Research on Unbalanced Vibration Characteristics of Spindle with Disk Workpiece

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Abstract

Disk-type components are common functional parts in rotating mechanisms and usually require corresponding dynamic balancing testing and calibration before assembly and debugging to meet certain balance accuracy levels. Traditional methods for detecting unbalance in disk-type workpieces are mainly conducted on dynamic balancing machines, which have issues such as high testing costs and low testing efficiency. Therefore, it is of great significance to study online calculation methods for workpiece unbalance. This paper takes the basic structure of the spindle system of a CNC machine tool as a prototype and establishes an integrated dynamic model of the workpiece and spindle based on the structural characteristics of the disk-type workpiece. A system simulation program is developed using software to study the vibration response of the unbalanced spindle, the coupled unbalanced vibration response between the workpiece and the spindle, and analyze the laws of the influence coefficients between the workpiece and the spindle system, as well as the impact of workpiece unbalance mass on the unbalanced vibration of the spindle. This provides a theoretical basis for studying online calculation methods for unbalance in disk-type workpieces and technical support for solving issues such as high testing costs and low efficiency in the detection of disk-type workpieces.

Keywords

Disc workpieces; Unbalanced vectors; Spindle; Modelling simulations.

1. INTRODUCTION

Disk-type components are common parts in rotating mechanisms, and these components have a large ratio of diameter to thickness. Even a small unbalance mass can result in significant unbalance, which can have a significant impact on the overall balance performance of the rotor system. Therefore, most disk-type components require specialized balance performance testing and unbalance correction before assembly and debugging, in order to meet certain balance accuracy levels. Currently, commonly used dynamic balancing methods include process balancing, assembly balancing, and whole machine balancing. These methods are all post-balance methods for the workpiece, requiring specialized balance equipment with limited versatility. The balance cost is high, and the involvement of professional testing personnel increases the production and balance costs of the components and the entire machine. The complex balance process and low efficiency limit the production efficiency of disk-type components[1]. Therefore, it is of great significance to research online detection of unbalance during the machining process of disk-type components to address the issues of high balance cost and low efficiency. J.H. Koo et al. used the finite element method and transfer matrix method to guide the influence coefficient method for counterweighting[2].X.L. Yun et al.

ISSN: 2472-3703 DOI: 10.6911/WSRJ.202311_9(11).0004

investigated the relationship between dynamic influence coefficients and the system model by analysing the relationship between the unbalance excitation and the spindle unbalance response[3].R.A.F. Diaz et al. developed a numerical model of a turbine generator by using modal analysis, which enables the balancing weight and its angular positioning to be a priori evaluation[4]. Yun Zhang et al. constructed a spindle dynamics model to analyse the unbalanced vibration characteristics of a high-speed spindle and achieved the suppression of unbalanced vibration without test weight through simulation experiments[5].

The online measurement of workpiece unbalance is based on an understanding of the characteristics of spindle unbalance vibration[6]. In this paper, a dynamic model of the workpiece-spindle system was established using a typical disc workpiece processed on a CNC machine tool as an example. Based on this model, simulation analysis was conducted to study the response of the spindle to unbalance vibration, as well as the relationship between the unbalance vibration response of the workpiece and the spindle coupling. The feasibility of the online measurement method for workpiece unbalance was verified.

2. MODELLING OF THE DISC WORKPIECE-SPINDLE SYSTEM

Typically, disk-workpieces have a large diameter-to-thickness ratio, allowing the unbalance to be equivalent within the same plane. The workpiece is mounted on the machine tool chuck. Therefore, when modeling, the machine tool system can be simplified into four parts: the spindle, bearings, chuck, and disk-like workpiece for analysis. Based on the research on the basic structural characteristics and mechanical boundary conditions of the CNC machine tool spindle system[7], the dynamic characteristics of unbalanced disk-like workpieces are incorporated to establish a complex rotor system dynamic model that couples the workpiece and spindle system. Based on model analysis, this paper studies the coupling excitation forms and response characteristics between workpiece imbalance and spindle imbalance, and analyzes the influence of workpiece imbalance on the vibration response of spindle system imbalance. The influence coefficient model between workpiece imbalance excitation and measuring point response and its inverse problem mathematical model are established to investigate the relationship between spindle dynamic response and workpiece imbalance quantity and spindle system parameters. The research on the dynamic characteristics of disc workpiece-spindle system has important significance for disc workpiece-spindle system dynamics.

When modeling based on a certain CNC machine tool prototype, in order to facilitate the construction of the system dynamics model, the CNC machine tool is simplified and analyzed to reduce the impact of minor factors on the overall analysis. Taking into account factors such as the spindle system structure and its dynamics parameters, the support system and dynamics parameters, etc., the physical model of the workpiece-spindle-bearing system is established as shown in Figure 1.

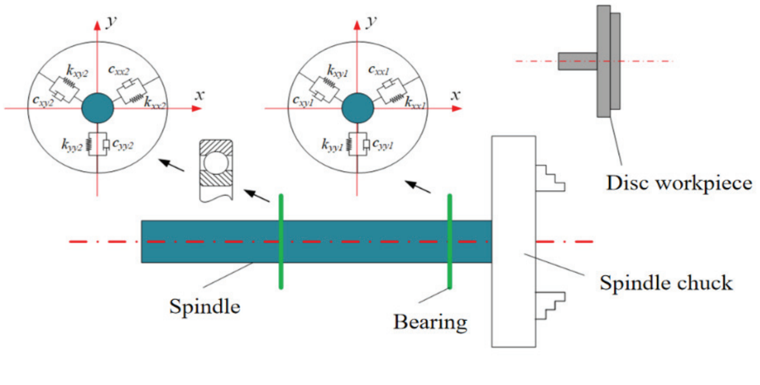

Figure 1. Physical model of workpiece-spindle-bearing system

ISSN: 2472-3703 DOI: 10.6911/WSRJ.202311_9(11).0004

In Figure 1, the workpiece-spindle-bearing system is simplified into four parts: the spindle, the bearings, the spindle chuck and the disc workpiece. The front and rear support bearings are equivalent to an anisotropic damping mechanism.

Based on the above analysis, the mathematical model of the system can be further constructed. During the dynamic balancing process, the deformation of the workpiece-spindle-bearing system caused by gravity and imbalance is relatively small, so the deformation of the overall structural characteristics of the rotor system is also very small, with only minor deformation that can be ignored[8]. Therefore, under this premise, the study of dynamic balancing of the workpiece-spindle-bearing system meets the following two conditions: First, the dynamic balancing performance of the workpiece-spindle-bearing system is independent of its rotational speed; Second, at the balanced rotational speed, there will be a small amount of residual imbalance, and for any other rotational speed, the imbalance will not exceed the maximum imbalance[9]. Therefore, the mathematical model of the workpiece-spindle-bearing system can be established as shown in Figure 2.

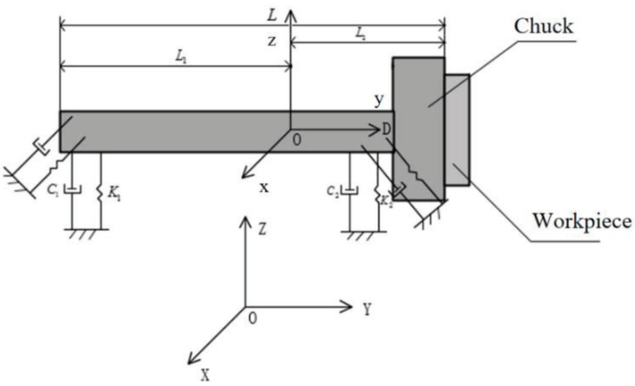

Figure 2. Mathematical model of workpiece-spindle-bearing system

In Figure 2, there are two coordinate systems: OXYZ is the fixed coordinate system of the system, oxyz is the coordinate system rotating synchronously with the spindle. In the oxyz coordinate system, point o is located at the centroid of the spindle. Initially, the fixed coordinate system coincides with the rotating coordinate system. The stiffness and damping of the rear bearing are equivalent to the X and Z directions, represented by K_1 and C_1 respectively; the stiffness and damping of the front bearing are equivalent to the X and Z directions, represented by K_2 and C_2 respectively; the total mass of the spindle, chuck and workpiece is set as m. The unbalanced mass of the spindle is set as mu, and its coordinates relative to the rotating coordinate system are (ux,uy,uz) respectively. The moments of inertia of the spindle centroid rotating around the x, y, and z axes are Jx, Jy, and Jz respectively. Under general conditions, the axial displacement of the spindle is very small and is often negligible. Based on the above assumptions and combined with dynamics knowledge, the dynamic equations are established as follows:

$$\begin{cases} m\ddot{X}+c_1\dot{X}_1+c_2\dot{X}_2+k_1X_1+k_2X_2=m_u\omega^2\left(u_x\cos\omega t+u_z\sin\omega t\right) \\ m\ddot{Z}+c_1\dot{Z}_1+c_2\dot{Z}_2+k_1Z_1+k_2Z_2=m_u\omega^2\left(u_z\cos\omega t-u_x\sin\omega t\right) \\ J_x\ddot{\theta}-c_1\dot{X}_1l_1+c_2\dot{X}_2l_2-k_1X_1l_1+k_2X_2l_2-J_y\omega\dot{\varphi} \\ =-m_u\omega^2\left(u_z\cos\omega t-u_x\sin\omega t\right)u_y \\ J_x\ddot{\varphi}-c_1\dot{Z}_1l_1+c_2\dot{Z}_2l_2-k_1Z_1l_1+k_2Z_2l_2-J_y\omega\dot{\theta} \\ =m_u\omega^2\left(u_x\cos\omega t-u_z\sin\omega t\right)u_y \end{cases} \tag{1}$$

ISSN: 2472-3703 DOI: 10.6911/WSRJ.202311_9(11).0004

Under small amplitude vibration conditions, the following is satisfied:

$$\begin{aligned} X_1 &= X - l_1 \sin\theta \approx X - l_1\theta \\ X_2 &= X + l_2 \sin\theta \approx X + l_2\theta \\ Z_1 &= Z - l_1 \sin\varphi \approx Z - l_1\varphi \\ Z_2 &= Z + l_2 \sin\varphi \approx Z + l_2\varphi \end{aligned} \tag{2}$$

$l_1 + l_2 = l$ in Equation (1), so equation (1) can be further transformed into:

$$\begin{cases} m\ddot{X} + (c_1 + c_2)\dot{X} + (k_1 + k_2)X - (c_1 l_1 - c_2 l_2)\dot{\theta} - (k_1 l_1 - k_2 l_2)\theta \\ = m_u \omega^2 (u_x \cos\omega t + u_z \sin\omega t) \\ m\ddot{Z} + (c_1 + c_2)\dot{Z} + (k_1 + k_2)Z - (c_1 l_1 - c_2 l_2)\dot{\varphi} - (k_1 l_1 - k_2 l_2)\varphi \\ = m_u \omega^2 (u_z \cos\omega t - u_x \sin\omega t) \\ J_x \ddot{\theta} + (c_2 l_2 - c_1 l_1)\dot{X} + (k_2 l_2 - k_1 l_1)X + \left(c_1 l_1^{\,2} + c_2 l_2^{\,2}\right)\dot{\theta} + \left(k_1 l_1^{\,2} + k_2 l_2^{\,2}\right)\varphi - J_y \omega \dot{\varphi} \\ = -m_u \omega^2 (u_z \cos\omega t - u_x \sin\omega t) u_y \\ J_x \ddot{\varphi} + (c_2 l_2 - c_1 l_1)\dot{Z} + (k_2 l_2 - k_1 l_1)Z + \left(c_1 l_1^{\,2} + c_2 l_2^{\,2}\right)\dot{\varphi} + \left(k_1 l_1^{\,2} + k_2 l_2^{\,2}\right)\varphi + J_y \omega \dot{\theta} \\ = m_u \omega^2 (u_x \cos\omega t + u_z \sin\omega t) u_y \end{cases} \tag{3}$$

Equation (3) only considers the excitation factor of spindle imbalance. Here, it is further assumed that the workpiece imbalance mass is mw, and the coordinates of the workpiece imbalance mass relative to the spindle centroid are (ux1, uy1, uz1). Then equation (3) can be further transformed into:

$$\begin{cases} m\ddot{X} + (c_1 + c_2)\dot{X} + (k_1 + k_2)X - (c_1 l_1 - c_2 l_2)\dot{\theta} - (k_1 l_1 - k_2 l_2) \\ = m_u \omega^2 (u_x \cos\omega t + u_z \sin\omega t) + m_1 \omega^2 (u_{x1} \cos\omega t + u_{z1} \sin\omega t) \\ + m_2 \omega^2 (u_{x2} \cos\omega t + u_{z2} \sin\omega t) \\ m\ddot{Z} + (c_1 + c_2)\dot{Z} + (k_1 + k_2)Z - (c_1 l_1 - c_2 l_2)\dot{\varphi} - (k_1 l_1 - k_2 l_2) \\ = m_u \omega^2 (u_z \cos\omega t - u_x \sin\omega t) + m_1 \omega^2 (u_{z1} \cos\omega t - u_{x1} \sin\omega t) \\ + m_2 \omega^2 (u_{x2} \cos\omega t - u_{z2} \sin\omega t) \\ J_x \ddot{\theta} + \left(c_1 l_1^{\,2} + c_2 l_2^{\,2}\right)\dot{\theta} + \left(k_1 l_1^{\,2} + k_2 l_2^{\,2}\right)\theta + (c_2 l_2 - c_1 l_1)\dot{X} + (k_2 l_2 - k_1 l_1)X - J_y \omega \dot{\varphi} \\ = -m_u \omega^2 (u_z \cos\omega t - u_x \sin\omega t) u_y - m_1 \omega^2 (u_{z1} \cos\omega t - u_{x1} \sin\omega t) u_{y1} \\ - m_2 \omega^2 (u_{z2} \cos\omega t - u_{x2} \sin\omega t) u_{y2} \\ J_z \ddot{\varphi} + \left(c_1 l_1^{\,2} + c_2 l_2^{\,2}\right)\dot{\varphi} + \left(k_1 l_1^{\,2} + k_2 l_2^{\,2}\right)\varphi + (c_2 l_2 - c_1 l_1)\dot{Z} + (k_2 l_2 - k_1 l_1)Z - J_y \omega \dot{\theta} \\ = m_u \omega^2 (u_x \cos\omega t + u_z \sin\omega t) u_y + m_1 \omega^2 (u_{x1} \cos\omega t + u_{z1} \sin\omega t) u_{y1} \\ + m_2 \omega^2 (u_{x2} \cos\omega t + u_{z2} \sin\omega t) u_{y2} \end{cases} \tag{4}$$

The first term on the right side of equation (4) is the excitation of the spindle system imbalance, the second term is the excitation of the workpiece imbalance, and the third term is the excitation of the simulated correction imbalance. When the second and third terms on the right side of equation (4) are zero, the excitation and vibration response equations of the spindle system imbalance can be simulated. When only the third term on the right side of

ISSN: 2472-3703 DOI: 10.6911/WSRJ.202311_9(11).0004

equation (4) is zero, the coupling between the spindle imbalance and workpiece imbalance can be simulated to examine the influence of workpiece imbalance on spindle system imbalance and establish the relationship between workpiece imbalance and vibration response. When all three terms on the right side of equation (4) are nonzero, the online correction of workpiece imbalance can be simulated.

3. SIMULATION ANALYSIS OF THE DISC WORKPIECE-SPINDLE SYSTEM

The online measurement of disc workpiece unbalance is based on a deep understanding of the unbalance law of the spindle system, and dynamic balancing technology is also used in the study of the workpiece-spindle-bearing system. By establishing the mathematical model of workpiece-spindle-bearing system and compiling the simulation system program, we can study the unbalanced vibration characteristics of the spindle and the unbalanced coupling response of the workpiece-spindle, which is of great significance to the calculation of workpiece unbalance.

The simulation steps of the dynamic characteristics of the workpiece-spindle-bearing system are mainly divided into the following aspects: firstly, based on the structure of the spindle system of the CNC lathe, the appropriate spindle system and simulation parameters of the simulated workpiece are selected, and the parameters in the model are appropriately assumed and optimized. Secondly, based on the set system parameters, programming simulation is carried out through LABVIEW software to analyse the spindle unbalance vibration response. Then, the unbalanced mass of the spindle and workpiece is kept constant, and the unbalanced mass of the spindle system is changed by adjusting the unbalanced mass of the workpiece, so as to simulate and analyse the unbalanced coupled vibration response of the spindle-workpiece system. Finally, based on changing the axial distance of the unbalanced mass of the workpiece relative to the end face of the chuck, the influence of different thicknesses of the disc workpiece on the system influence coefficient is simulated and analysed.

Based on the mathematical model of the workpiece-spindle-bearing system, LABVIEW software is used to prepare the system simulation program to analyse the unbalanced vibration response of the spindle, the unbalanced vibration response of the workpiece coupled with the spindle and the influence of the thickness of the workpiece on the influence coefficient of the system, so as to provide reference for the study of the formation of the unbalance of the disc-type workpieces and the on-line control method.

3.1. Spindle system simulation parameters

Table 1. Spindle system simulation parameters

Spindle centre of mass distance to front end (m)	0.3133
Spindle centre of mass distance to rear end (m)	0.4467
Spindle diameter(m)	0.085
Spindle mass (kg)	65.135
Spindle front bearing stiffness (N/m)	3×107
Spindle rear bearing stiffness (N/m)	1.5×107
Front bearing damping (N.s/m)	2000
Rear bearing damping (N.s/m)	1000
Spindle unbalance mass (kg)	0.08
Unbalanced mass coordinates ux (m)	0.025
Unbalanced mass coordinates uy (m)	0.2
Unbalanced mass coordinates uz (m)	0.025

ISSN: 2472-3703 DOI: 10.6911/WSRJ.202311_9(11).0004

In order to carry out simulation verification of the dynamic model, the structure of the spindle system of a CNC lathe is taken as the basis here, and appropriate assumptions are made on the parameters in the model, and then the unbalanced vibration response problem of the spindle system is simulated and analysed through simulation. The simulation parameters of the set spindle-bearing system are shown in Table 1.

3.2. Model workpiece simulation parameters

The disc-type workpiece model is simplified and hypothesized. The imbalance mass of the workpiece is equivalent to the imbalance excitation factor on the end face of the chuck, with the corresponding hypothetical parameters shown in Table 2:

Table 2. Spindle system simulation parameters

Workpiece unbalanced mass size(kg)	Workpiece unbalanced mass coordinates(m)			equivalent imbalance measure (g.cm∠O)
	ux	uy	uz	
0.05	0.08	0.4	0.08	56.57∠45

Based on the above parameter assumptions, the following simulation analysis is carried out for the unbalanced vibration response characteristics of the spindle system at a rotational speed of 1200 r/min.

3.3. Simulation of spindle unbalance vibration response

Based on the assumptions of the parameters of the spindle-bearing system, Labview software is used for programming to simulate the unbalanced vibration response of the spindle, and Figure 3 shows the unbalanced vibration response waveforms of the front and rear ends of the spindle and the axial trajectory diagram output from the system simulation programme.

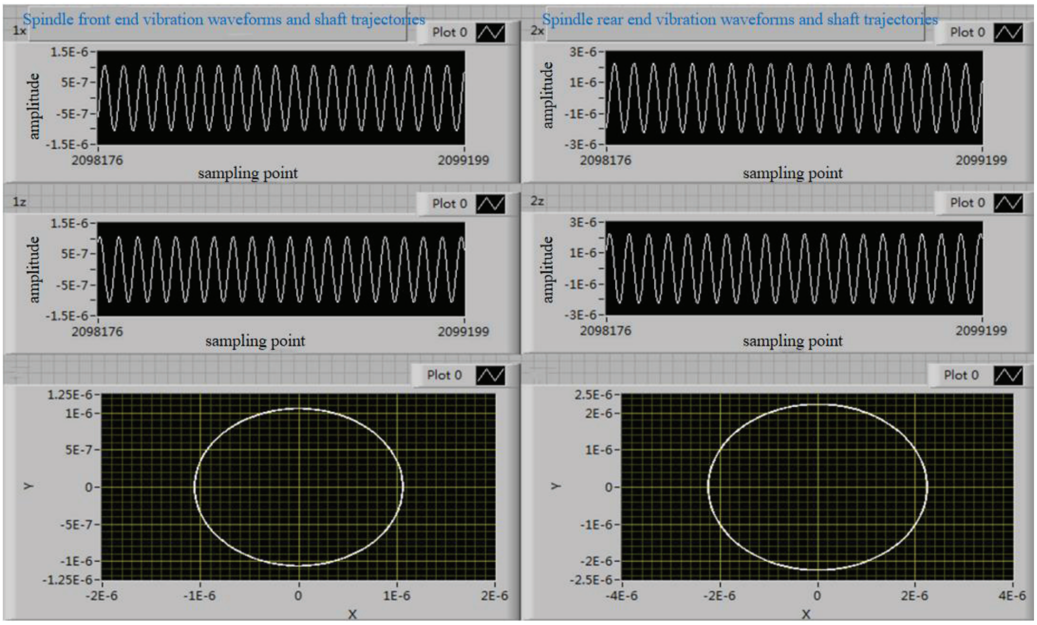

Figure 3. Simulation waveform

Based on Figure 3, it can be seen that the workpiece-spindle system belongs to a linear steady-state system under harmonic excitation, and its steady-state response is harmonic vibration with the same frequency as the rotation speed.

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Specifically, the vibration waveform in Figure 3 shows that under the imbalance excitation introduced by the rotating workpiece, the displacement response of the spindle system is periodic and sinusoidal. This demonstrates that the workpiece-spindle coupling system exhibits the characteristic vibration response of a linear time-invariant system to harmonic excitation. The imbalance force acting on the spindle through the workpiece-spindle interface serves as a sinusoidal excitation, which leads to the steady-state harmonic vibration at the same frequency in the spindle system.

3.4. Workpiece-spindle system unbalance coupling vibration response

To investigate the influence of workpiece imbalance on the spindle vibration, the positions of the imbalance masses of the spindle and workpiece are kept unchanged. The workpiece imbalance mass is set to 0, 0.001 kg, and 0.05 kg respectively. Simulations are performed by changing the imbalance mass of the spindle system. The simulation results are shown in Figure 4.

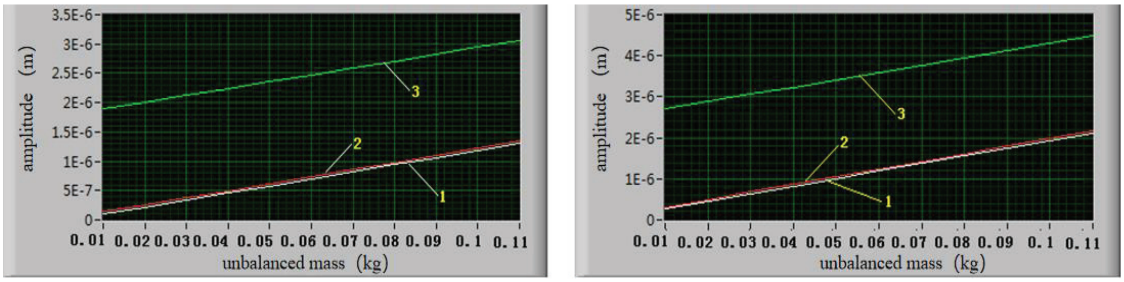

Figure 4. Unbalance vibration response at front and rear ends of spindle when unbalanced mass changes

Figure 4 shows the unbalanced vibration response at the front end of the spindle and the unbalanced vibration response at the rear end of the spindle, respectively, and the straight lines 1, 2, and 3 in the figure are the coupled unbalanced vibration response when the unbalanced mass of the workpiece is 0, 0.001, and 0.05, respectively. Based on Figure 4, it can be seen that the influence of workpiece imbalance on the spindle system's unbalance vibration response is quite stable. The vibration response caused by workpiece imbalance and the spindle system imbalance exhibit an approximate linear superposition relationship. Therefore, it is feasible to inversely calculate the disc-type workpiece imbalance based on the measured spindle imbalance vibration.

Specifically, the growth trends of the vibration amplitude with respect to the spindle imbalance under different workpiece imbalances are nearly parallel straight lines. This indicates the vibration contributions of the spindle imbalance and the workpiece imbalance can be decoupled and treated as linear summations. Given this linear superposition principle, if the total vibration of the coupled spindle-workpiece system is measured, the spindle's own vibration component can be estimated based on its known imbalance level. By subtracting this estimated spindle vibration from the total measured vibration, the remaining vibration caused by the workpiece imbalance can be obtained.

In this way, the disc-type workpiece imbalance can be inversely quantified without directly measuring the workpiece itself. This verifies the feasibility of monitoring workpiece imbalance conditions through spindle vibration analysis, which has important practical value for machining process optimization and quality control.

3.5. Influence of disc workpiece thickness on the system impact factor

To investigate the influence of disc-type workpiece thickness on the influence coefficient, we set the thickness of the chuck to be 86mm. The unbalance amount of the disc-shaped workpiece

ISSN: 2472-3703 DOI: 10.6911/WSRJ.202311_9(11).0004

is concentrated on the outer edge of the disc, with a radius of 113.12mm. Simulation results were obtained by varying the axial distance of the unbalance mass relative to the end face of the chuck, as shown in Figure 5.

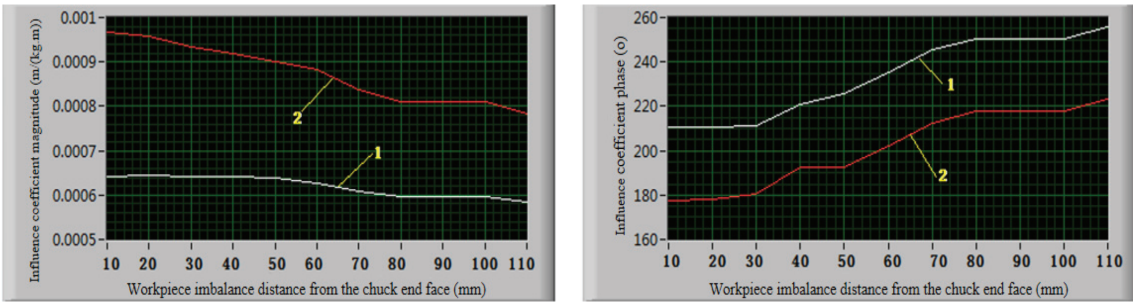

Figure 5. Workpiece thickness effect on the system influence coefficient

Figure 5 shows the amplitude and phase changes of the influence coefficients of the workpiece unevenness with respect to the spindle measurement point, respectively. In Figure 5, Curve 1 represents the amplitude and phase of the influence coefficient relative to the measurement point at the front end of the spindle, while Curve 2 represents the amplitude and phase of the influence coefficient relative to the measurement point at the rear end of the spindle. From Figure 5, it can be observed that when the distance of the unbalance mass relative to the end face of the chuck changes, there is a certain influence on the amplitude and phase of the influence coefficient with respect to the measurement points at the front and rear ends of the spindle. However, within a certain range, the changes in the amplitude and phase of the influence coefficient are relatively small and can be approximated as constants. To further analyze this, we conducted simulations by reducing the distance of the unbalance mass from the end face of the chuck for each variation, as shown in Figure 6.

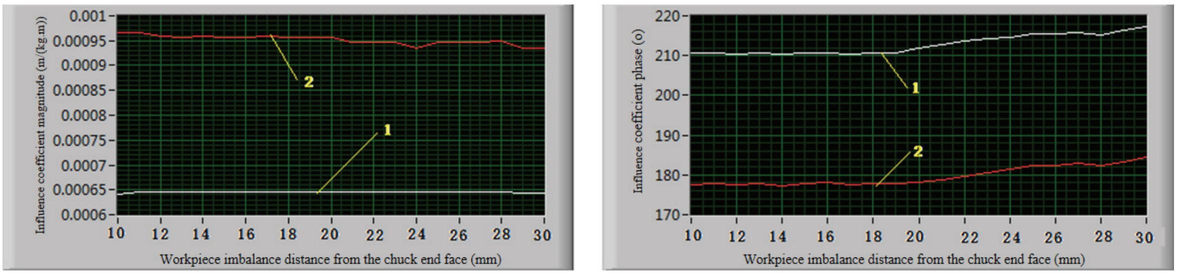

Figure 6. Workpiece thickness effect on the system influence coefficient

In Figure 6, Curve 1 represents the amplitude and phase of the influence coefficient relative to the measurement point at the front end of the spindle, while Curve 2 represents the amplitude and phase of the influence coefficient relative to the measurement point at the rear end of the spindle. From Figure 6, it can be observed that within the range of the unbalance mass distance from the end face of the chuck (less than 20mm or even 23mm), the changes in the amplitude and phase of the influence coefficient are relatively small, indicating that the influence coefficient is relatively stable. Therefore, for workpieces with a thickness-to-radius ratio greater than 4, they can be considered as thin disc workpieces, and their unbalance amount can be effectively equivalent to the end face of the chuck. As a result, the single-plane dynamic balancing method can be used to detect and correct the unbalance amount of the workpiece in reverse.

ISSN: 2472-3703 DOI: 10.6911/WSRJ.202311_9(11).0004

4. CONCLUSION

On the basis of the analysis of spindle system unbalance vibration problem, the paper incorporates the dynamic characteristics of disc workpiece unbalance, establishes the spindle workpiece integrated dynamic model, and studies the workpiece-spindle system unbalance coupling vibration response problem through the model simulation, and verifies the feasibility of online measurement of workpiece unbalance. The specific content reflects the following aspects.

(1) The basic structure model of workpiece-spindle-bearing is established with the basic structure of CNC machine tool spindle as a prototype, based on this model, and combined with the related knowledge of rotor dynamics, the kinetic equations of workpiece-spindle-bearing system are established.

(2) Aiming at the assumptions of the workpiece-spindle-bearing system dynamic equation and related parameters, the unbalanced vibration response of the spindle system is simulated and analysed by using LABVIEW language programming and simulation, and the correctness of the programming is verified.

(3) Through the analysis of workpiece-spindle system unbalance coupling vibration response, the feasibility of online reverse measurement of workpiece unbalance is verified; through the simulation analysis of the influence coefficient of the thickness of disc workpiece on the system, it is proposed that the workpiece with a diameter-to-thickness ratio of more than 4 can be regarded as a thin disc workpiece, and its unbalance can be equated to the end surface of the chuck, which provides preliminary technical support for the on-line measurement of the unbalance of disc workpieces.

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