Vertical Vibration Control of Cold Rolling Mill's Rolls Based on Time-delay Feedback Control Method

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Abstract. In this paper, a two degree of freedom nonlinear vertical vibration equation of the cold rolling mill with the dynamic rolling force was established, then the delay feedback control method was introduced into the equation to controlled the vertical vibration of the system. The amplitude-frequency equations of primary resonance of system was carried out by using the multi-scale method, and the resonance characteristics of different parameters of delay feedback control method were obtained by adopting the actual parameters of rolling mill. It is found that the size of the resonance amplitude value was effectively controlled and the resonance region and jumping phenomenon of the system were eliminated by selecting the appropriate time-delay parameters combination, which provides an effective theoretical reference for solving mill vibration problems.

1 Introduction

The vibration of rolling mill often occurs in rolling process. The occurrence of vibration not only affects the quality of rolling products, but also leads to breakdown of the rolling equipment. There are mainly two kinds of vibration in rolling mill, namely torsional vibration of main drive system and vertical vibration of rolls. Especially, the vertical vibration of rolls directly effects the exit thickness of strip. In order to restrain vertical vibration of rolls and improve stability of rolling process, the active control theory and technology has been used and to control the vertical vibration phenomenon. Zhang adopted the robust theory to control the strip tension of cold rolling mill and then inhibited the occurrence of vertical vibration [1]. Fuzzy neural network is used in hydraulic AGC system and the accuracy of strip is improved[2]. Time-delay phenomenon is very common in the control system. Plenty of researches have been studied on it in different research fields [3, 4]. Time-delay phenomenon also exists in the control system of rolling mill.

In this paper, in order to control resonance characteristics of rolling mill, the time-delay feedback control method is applied to the two degrees of freedom vertical vibration equation of cold rolling mill. By using the multi-scale method, the primary resonance amplitude-frequency characteristic equation is solved out, and the impact of time-delay parameters on the primary resonance are studied. The simulation results indicate that the size of resonance amplitude is effectively controlled, and the resonance region and jumping phenomenon of the system are eliminated.

2 The time-delay feedback control model of rolling mill's two degree freedom vibration system

Considering the effects of dynamic rolling force [5] and the up-down symmetry of rolling mills, a two degree of freedom nonlinear vertical vibration model of rolling mill is established in Figure 1.

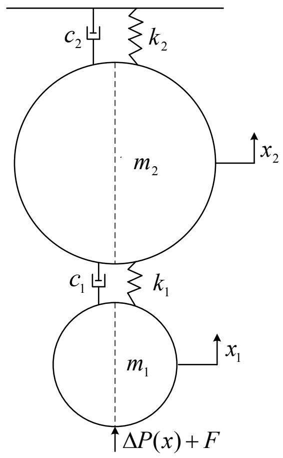


Figure 1. The mechanical vertical vibration model of rolls with nonlinear dynamic force.

The nonlinear dynamic equation of rolling mill in Figure 1 can be established as

$$\begin{cases} m_1\ddot{x}_1 + c_1(\dot{x}_1 - \dot{x}_2) + k_1(x_1 - x_2) = \Delta P(x) + F \\ m_1\ddot{x}_2 + k_2x_2 + c_2x_2 - k_1(x_1 - x_2) - c_1(\dot{x}_1 - \dot{x}_2) = 0 \end{cases} \quad (1)$$

Where, m_1 and m_2 are defined as the equivalent mass of top work roll and top back-up roll, respectively. k_1 is defined as equivalent stiffness between the top work roll and top back-up roll. k_2 and c_2 are defined as equivalent stiffness and equivalent damping between the top back-up roll and top beam of mill housing, respectively. F is defined as external excitation. $\Delta P(x)$ is dynamic variation of rolling force, $\Delta P(x)=b_1x+b_2x^2+b_3x^3$, $x=2x_1$.

The time-delay feedback control is applied in equation (1) and it can be expressed as

$$\begin{cases} m_1\ddot{x}_1+c_1(\dot{x}_1-\dot{x}_2)+k_1(x_1-x_2)=\Delta P(x)+F \\ \quad +g_1x_1(t-\tau_1)+g_2\dot{x}_1(t-\tau_2) \\ m_1\ddot{x}_2+k_2x_2+c_2\dot{x}_2-k_1(x_1-x_2)-c_1(\dot{x}_1-\dot{x}_2)=0 \end{cases} \tag{2}$$

Where, τ_1 and τ_2 are time-delay parameters, g_1 and g_2 are gain factor.

The equation (2) can be simplified as

$$\begin{cases} \ddot{x}_1+\omega_1^2x_1=\alpha_1x_2+\beta_1(\dot{x}_1-\dot{x}_2)+\gamma_1x_1^2 \\ \quad +\eta_1x_1^3+F_0+d_1x_1(t-\tau_1)+d_2\dot{x}_1(t-\tau_2) \\ \ddot{x}_2+\omega_2^2x_2=\alpha_2x_1+\beta_2\dot{x}_1+\gamma_2\dot{x}_2 \end{cases} \tag{3}$$

Where, $\omega_1^2=(k_1-2b_1)/m_1$, $\omega_2^2=(k_1+k_2)/m_2$, $\alpha_1=k_1/m_1$, $\beta_1=-c_1/m_1$, $\gamma_1=4b_2/m_1$, $\eta_1=8b_3/m_1$, $\alpha_2=k_1/m_2$, $\beta_2=c_1/m_2$, $\gamma_2=-(c_1+c_2)/m_2$, $d_1=g_1/m_1$, $d_2=g_2/m_1$, $F_0=F/m_1$.

3 Primary resonance analysis

Assuming equation (3) is a small nonlinear system, the right side of the equation (3) multiply by small parameter

$$\begin{cases} \ddot{x}_1+\omega_1^2x_1=\varepsilon[\alpha_1x_2+\beta_1(\dot{x}_1-\dot{x}_2)+\gamma_1x_1^2 \\ \quad +\eta_1x_1^3+F_0+d_1x_1(t-\tau_1)+d_2\dot{x}_1(t-\tau_2)] \\ \ddot{x}_2+\omega_2^2x_2=\varepsilon[\alpha_2x_1+\beta_2\dot{x}_1+\gamma_2\dot{x}_2] \end{cases} \tag{4}$$

In the case of primary resonance, $F_0=F_1\cos(\omega t)$, set $\omega_1^2=\omega^2+\varepsilon\sigma$, and substituting it in equation (4)

$$\begin{cases} \ddot{x}_1+\omega^2x_1=\varepsilon[\alpha_1x_2+\beta_1(\dot{x}_1-\dot{x}_2)+\gamma_1x_1^2+\eta_1x_1^3 \\ \quad +F_0+d_1x_1(t-\tau_1)+d_2\dot{x}_1(t-\tau_2)-\sigma x_1] \\ \ddot{x}_2+\omega_2^2x_2=\varepsilon[\alpha_2x_1+\beta_2\dot{x}_1+\gamma_2\dot{x}_2] \end{cases} \tag{5}$$

Where, σ is frequency modulation parameter,

By using multiple scales method, assuming it has the first-order approximate solution as follows

$$\begin{cases} x_1=x_{11}(T_0,T_1)+\varepsilon x_{12}(T_0,T_1) \\ x_2=x_{21}(T_0,T_1)+\varepsilon x_{22}(T_0,T_1) \end{cases} \tag{6}$$

Substituting equation (6) into equation (5), separating terms each order of ε, it has

$$\begin{cases} D_0^2x_{11}+\omega^2x_{11}=0 \\ D_0^2x_{21}+\omega_2^2x_{21}=0 \end{cases} \tag{7}$$

$$\begin{cases} D_0^2x_{12}+\omega^2x_{12}=-2D_0D_1x_{11}+\alpha_1x_{21}+\beta_1D_0(x_{11}-x_{21}) \\ \quad +\gamma_1x_{11}^2+\eta_1x_{11}^3+F_1\cos\omega t+d_1x_{11}(t-\tau_1) \\ \quad +d_1D_0x_{11}(t-\tau_2)-\sigma x_{11} \\ D_0^2x_{22}+\omega_2^2x_{22}=-2D_0D_1x_{21}+\alpha_2x_{11}+\beta_2D_0x_{11}+\gamma_2D_0x_{21} \end{cases} \tag{8}$$

Set the solution of equation (7) was

$$\begin{cases} x_{11}=A_1(T_1)e^{i\omega T_0}+\varepsilon\overline{A}_1(T_1)e^{-i\omega T_0}+cc \\ x_{21}=A_2(T_1)e^{i\omega_2T_0}+\varepsilon\overline{A}_2(T_1)e^{-i\omega_2T_0}+cc \end{cases} \tag{9}$$

Substituting equation (8) into equation (7), the following equation can be expressed as

$$\begin{cases} D_0^2x_{12}+\omega^2x_{12}=(-2i\omega D_1A_1+i\omega\beta_1A_1+3\eta_1A_1^2\overline{A}_1+0.5F_1 \\ \quad +d_1A_1e^{-i\omega\tau_1}+i\omega d_2A_1e^{-i\omega\tau_2}-\sigma A_1)e^{i\omega T_0}+(\alpha_1A_2 \\ \quad -i\omega_2\beta_1A_2)e^{i\omega_2T_0}+\gamma_1A_1^2e^{i2\omega T_0}+\eta_1A_1^3e^{i3\omega T_0}+2\gamma_1A_1\overline{A}_1+cc \\ D_0^2x_{22}+\omega_2^2x_{22}=(-2i\omega_2D_1A_2+i\omega_2\gamma_2A_2)e^{i\omega_2T_0} \\ \quad +(\alpha_2A_1+i\omega\beta_2A_1)e^{i\omega T_0}+cc \end{cases} \tag{10}$$

The secular term of equation (10) can be obtained as

$$\begin{cases} -2i\omega D_1A_1+i\omega\beta_1A_1+3\eta_1A_1^2\overline{A}_1+0.5F_1 \\ \quad +d_1A_1e^{-i\omega\tau_1}+i\omega d_2A_1e^{-i\omega\tau_2}-\sigma A_1=0 \\ -2i\omega_2D_1A_2+i\omega_2\gamma_2A_2=0 \end{cases} \tag{11}$$

The polar coordinate of A_1 and A_2 of the equation (11) is introduced as follows

$$\begin{aligned} A_1&=0.5a(T_1)e^{i\varphi_1(T_1)} \\ A_2&=0.5b(T_1)e^{i\varphi_2(T_1)} \end{aligned} \tag{12}$$

Substituting equation (12) into equation (11), separate the real part and the imaginary part and get

$$\begin{cases} \dot{a}=\dfrac{\varepsilon}{2\omega}[a\beta_1\omega-F_1\sin\varphi_1-d_1a\sin(\omega\tau_1) \\ \quad +a\omega d_2\cos(\omega\tau_2)] \\ \dot{\varphi}_1=-\dfrac{\varepsilon}{2a\omega}[\dfrac{3}{4}\eta_1a^3+F_1\cos\varphi_1+d_1a\cos(\omega\tau_1) \\ \quad +a\omega d_2\sin(\omega\tau_2)-a\sigma] \\ \dot{b}=\dfrac{1}{2}\varepsilon\gamma_2b \\ \dot{\varphi}_2=0 \end{cases} \tag{13}$$

In the steady-state, existing $\dot{a}=\dot{b}=\dot{\varphi}_1=\dot{\varphi}_2=0$, eliminate φ_1 and φ_2, then the primary resonance amplitude-frequency equation of the controlled equation of system can be obtained as

$$\begin{aligned} &[\frac{3}{4}\eta_1a^3+d_1a\cos(\omega\tau_1)+a\omega d_2\sin(\omega\tau_2)-a\frac{\omega_1^2-\omega^2}{\varepsilon}]^2 \\ &+[a\beta_1\omega-d_1a\sin(\omega\tau_1)+a\omega d_2\cos(\omega\tau_2)]^2=F_1^2 \end{aligned} \tag{14}$$

4 Simulation

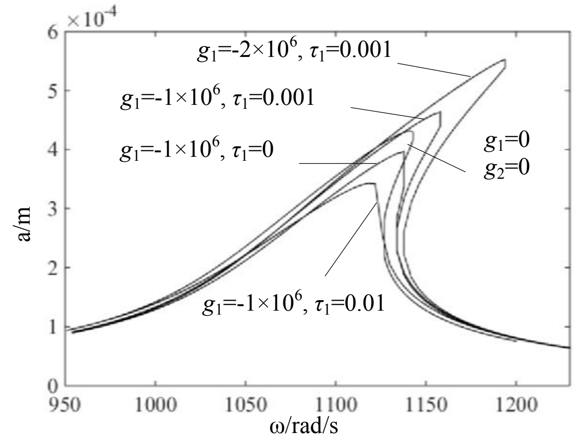
Taking the actual structure parameters and technological parameters of the rolling mill as an example, the detail parameters are as follows:

m_1=2.182×10^4kg, m_2=1.225×10^5kg, k_1=6.0186×10^{10} N/m, k_2=3.1825×10^{10}N/m, c_1=1.01×10^6 N•s/m, c_2=1.25×10^6 N•s/m, F_0=0.5×10^6 N.

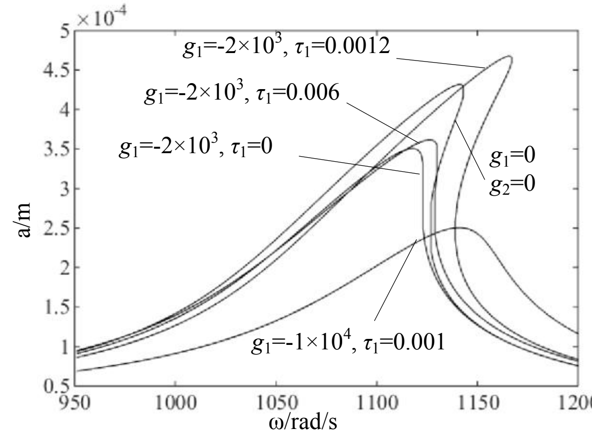
The nonlinear parameters of dynamic rolling force are as follows:

b_1=-4.478×10^9N/m, b_2=-2.910 ×10^{12} N/m^2, b_3=-3.256×10^{15} N/m^3, $\varepsilon = 0.01$.

Figures 2 illustrates primary resonance amplitude-frequency curves in different delay-time parameters.



(a) g_2=-1×10^3 and τ_2 =0.001



(b) g_1=-1×10^6 and τ_1 =0.001

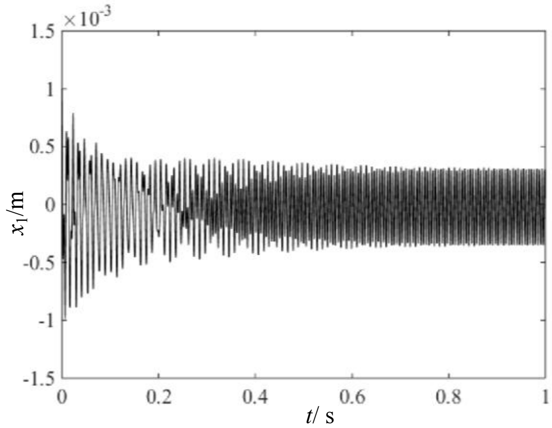
Figure 2. Primary resonance amplitude-frequency curves.

Figure 2(a) is the primary resonance amplitude-frequency curves of system when g_1 and τ_1 take different combinations and the time delay parameters g_2=-1×10^3, τ_2=0.001. When g_1=0 and g_2=0, it is the primary resonance amplitude-frequency curve of original system. If τ_1=0(namely no delay), when g_1=-1×10^6, the amplitude and resonance zone decreases. As τ_1 increased, the amplitude and resonance zone increases firstly, then the amplitude and resonance zone decreases, it means adjusting the parameters g_1 and τ_1 can effectively control the size of the resonance amplitude values.

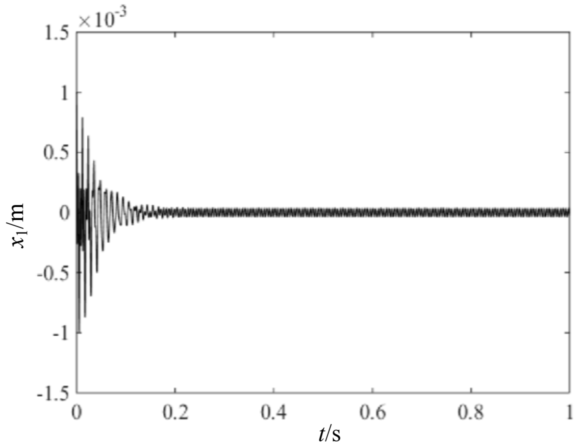
Figure 2(b) illustrates the primary resonance amplitude -frequency curves of system when g_2 and τ_2 take different combinations and the time delay parameters g_1=-1×10^6, τ_1=0.001. when g_2 is constant and τ_2 increases, firstly the amplitude decreases with τ_2 is no delay, then the amplitude increases with τ_2 increases. When τ_2 increase to a certain value, the amplitude decreases. when g_2 increases, the amplitude and resonance zone decreases.

Figures 3 are the time-domain curves of m_1 in the case of no control and control condition with ω is equal to 1140 rad/s.

Figure 3(a) is g_1 and g_2 are both equal to zero, namely no control. Figure3 (b) is time-domain curve in control, it can be seen that the resonance amplitude is less than no control, it means it could be improve resonance problem by adopting time-delay feedback method. So it can be seen adjusting the time-delay parameters can effectively reduce the amplitude of system, eliminate the resonance phenomenon and make the system tend to be stable more quickly.



(a) g_1=0, g_2=0, τ_1 =0, τ_1 =0



(b) g_1=-2×10^6, g_2=-1×10^3, τ_1 =0.001, τ_1 =0.001

Figure 3. The time-domain curves of m_1 with uncontrolled and controlled.

In summary, by using the time-delay feedback method to control the two degree of freedom vibration system and selecting the appropriate time-delay feedback parameters can effectively improve the primary resonance amplitude frequency characteristic of system and make the mill system in more stable state.

5 Conclusion

This paper firstly established a two degree of freedom vibration equation of the cold rolling mill with the dynamic rolling force. then the delay feedback control method is introduced into the equation to control the vertical vibration of system. In addition, the primary resonance amplitude-frequency characteristic controlled equation of system is solved out and the effect of time-delay parameters on the primary and internal resonance are analyzed.

The appropriate time-delay parameter combination is selected to effectively control the amplitude value of resonance. It eliminates the internal resonance region and jumping phenomenon of system, improved the stability of system, which provides an effective theoretical reference for the study of inhibiting the mill vibration problems.

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