passive vibration suppression scheme is adopted to carry out modal analysis and frequency response analysis of large thin-walled axisymmetric body without internal support, internal six-point rigid internal support, and surface contact internal support. Analyze the optimal clamping scheme based on numerical simulation results.

3. Finite element simulation analysis

3.1 Modal analysis

The large thin-walled axisymmetric body is a cylindrical shell structure. A finite element model as shown in Fig. 4 is established, and a boss feature to be milled is set on the outer peripheral wall. The material properties of the model are the same as those in Table 1. After the geometric model is imported into Nastran for geometric cleaning and finite element meshing and calculation, the tetrahedral mesh is used for meshing. The entire large thin-walled axisymmetric body is divided into 206789 nodes and 102670 units. Using Nastran software to carry out free modal analysis on the previously built model, the natural frequencies of each order and the corresponding mode shapes are obtained. Low frequency modes are mainly avoided, and only the first 10 order vibration modes are calculated. In the free modal analysis, the solid element has 6 rigid body degrees of freedom, so the natural frequency of the first 6 orders of rigid body modes is about 0 Hz. The simulation results are

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shown in Fig. 5 and the natural frequency values are shown in Table 3 (the first 6 orders are omitted). The results show that large thin-walled axisymmetric body will deform and bend at 40 ~ 60Hz.

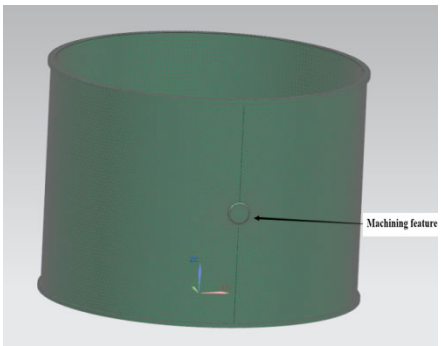

Fig. 4 The effect of cutting force on the part

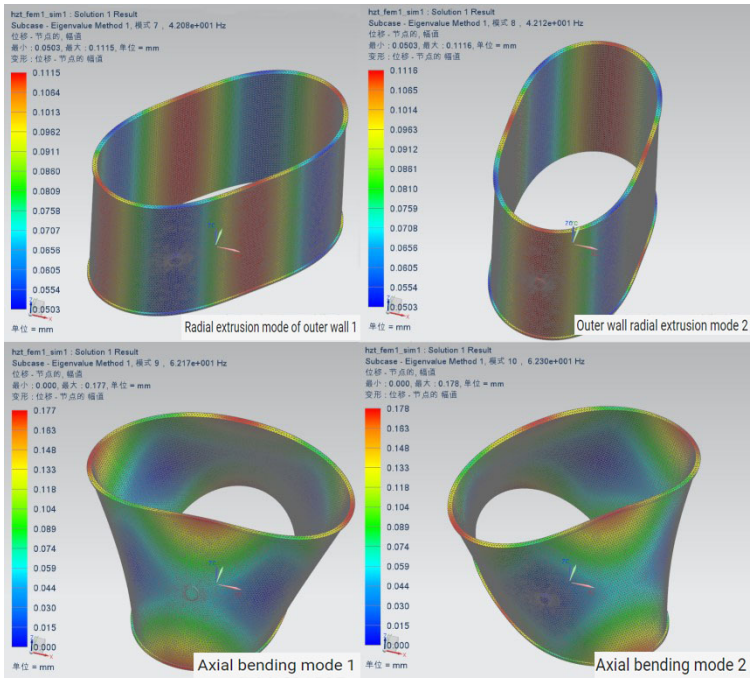

Fig. 5 Free mode partial mode

Table 3 Description of natural frequency and mode shape of large thin-wall axisymmetric part

Order	Simulation frequency/Hz	Mode description
1	42.08	Radial extrusion mode of outer wall 1
2	42.12	Radial extrusion mode of outer wall 2
3	62.17	Axial bending mode 1
4	62.3	Axial bending mode 2

In order to test the inherent performance of three structural forms without internal support, six rigid internal support, and surface contact internal support, a constraint modal analysis was performed on the three structural forms. In actual milling processing, the axial displacement and circumferential rotation of large thin-walled axisymmetric bodies are restricted, so fixed constraints are imposed on both end faces. In addition, the support of the six rigid internal support and the surface contact internal support contacted the inner wall. The simulation object attributes were set to face-to-face contact, and fixed constraints were imposed on the inside of the support. The simulation conditions are shown in Table 4.

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Table 4 Simulation condition

Type	Constraint	Simulation object
Simple without internal support	Fixed constraints on both ends of the rotating body	No
Six rigid inner supports	Fixed constraints on both ends of the rotating body	Face to face
	Fixed inside restraint	
Face contact inner support	Fixed constraints on both ends of the axisymmetric body	Face to face
	Fixed inside restraint	

The results of the constrained modal analysis of the three structural forms are shown in Fig. 6. The mode shapes are mainly the axial tensile deformation and the deformation of the cylindrical wall along the radial direction and the bending deformation along the flange. The large thin-walled axisymmetric body account for 100%, 46.33%, and 20.33% of the natural frequency resonance regions of the parts process system without internal support, six rigid supports, and surface contact internal supports. The large thin-walled axisymmetric body is completely in the severe vibration zone of the machine tool system without internal support, so it produces severe vibration. With the rigid support of the radial support increased significantly under the six rigid supports, the first-order natural frequency reaches 361.3Hz, and the performance is relatively improved by 104.24%, but it still does not avoid the severe vibration zone of the machine tool system. The first-order natural frequency of the surface contact internal support structure is 423 Hz, which is 232.58% higher than that without internal support. Theoretically, it can significantly suppress milling vibration.

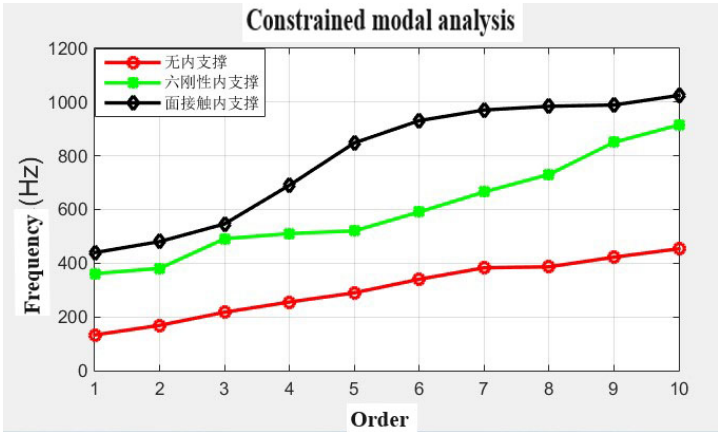

Fig. 6 Constrained modal analysis results

3.2 Frequency response analysis

Through constrained modal analysis, the relative unsupported performance of the six-rigid internal support and the surface-contact internal support has been greatly improved, but there is not much difference in the natural frequencies of the first few steps, so the three forms of dynamics need to be analyzed by frequency response. performance. Milling a large thin-walled axisymmetric body's outer wall at the upper end of the machining center to obtain machining features, the formula for calculating the milling force is:

$$F = 1150 \cdot t^{1.08} \cdot S_z^{0.88} \cdot D^{-1.8} \cdot B^{0.90} \cdot n^{-0.18} \cdot Z \cdot g \tag{5}$$

Take t=200mm, B=1mm, D=300mm, Z=12, n=500r/min, S_z=0.1mm/tooth, g=9.8N/m^2. Get F=964N. Simplify the cutting action of milling force into dynamic excitation, take $F_x = F_y = F_z = 557\text{N}$. Therefore, in the simulation, the excitation amplitude is 557N, and the directions are the radial,

ISSN: 2414-1895 DOI: 10.6919/ICJE.202002_6(2).0016

tangential, and vertical directions of the excitation point relative to the cylinder, respectively. The frequency is f =100~700Hz and the step is 20Hz.

The frequency response analysis results are shown in Fig. 7 (only the state diagram with the largest displacement amplitude is taken). The results show that when the large thin-walled axisymmetric body is unsupported, when the excitation frequency f=260Hz, the displacement peak is 0.466mm, and the vibration phenomenon is very obvious. The six-rigid internal support can greatly reduce the amplitude of vibration. When the excitation is on the opposite side of the support, the displacement amplitude is as low as 0.167mm, but the excitation is as high as 0.306mm at the midpoint between the two adjacent supports. When the amplitude of the form displacement of the large-scale thin-walled rotating body supported by the surface is f=620Hz, the amplitude is only 0.0475mm.

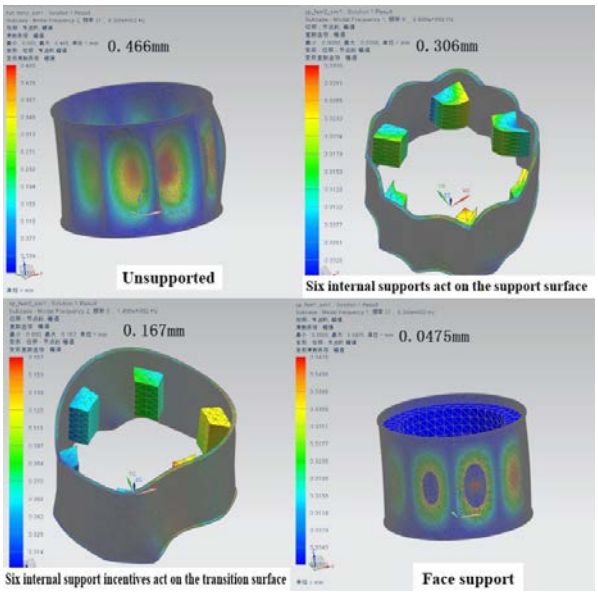

Fig. 7 Frequency response partial mode

The results of Table 5 are obtained by processing the simulation data. The results show that with the increase of support surface, the processing rigidity of large thin-walled slewing bodies has increased significantly. The overall rigidity of the six-rigid internal support has increased by 51.95 ~ 178.43% relative to the no-supported rigidity 8 times as much. Next, the consistency of the surface processing quality of the three structural forms is analyzed. The simple unsupported and the surface contact internal support are spatial axisymmetric forms. The consistency during the ring milling process is stable, so only the six rigid internal supports are analyzed. Twelve equally spaced sample points are taken between two adjacent supports of the six rigid inner supports, as shown Fig. 8. Frequency response analysis was performed for each sample point. The displacement amplitude of the i sample point was x_i μm. The simulation results are shown in Fig. 9. The simulation results show that the smaller the magnitude of the displacement of the sample points near the support, the larger the magnitude of the displacement, and the difference of the magnitude of the displacement reaches 163.27 μm, which results in poor consistency of surface processing quality.

Table 5 Average machining stiffness of different structural forms

Structure type	Without internal support	Six internal supports are excited on the transition surface	Six internal supports are excited on the support surface	Face contact inner support
resultant force F/N	964	964	964	964
Displacement amplitude / μm	465	306	167	47.5

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Average processing stiffness / (N / mm)	2073.118	3150.327	5772.455	20294.74
Relative improvement	-	51.95%	178.43%	878.97%

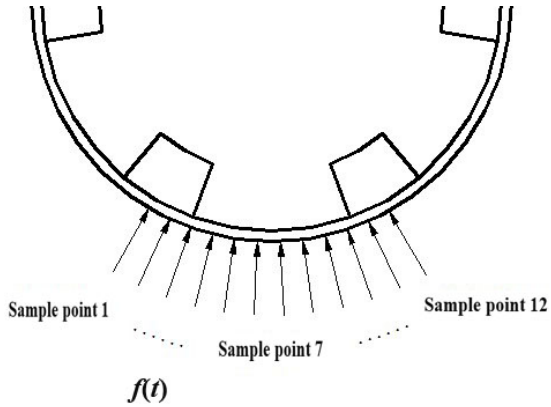

Fig. 8 Six schematic diagram of selection of adjacent sample points for rigid inner support

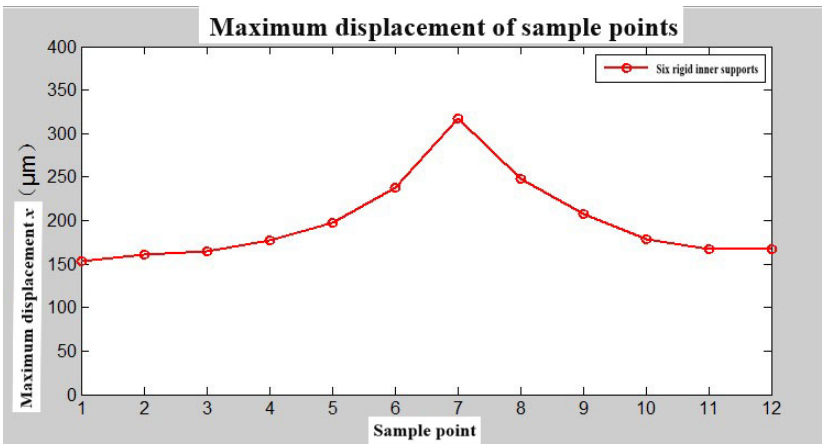

Fig. 9 The amplitudes of amplitudes excited by 12 equal interval sample points

Comprehensive analysis shows that large thin-walled axisymmetric body without internal support have poor machining rigidity due to the thin-walled structural characteristics, resulting in large displacement and deformation amplitudes during the milling process, very severe vibration, and even severe deformation. Does not meet processing requirements. The use of six-rigid internal support will greatly improve the processing axisymmetric of the outer wall of the rotating body, which can meet the processing requirements of the surface of the part that is not high, but it does not completely avoid the severe vibration zone of the machine tool system. Problems with poor quality consistency. The surface contact internal support can solve the problems of insufficient rigidity of the rotating body and the vibration interval of the machine tool, and provide the possibility for higher manufacturing requirements.

4. Conclusion

According to the theory of thin shell, a milling vibration mechanical model of the large thin-walled axisymmetric body was established, and the constraint mode of a slewing body with a diameter of 800 mm was simply calculated at both ends. The results show that its natural frequency is between 120~800 Hz, The severe vibration range of the machine-tool system, the swing body

ISSN: 2414-1895 DOI: 10.6919/ICJE.202002_6(2).0016

vibrates violently without support. Based on the above theoretical model, using Nastran software, a finite element analysis method for milling vibration of large thin-walled slewing bodies was constructed. Based on the slewing body with a diameter of 800 mm, three types of internal support, six rigid internal supports, and surface contact internal supports were analyzed. Modal and frequency response in the form. Based on the simulation analysis, the following conclusions are reached:

1) According to theoretical calculations and simulation results, milling vibration of large thin-walled axisymmetric parts without internal support is obvious.

2) Large thin-walled axisymmetric body adopts six rigid internal supports, which can avoid most of the severe vibration zone. The processing rigidity performance is at least 51.95% higher than that without support.

3) The surface contact internal support can effectively avoid the severe vibration interval, and its displacement amplitude is only 47.5 μm through frequency response analysis. The surface contact internal support structure inhibits milling vibration significantly.

The above studies show that the use of surface contact internal support can effectively suppress milling vibration during the milling process of large thin-walled axisymmetric body. The next step is to design a new type of fixture with surface contact and internal support to suppress milling vibration of such parts.

Acknowledgements

Young Science and Technology Innovation Team of Aero Engine Intelligent Manufacturing, SWPU(No. 2019CXTD02).

Science and Technology Achievement Transformation and Guiding Plan of Sichuan Province(No. 18CGSF0057).

Sichuan science and technology project, China (No.2018JY0366).

References

[1] Rivin E I, Agapiou J, Contributor, et al. Tooling structure: interface between cutting edge and machine tool, CIRP Annals - Manufacturing Technology, vol. 2 (2000), 591-634.

[2] S.R. Dou, H.B. Gui, C.H. Li, et al. Constrained damping topology optimization for vibration control of cylin-drical shells, Journal of Vibration and Shock, vol. 22 (2015), 149-153. (In Chinese)

[3] Tsai J, Liao C. Finite-element modeling of static surface errors in the peripheral milling of thin-walled workpieces, Journal of Materials Processing Technology, Vol. 94 (1999)No.2-3, p.235-246.

[4] van Dijk N J M, van de Wouw N, Doppenberg E J J, et al. Robust activechatter control in the high-speed milling process, IEEE Transactions on Control Systems Technology, Vol. 20(2012)No. 4, p. 901-917.

[5] W.L. Mao. Study on vibration and deformation of thin-walled parts in high speed milling, Lanzhou: Lanzhou University of Technology, 2011. (In Chinese)

[6] D.Z. Wen, Y.S. Chen. Vibration analysis of aero-engine casing under radial harmonic excitation, Journal of Aer-ospace Power, Vol. 30(2015)No. 1: 96-105. (In Chinese)

[7] M.K. Wang, Z.B. Chen, Y.H. Jiao， et al. Vibration characteristics of thin walled cylindrical shells with constrained damping, Journal of Harbin Institute of Technology, vol. 1(2017), 72-79 (In Chinese)

[8] Z.B. Zhou. Deformation prediction and fixture layoout optimization of thin walled workpiece, Shanghai: Donghua University, 2011. (In Chinese)

[9] Y. Wang, Z. Luo. Forced vibration response characteristics of thin cylindrical shell, Journal of Vibration and Shock, Vol. 7(2015), 103-108. (In Chinese)

[10] Rashid A, Nicolescu C M. Active vibration control in palletised workholding system for milling, International Journal of Machine Tools & Manufacture: Design, research and application, Vol. 12-13(2006), 1026-1636.

[11] Ratchev S, Liu S, Huang W, et al. An advanced FEA based force induced error compensation strategy in milling, International Journal of Machine Tools and Manufacture, Vol. 46(2006)No. 5: 542-551.

ISSN: 2414-1895 DOI: 10.6919/ICJE.202002_6(2).0016

[12] J. Zhai. Research on predictive control for vibrations of thin-walled workpieces in milling, Hubei: Huazhong University of Science and Technology, 2014. (In Chinese)

[13] P.W. Chen. Research on predictive control for vibrations of thin-walled workpieces in milling, Hubei: Huazhong University of Science and Technology, 2014. (In Chinese)

[14] Z.G. Wu, L.B. Liu, J.F. Sun, et al. Study on controlling machining distortion of aeroengine TC4 casing, Aeronautical Manufacturing Technology, vol. 21(2017) 62-67,72. (In Chinese)

[15] J.X. Ren, J. Yang, J.H. Zhou, et al. Method of controlling deformation of aeroengine casing in NC machining, Aeronautical Manufacturing Technology, vol. 23(2014) 86-89. (In Chinese)

[16] H.Q. Wang, Y. Zhang, D.Y. Wang. Simulation analysis and control of NC machining deformation for large thin-walled aluminum alloy shell (Part 2), Metal Working, vol. 24(2010) 31-33. (In Chinese)

[17] J.D. Shi. Research on the machining process of the superalloy thin casing parts for aero-engine, Dalian: Dalian University of Technology, 2012. (In Chinese)

[18] W.C. Zhou. Research on machining distortion of thin-walled rotary part, Nanjing: Nanjing University of Aeronautics and Astronautics, 2015. (In Chinese)

[19] Masoudi S, Amirian G, Saeedi E, et al. The effect of quench-induced residual stresses on the distortion of machined thin-walled parts, Journal of Materials Engineering and Performance, vol. 10(2015) 3933-3941.

[20] Lin Z, Lai W, Lin H, et al. The study of ultra-precision machining and residual stress for NiP alloy with different cutting speeds and depth of cut, Journal of Materials Processing Technology, vol. 1(2000) 200-210.

[21] D. Liu, W.Y. Chen. Simulation of thin wall integral structure distortion during machining, JOURNAL OF SYSTEM SIMULATION, vol. 6(2008) 1589-1593. (In Chinese)

[22] P.J. Ma, W. Xu, Y. Gao， et al. Research on chatter suppression in thin-walled casing milling, Aeronautical Manufacturing Technology, vol. 11(2018) 16-23. (In Chinese)

[23] Liu K J, Rouch K E. Optimal passive vibration control of cutting process stability in milling, Journal of Materials Processing Technology, vol. 1-2(1991) 285-294.

[24] Sallese L, Innocenti G, Grossi N, et al. Mitigation of chatter instabilities in milling using an active fixture with a novel control strategy, The International Journal of Advanced Manufacturing Technology, vol. 9-12(2017) 2771-2787.

[25] Brecher C, Manoharan D, Ladra U, et al. Chatter suppression with an active workpiece holder, Production Engineering, vol. 2-3(2010) 139-245.

[26] Y.H. Xie, J.F. Yin, J.H. Wen. Design of vibro-acoustic control of the cylindrical shell via a dynamic vibration absorber annulus, Noise and Vibration Control, vol. z1(2018) 264-268.(In Chinese)

[27] Julai, S., Tokhi, et al. Active vibration control of flexible plate structures with distributed disturbances, Journal of Low Frequency Noise, Vibration and Active control, vol. 2(2012) 123-150.

[28] J.T. Wang. Research on machining deformation of large-scale cylinder, Shenyang: Shenyang Aerospace University, 2016.(In Chinese)

[29] W. Zhang. Simulation study on deformation of low pressure thin-walled casing with residual stress, Dalian: Dalian University of Technology, 2017. (In Chinese)

[30] J. Yu, Y.Q. Wang. Simulation research on casing auxiliary support based on accurate milling force modeling, Modular Machine Tool & Automatic Manufacturing Technique, vol. 3(2018) 30-33.(In Chinese)

[31] C.M. Wang. Technological measures for controlling the deformation of large size casing parts, Aeronautical Manufacturing Technology, vol. 22(2012) 96-98.

[32] X.Y. Yu. Optimization of aviation thin-walled parts tooling and it's adaptive research, Shenyang: Northeastern University, 2015. (In Chinese)