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Research on Static and Dynamic Performance of Spindle System of the Grinding Head

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Abstract: The article studied the vibration resistance of spindle system in grinding head of roller grinding machine by means of harmonic response and modal analysis, the system modal analysis use the method of the transfer matrix, obtained the first five orders of the natural frequency of spindle system, calculated the natural frequency of spindle system by the displacement-frequency curve of the spindle system harmonic response. By studying the M84100A type spindle system in grinding head of roller grinding machine, obtained the conclusion that the natural frequency was far from the work frequency, so laid a foundation of the spindle system dynamic design.

Key words: Roller grinding machine, static and dynamic analysis, transfer matrix

INTRODUCTION

Roll grinder is indispensable production equipment. With the improvement of product quality requirements, the precision of roll grinder is increasing (Ast *et al*., 2009). As an important part of the roll grinder, the Spindle system static and dynamic performance directly affects the overall performance and the roll accuracy and surface quality (Gao *et al*., 2008). In this study, choose M84100A type roll grinding machine spindle system as the research object mainly used the transfer matrix method to analyze the vibration resistance of the spindle system.

Transfer matrix method: The transfer matrix method is mainly used for bending vibrations of the shaft components, torsional vibration and torsional vibration of the drive system to establish dynamics model (Huang and Lund, 1998). The method uses a lumped mass modeling, it can be used to deal with elastic support, multi-sectional, multi-supporting structure, the transfer matrix can measure the impact of the gyroscopic and unbalanced moment easily (Li, 2003).

Transfer matrix model of core elements: A simplified model of the spindle system.

The pulley, sleeve, locking nut Etc. increase the quality of the shaft section, the diameter of the equivalent mass can be calculated in accordance with the Eq. 1:

$$d_m = \sqrt{d^2 + 4m/(\pi\rho l)} \tag{1}$$

Where:

dm = Diameter of the shaft section equivalent mass
d = Diameter of the spindle shaft section
m = Quality of the bushings, pulleys, etc.
ρ = Density of spindle material
l = Length of the shaft sectio

Pulleys, bushings and other parts in addition to increasing the quality, but also increase the bending stiffness of the shaft section, The equivalent stiffness diameter d_k of the natural shaft section can be calculated according to the Eq. 2:

$$d_k = d + \frac{3B}{4} \tag{2}$$

Where:

d_k = He equivalent stiffness diameter
B = Width of the pulleys, bushings and other parts

Transfer matrix of the right end massless shaft segment: The transfer matrix links the two points of the system state vector. Usually the two selected points are those points of forming a component boundary (Boyaci *et al*., 2009):

- **Quality components:** The i-th segment axis l_i and i-th lumped mass m_i, respectively isolated from the system. Wherein the l_i the axial length, m_i is the concentrated mass, E is the modulus of elasticity of the material (Yin *et al*., 2005), I is the sectional moment of inertia about the neutral

For the ith concentrated mass m_i, assuming that the quality is absolutely rigid body and ignore its rotational

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inertia, when the system motion is transversely harmonic vibration (Jin and Cai, 2002), the mass, m_i motion direction is the Left (L) or Right (R), the lateral displacement y should be equal to the angular:

$$\theta\left(=\frac{dy}{dx}\right)$$

That:

$$\left.\begin{aligned} y_i^R &= y_i^L \\ \theta_i^R &= \theta_i^L \end{aligned}\right\} \tag{3}$$

The left and right sides bending moment M on the mass m_i should have the following relationship with the shear Q:

$$\left.\begin{aligned} M_i^R &= M_i^L \\ Q_i^R &= Q_i^L + m_i\omega^2 y_i^L \end{aligned}\right\} \tag{4}$$

Combining Eq. 3 and 4, write into matrix form according to the matrix multiplication rule obtained:

$$\begin{Bmatrix} y \\ \theta \\ M \\ Q \end{Bmatrix}_i^R = \begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 \\ 0 & 0 & 1 & 0 \\ m\omega^2 & 0 & 0 & 1 \end{bmatrix}_i \begin{Bmatrix} y \\ \theta \\ M \\ Q \end{Bmatrix}_i^L \tag{5}$$

Abbrevd:

$$Z_i^R = P_i Z_i^L \tag{6}$$

If m_i is simplified by the larger diameter part, the moment of inertia J_i can be not ignored. The first part of the Eq. 4 will turn into:

$$M_i^R = M_i^L - J_i\omega^2\theta_i^L \tag{7}$$

The remaining three formulas are unchanged, the mass element left and right sides condition transfer relationship Used to calculate the moment of inertia is obtained (Rehorn *et al*., 2004), being:

$$\begin{Bmatrix} y \\ \theta \\ M \\ Q \end{Bmatrix}_i^R = \begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 \\ 0 & -J\omega^2 & 1 & 0 \\ m\omega^2 & 0 & 0 & 1 \end{bmatrix}_i \begin{Bmatrix} y \\ \theta \\ M \\ Q \end{Bmatrix}_i^L \tag{8}$$

- **Shaft segment elements:** Figure 1 expresses force state diagram for massless shaft unit

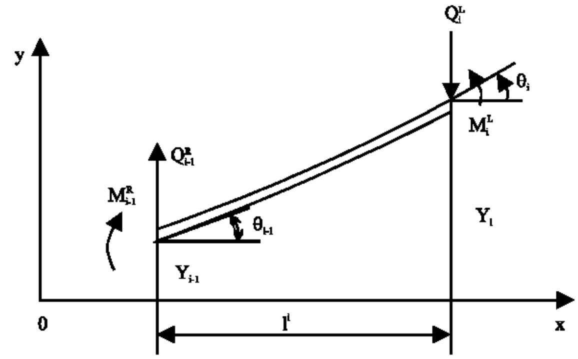

Fig. 1: Transfer matrix of the massless shaft section

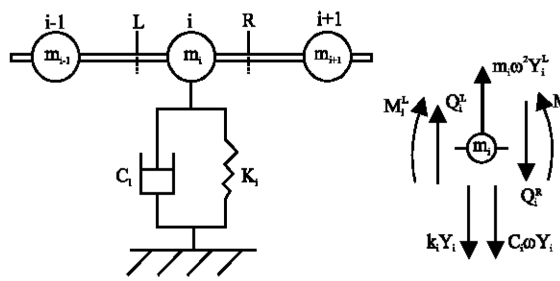

Fig. 2: Supporting elements transfer

In the Fig. 2, the ith axis without taking the shaft section quality into account, by the force equilibrium condition, obtained:

$$\left.\begin{aligned} Q_i^L &= Q_{i-1}^R \\ M_i^L &= M_{i-1}^R + Q_{i-1}^R l_i \end{aligned}\right\} \tag{9}$$

According to the mechanics of materials, the following relationship can be obtained:

$$\left.\begin{aligned} M &= EI\frac{d\theta}{dx} = EI\frac{d^2y}{dx^2} \\ \theta &= \frac{1}{EI}\int M dx \\ y &= \int \theta dx \end{aligned}\right\} \tag{10}$$

Gets:

$$\begin{aligned} \theta_i^L &= \theta_{i-1}^R + \frac{1}{(EI)_i}\int_0^{l_i} M_i^L dx = \theta_{i-1}^R + \frac{1}{(EI)_i}\int_0^{l_i}(M_{i-1}^R + Q_{i-1}^R x)dx \\ &= \theta_{i-1}^R + \frac{M_{i-1}^R l_i}{(EI)_i} + \frac{Q_{i-1}^R l_i^2}{2(EI)_i} \end{aligned} \tag{11}$$

$$\begin{aligned} y_i^L &= y_{i-1}^R + \int_0^{l_i}\theta_i^L dx = y_{i-1}^R + \int_0^{l_i}[\theta_{i-1}^R + \frac{M_{i-1}^R x}{(EI)_i} + \frac{Q_{i-1}^R x^2}{2(EI)_i}]dx \\ &= y_{i-1}^R + \theta_{i-1}^R l_i + \frac{l_i^2}{2(EI)_i}M_{i-1}^R + \frac{l_i^3}{6(EI)_i}Q_{i-1}^R \end{aligned} \tag{12}$$

The simultaneous Eq. 11 and 12:

$$\begin{Bmatrix} y \\ \theta \\ M \\ Q \end{Bmatrix}_i^L = \begin{bmatrix} 1 & 1 & \frac{l^2}{2EI} & \frac{l^3}{6EI} \\ 0 & 1 & \frac{l}{EI} & \frac{l^2}{2EI} \\ 0 & 0 & 1 & 1 \\ 0 & 0 & 0 & 1 \end{bmatrix}_i \begin{Bmatrix} y \\ \theta \\ M \\ Q \end{Bmatrix}_{i-1}^R \qquad (13)$$

Abbreved:

$$Z_i^L = F_i Z_{i-1}^R \qquad (14)$$

The simultaneous Eq. 5, 8 and 13 can be obtained:

$$Z_i^R = P_i Z_i^L = P_i F_i Z_{i-1}^R = T_i Z_{i-1}^R \qquad (15)$$

That:

$$\begin{Bmatrix} y \\ \theta \\ M \\ Q \end{Bmatrix}_i^R = \begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 \\ 0 & 0 & 1 & 0 \\ m\omega^2 & 0 & 0 & 1 \end{bmatrix}_i \begin{bmatrix} 1 & 1 & \frac{l^2}{2EI} & \frac{l^3}{6EI} \\ 0 & 1 & \frac{l}{EI} & \frac{l^2}{2EI} \\ 0 & 0 & 1 & 1 \\ 0 & 0 & 0 & 1 \end{bmatrix}_i \begin{Bmatrix} y \\ \theta \\ M \\ Q \end{Bmatrix}_{i-1}^R$$

$$= \begin{bmatrix} 1 & 1 & \frac{l^2}{2EI} & \frac{l^3}{6EI} \\ 0 & 1 & \frac{l}{EI} & \frac{l^2}{2EI} \\ 0 & 0 & 1 & 1 \\ m\omega^2 & ml\omega^2 & \frac{ml^2\omega^2}{2EI} & 1+\frac{ml^3\omega^2}{6EI} \end{bmatrix}_i \begin{Bmatrix} y \\ \theta \\ M \\ Q \end{Bmatrix}_{i-1}^R \qquad (16)$$

- **Supporting elements:** Figure 2 is the supporting elements transfer matrix, the i-th bearing equivalent stiffness k_i, equivalent viscous damping coefficient c_i, lumped mass m_i is known. the relationship between the i point two sides the corner, bending moment and shear involves the following:

$$\left.\begin{aligned} y_i^R &= y_i^L \\ \theta_i^R &= \theta_i^L \\ M_i^R &= M_i^L \\ Q_i^R &= Q_i^L + (m_i\omega^2 - ic_i\omega - k_i)y_i^L \end{aligned}\right\} \qquad (17)$$

Therefore the state vector transfer relationship formula of the support point left and right sides is the following:

$$\begin{Bmatrix} y \\ \theta \\ M \\ Q \end{Bmatrix}_i^R = \begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 \\ 0 & 0 & 1 & 0 \\ m\omega^2 - ic_i\omega - k & 0 & 0 & 1 \end{bmatrix}_i \begin{Bmatrix} y \\ \theta \\ M \\ Q \end{Bmatrix}_i^L \qquad (18)$$

Dynamic analysis of the spindle system: The coupling points of the respective units of the kinetic model are numbered, let the leftmost number is 0, the rightmost number is N, apply the expression of the transfer matrix of the above-described various units (Wu *et al*., 2012), point by point passing, set up the state vector relationship from 0-N between, i.e., the system of the transfer equation:

$$Z_N = T_N z_{N-1} = T_N T_{N-1} z_{N-2} = T_N T_{N-1} \cdots T_2 T_1 z_0 = T z_0 \qquad (19)$$

Because the transfer matrix T_i is 4-order matrix and their product, the system transfer matrix T is also 4-order squares, generally expressed as:

$$\begin{Bmatrix} y \\ \theta \\ M \\ Q \end{Bmatrix}_N = \begin{bmatrix} u_{11} & u_{12} & u_{13} & u_{14} \\ u_{21} & u_{22} & u_{23} & u_{24} \\ u_{31} & u_{32} & u_{33} & u_{34} \\ u_{41} & u_{42} & u_{43} & u_{44} \end{bmatrix} \begin{Bmatrix} y \\ \theta \\ M \\ Q \end{Bmatrix}_0 \qquad (20)$$

Such as the spindle assembly of the general machine have the two free ends, i.e., $M_0 = Q_0 = 0$, $M_N = Q_N = 0$, into Eq. 20, By substitution of the boundary condition, obtained:

$$\left.\begin{aligned} y_N &= u_{11}y_0 + u_{12}\theta_0 \\ \theta_N &= u_{21}y_0 + u_{22}\theta_0 \\ 0 &= u_{31}y_0 + u_{32}\theta_0 \\ 0 &= u_{41}y_0 + u_{42}\theta_0 \end{aligned}\right\} \qquad (21)$$

When the spindle transverse vibration, the y_0 and $è_0$ are insufficiency zero, so the Eq. 21 latter two equations must be equal to zero:

$$\Delta(\omega) = \begin{vmatrix} u_{31} & u_{32} \\ u_{41} & u_{42} \end{vmatrix} = 0 \qquad (22)$$

The transfer matrix elements u_{31}, u_{32}, u_{41}, u_{42} is a function of ω, the frequency value which satisfies the Eq. 22 is the f the natural frequency of the vibration system, the Eq. 22 is the transverse vibration frequency equation, Calculate the root of the equation ω_i. These roots, are frequency values satisfied the boundary conditions which is the each order critical speed of the rotor system.

Spindle system transfer matrix spindle: The spindle unit is mainly composed of three parts: spindle, bearing and transmission parts. The spindle system according to the size of each axis segment diameter and the shaft of the different parts with the principle of the transfer matrix method is discretized into a number of components to form the dynamic model of the chain structure

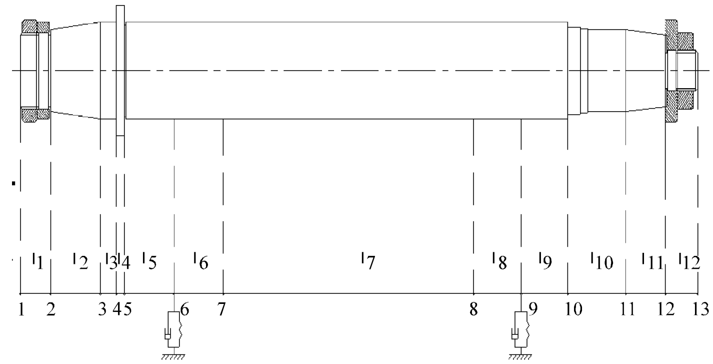

Fig. 3: Spindle system's model

(Mansor *et al*., 2012). The spindle material modulus of elasticity E is 206.0 GPa and density is 7850 kg m^{-3}. As is shown in Fig. 3.

The number 1, 2, 3, 4, 5, 7, 8, 10, 11, 12, and 13 in the Fig. 3 are the mass element ordinals; the I_1, I_2, I_3...I_{12} are the axis segment ordinals, the Eq. 6 and 9 Supporting elements ordinals. The relationship between the state vector from 1-13 is the system transfer equation:

$$Z_{13} = T_{25}z_{13} = T_{25}T_{24}z_{12} = T_{25}T_{24}\cdots T_2T_1z_1 = Tz_1 \tag{23}$$

Rewritten:

$$\begin{Bmatrix} y \\ \theta \\ M \\ Q \end{Bmatrix}_{13} = \begin{bmatrix} u_{11} & u_{12} & u_{13} & u_{14} \\ u_{21} & u_{22} & u_{23} & u_{24} \\ u_{31} & u_{32} & u_{33} & u_{34} \\ u_{41} & u_{42} & u_{43} & u_{44} \end{bmatrix} \begin{Bmatrix} y \\ \theta \\ M \\ Q \end{Bmatrix}_{1} \tag{24}$$

The boundary conditions are $M_1 = Q_1 = 0$ and $M_{13} = Q_{13} = 0$.

Substitution the above data and a variety of known conditions into Eq. 22, get a frequency equation about ω.

The general solution of the process :the first multiply unit transfer matrix, rather than the substitution of value and then obtain a equation about ω by the boundary conditions and finally Solve it.

Spindle transfer matrix calculation: Substitute the above data and a variety of known conditions into Eq. 22, Use MATLAB software to solve it.

Harmonic response calculation results: The harmonic response analysis is a linear analysis, this article uses the modal superposition method. During post-processing, The harmonic response analysis starts with POST26 to find critical forcing frequency (to generate maximum displacement or stress frequency (point of concern in the model) and then to deal with the entire model in these critical mandatory frequency using POST1, the results can to represent the color map, vector map and list. the exciting force size is 1400 N, the direction is UX, the frequency range is from 0-3000 Hz, calculate the response in the first order natural frequency, calculation are shown in Fig. 4.

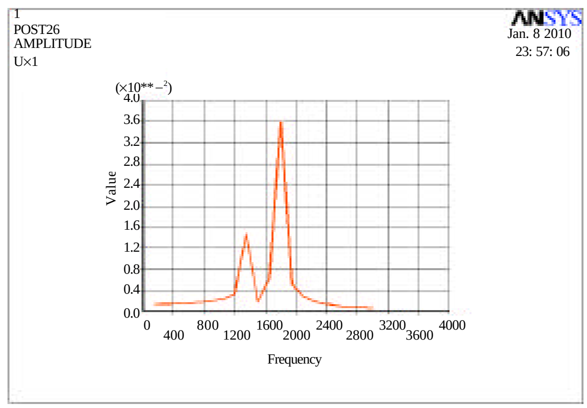

Fig. 4: Amplitude-frequency curve of spindle wheel point response

It can be seen that the maximum displacement of the spindle in the vicinity of the operating frequency is 1.5 μm, Resonance generates near the first order frequency, the maximum resonance displacement is 1.5 μm, the minimum dynamic stiffness: 1400/1.5 = 933 N μm^{-1}.

CONCLUSION

In this study, the transfer matrix method is applied in the modal analysis of the spindle system while use finite element software ANSYS for the spindle system static analysis, modal analysis and harmonic response analysis:

- Respectively get the natural frequency of the spindle, the first order frequency 1055 and 1174 Hz. The results of the two methods were compared and the results are relatively close to and away from the working frequency. So, the operating speed is safe
- Spindle unit static analysis and harmonic response analysis are carried using ANSYS, analysis spindle strength and stiffness, the result show that the strength and stiffness has a large margin, the further optimization can be done. Get the harmonic response near the spindle order frequency and maximum dynamic displacement of the spindle proved that the design is safe and reliable

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