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DYNAMIC DISTURBANCE SIMULATION OF MODAL CONTROL AND CONTROL MODEL OF TORSIONAL VIBRATION OF THE ROLLING MILL

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Keywords: Rolling mill's main driving system, Torsional vibration, Time domain simulations, Modal control and control model.

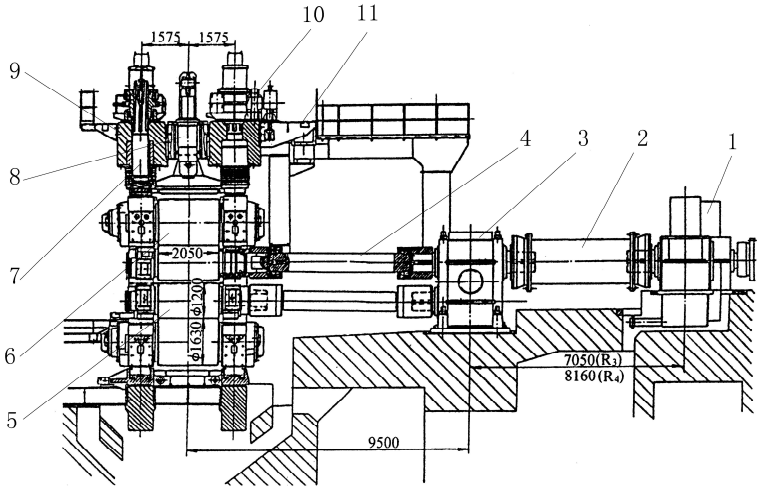
Abstract. For suppressing torsional vibration on main driving system of the rolling mill, the paper studied modal control and control model of torsional vibration. Dynamic disturbance of the rolling mill main driving system were induced by small perturbation from torque fluctuation. Modal control analysis were investigated on include amplitude and phase response on reference change or load change, bode diagrams, and time domain simulations. Torque fluctuation reflects non-steady cyclical or quasi-cyclical characteristics on the torsional vibration system. Control model had been reconstructed from inversion of dynamic topology theory. The result of simulation showed that the established control model for suppressing torsional vibration was effectiveness. Feedback control experiment of nipping and throwing steel on the rolling mill's main driving system using reaction control tactic was proved it could be easy input and output for controling to suppressing vibration.

INTRODUCTION

Torsional vibration had been a great influence on performance and the reliability of main driving system of machinery equipment. Torsional vibration of driving shafting was easily excitated by various outside changing, such as gap, time varying stiffness, friction or extemal torque excitation etl.. Torsional vibration instability mechanism and its overall dynamics become the key issues to achieve the optimal design of the rolling mill main driving system. Alternating load acting on the rolling mill's rotary driving shafts could often induce torsional vibration. Many control tactics were proposed on the basis of modern robust control theory. But those methods were hardly applied to practice because of their algorithm's high orders or non-decoupling. The reason was resulted from two limitations. One limitation was that the rotary machine was a critical unstable dynamic system; the other was the system included integrating processes which could not make the closed-loop system naturally seek a steady state operating level if the manipulated and disturbance variables remained constant for a sufficient period of time. In order to control this kind of unstable state of the systems, in our work, a new method of analyzing torsional vibration was proposed by using topology dynamic theory for easy to calculation and control. The study had solved the problems mentioned by analyzing and calculating the parameters of the multi-mass model. Transient load acting on all key parts of the rolling mill rotary shafts could be simulated by topology network torsional vibration control model, which could be confirmed the security and fatigue life of the main driving system on the rolling mill.

CONTROL MODEL OF TORSIONAL VIBRATION ON ROLLING MILL

Based on the systematic dynamic theory, when torsion angle of the rotatory system was taken as system input and torque taken as system output, topology analysis methods may be used to calculate or control on multi-branch torsional vibration of the rolling mill main driving systems. The structure of four roller mill was shown as Fig. 1. It's mechanical model was shown as Fig. 2. The control function of torsional vibration response was as follows:



1. gear reducer, 2. intermediate shaft, 3. herringbone gear frame, 4. gear coupled spindle, 5. work roll, 6. backup roll, 7. pressure device, 8. backup roll balance device, 9. back to release device, 10. rack, 11. platform.

Fig. 1 Frame and main driving system of four roller mill

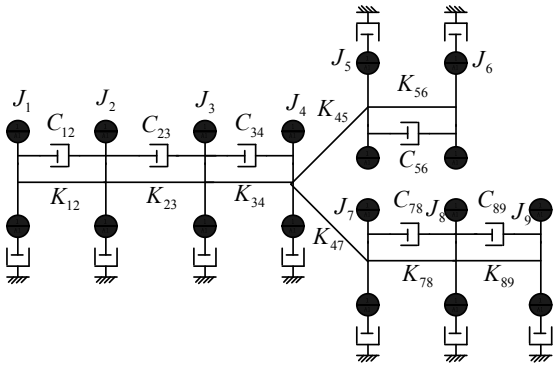


Fig. 2 Control model of torsional vibration of the four roller mill

$$G_{T_0,\theta_n} = P_{T_0,\theta_n} D_{2n+2} / D_1 \tag{1}$$

Where, $D_1$was characteristics of the formula of torsional vibration in topology network, while $i>2$, the D_i was transfer function in the system; $G_{T0,\theta n}$ was torsional vibration network function of system export using θ_n; $P_{T0,\theta n}$ was torsional vibration instigated point function.

$$P_{T_0,\theta_0} = L_1 L_3 \cdots L_{2n-1} Z_n^{-1} \tag{2}$$

$$D_{2n+2} = 1 + \sum_{i=2n+2}^{2N} L_i + \sum_{i_2=2n+4}^{2N} \sum_{i_1=2n+2}^{i_2-2} L_{i_2} L_{i_1} + \cdots + \sum_{i_{N-n-1}=2N-2}^{2N} \sum_{i_{N-n-2}=2N-4}^{i_{N-n-1}-2} \cdots \sum_{i_2=2n+4}^{i_3-2} \sum_{i_1=2n+2}^{i_2-2} L_{i_{N-n-1}} L_{i_{N-n-2}} \times \cdots L_{i_2} L_{i_1} + \prod_{i=n+1}^{N} L_{2i} \tag{3}$$

$$D_1 = 1 + \sum_{i=1}^{2N} L_i + \sum_{i_2=3}^{2N} \sum_{i_1=1}^{i_2-2} L_{i_2} L_{i_1} + \cdots + \sum_{i_{N-1}=2N-3}^{2N} \sum_{i_{N-2}=2N-5}^{i_{N-n-1}-2} \cdots \sum_{i_2=3}^{i_3-2} \sum_{i_1=1}^{i_2-2} L_{i_{N-1}} L_{i_{N-2}} \cdots L_{i_2} L_{i_1} + \sum_{i_N=2N-1}^{2N} \sum_{i_{N-1}=2N-3}^{i_{N-n-2}} \cdots \sum_{i_2=3}^{i_3-2} \sum_{i_1=1}^{i_2-2} L_{i_N} L_{i_{N-1}} \cdots L_{i_2} L_{i_1}$$

$$L_i = U_{i+1/2} \big/ Z_{i-1/2} \quad i = 1\ ,\ 3\ ,\ 5\cdots \tag{4}$$

$$L_i = U_{i/2} \big/ Z_{i/2} \quad i = 2\ ,\ 4\ ,\ 6\cdots \tag{5}$$

If the torque T_n was chosen as the output variable, there was only one forward path from T_0 to T_n. Then, the complex frequency control function of each output variable lead to

$$G_{T_0,T_n} = P_{T_0,T_n} D_{2n+1} / D_1 \tag{6}$$

$$P_{T_0,T_n} = L_1 L_3 \cdots L_{2n-1} \tag{7}$$

$$D_{2n+1} = 1 + \sum_{i=2n+1}^{2N} L_i + \sum_{i_2=2n+3}^{2N} \sum_{i_1=2n+1}^{i_2-2} L_{i_2} L_{i_1} + \cdots + \sum_{i_{N-n-1}=2N-3}^{2N} \sum_{i_{N-n-2}=2N-5}^{i_{N-n-1}-2} \cdots \sum_{i_2=2n+3}^{i_3-2} \sum_{i_1=2n+1}^{i_2-2} L_{i_{N-n-1}} L_{i_{N-n-2}} \cdots L_{i_2} L_{i_1} + \sum_{i_{N-n}=2N-1}^{2N} \sum_{i_{N-n-1}=2N-3}^{i_{N-n-2}} \cdots \sum_{i_2=2n+3}^{i_3-2} \sum_{i_1=2n+1}^{i_2-2} L_{i_{N-n-1}} L_{i_{N-n-2}} \cdots L_{i_2} L_{i_1} \tag{8}$$

Table 1 was the basis parameters of the four roller mill main driving systems. The mechanical elements and the inertias of the whole driving train were considered by the time.

Table 1 Parameters table of torsional vibration of four roller mill main driving systems

Moment of inertia/(kg·m^2)		Stiffness×10^6 /(N·m·rad^{-1})		Damp×10^{-2} /(N·m·s·rad^{-1})		Calculation of general mechanics				Topology network method			
						Circular frequency p /(rad·s^{-1})		Frequency f /Hz		Circular frequency p /(rad·s^{-1})		Frequency f /Hz	
J_1	4 060.000	$k_{1\to2}$	103.200	$c_{1\to2}$	1.000	p_1	98.345	f_1	15.652	p_1	98.6460	f_1	15.700
J_2	120.000	$k_{2\to3}$	164.800	$c_{2\to3}$	1.000	p_2	102.618	f_2	16.332	p_2	102.416	f_2	16.300
J_3	66.000	$k_{3\to4}$	100.000	$c_{3\to4}$	2.000	p_3	536.504	f_3	85.387	p_3	536.584	f_3	85.400
J_4	67.850	$k_{4\to5}$	6.321	$c_{4\to5}$	1.000	p_4	676.628	f_4	107.690	p_4	676.699	f_4	107.700
J_5	27.190	$k_{5\to6}$	6.105	$c_{5\to6}$	2.000	p_5	739.709	f_5	117.730	p_5	739.531	f_5	117.700
J_6	285.900	$k_{4\to7}$	55.260	$c_{4\to7}$	1.000	p_6	1 343.840	f_6	213.880	p_6	1 343.973	f_6	213.900
J_7	9.449	$k_{7\to8}$	6.134	$c_{7\to8}$	1.000	p_7	2 324.500	f_7	369.960	p_7	2 324.779	f_7	370.000
J_8	25.800	$k_{8\to9}$	5.841	$c_{8\to9}$	1.000	p_8	2 755.070	f_8	438.480	p_8	2 755.177	f_8	438.500
J_9	285.900												

The important eigen movement between the motor and the loads was expressed by f_{N1} =15.7 Hz. The corresponding anti-resonance frequency f_{A1} =14.8 Hz was big enough to meet a sufficient dynamic behaviour of the speed control. The important eigen movement between upper and lower transmission shafts was characterized by f_{N2} =16.3 Hz which was nearly equal to the corresponding anti-resonance frequency f_{A2} =16.1 Hz. In practise this eigen movement was only excited if there was an unsymmetrical load surge to top or bottom spindle. The mass ratio could be calculated if it was possible to approximate the multi-mass system by an equivalent two-mass system. This meaned that the low-frequency dynamic behaviour of the entire system must be dominated by the first natural frequency between the motor and the loads.

MODAL CONTROL AND TIME DOMAIN SIMULATIONS OF TORSIONAL VIBRATION ON ROLLING MILL

Torsional Vibration System Bode Diagrams. Bode diagrams was shown as Fig. 3, which admit further insight to the gains of the rolling mill main driving systems at the resonance points. Its behavior was normal and Fig. 4 resonance peak of the shaft speed was a bit high. For investigation on response to the alternated load changed must be excited by a outside load torque:

Torsional Vibration System Time Domain Simulations. Time domain simulations(considering clearances triggered by microscale effect) including the closed-loop speed control were completed. Since the normal proportional-integral speed controller was characterized by a high overshoot after a speed reference step for the rolling mill was very different. It was consisted of two separate controller parts, a proportional one for the reference behavior and a proportional-integral one for the disturbance behavior. Moreover the speed control loop contained a band stop filter whose stop frequency could be tuned due to the actual natural frequency of the driving system. The optimal

tuning of the stop frequency was about 5% to 10% below the corresponding natural frequency. It had to be investigated from case to case whether it was better to tune to the first order or the second order natural frequency. The rise time of the motor was about 90 ms. It was not possible to reduce the first natural vibration further by a band stop filter. The clearances in the gearbox had low effect on the vibration behavior. The influences on the motor and the shaft speed were therefore also proportionally smaller (Fig 5).

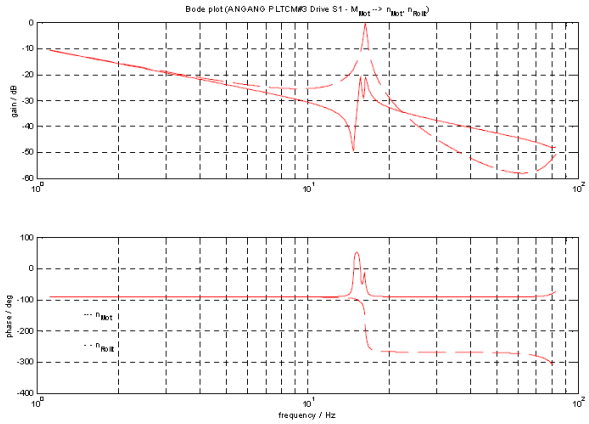


Fig. 3 Amplitude and phase response to reference change(n_Mot, -- n_Rollt)

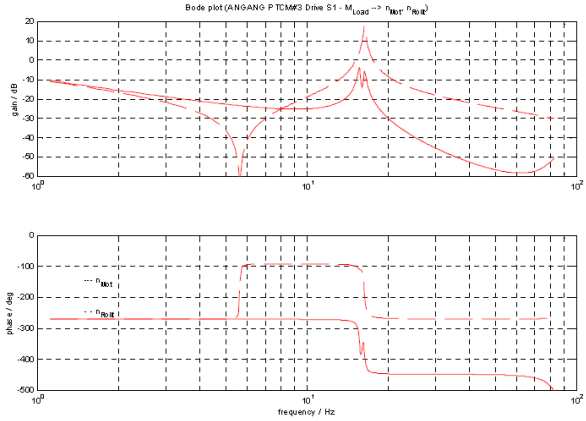


Fig. 4 Amplitude and phase response to load change(n_Mot, -- n_Rollt)

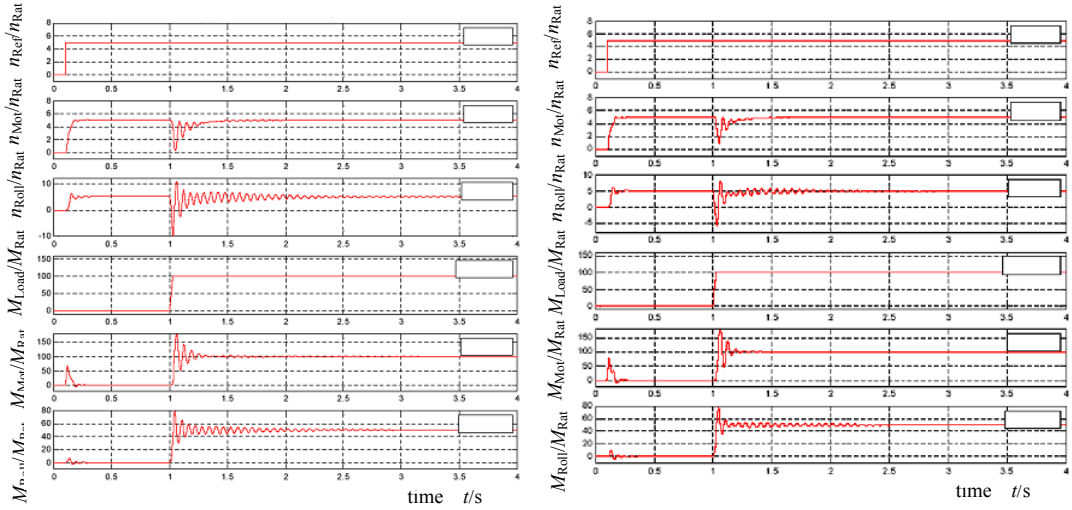


Fig. 5 Time domain simulations of torsional vibration of rolling mill

As it was expected due to the mass ratio and bode diagram for the load changing, the spindle speed was damped rather badly. However damping in the shaft gap and by tensional effects may be higher. Where, n_{Ref} was baseline motor speed; n_{Rat} was motor speed; n_{Mot} was actual motor speed; n_{Roll} was spindle speed; M_{Load} was load torque; M_{Rat} was motor torque; M_{Mot} was motor torque; M_{Roll} was spindle torque.

ESTABLISH FEEDBACK CONTROL MODEL AND ANALYSIS CONTROL PROCESSING

Through design and simulate on the experimental platform of coupling spindle of the rolling mill, feedback control model about torsional vibration driving system was established. Based on the experimental system, it may be realized to simulate the torque fluctuation of the coupling spindle of the rolling mill. The structure of experimental platform was shown as Fig. 6.

During the course of simulation experiment on torsional vibration of the rolling mill, coupling spindle was drived rotating by the electric motor. The reference torque sensor was tested and collected the torque signal from electric motor shaft. The other torque sensor was tested the dynamic torque signal from the coupling spindle. The brake system provided the alternated torque

changing to meet the practice conditions on simulation. Two kinds of testing data were input into the computer in order to compare and control the export power of the electric motor, which purpose was to reduce torsional vibration of the driving shafts. Based on Eq. (6), control equation which was description of simulated experimental platform were build as Eq. (9).

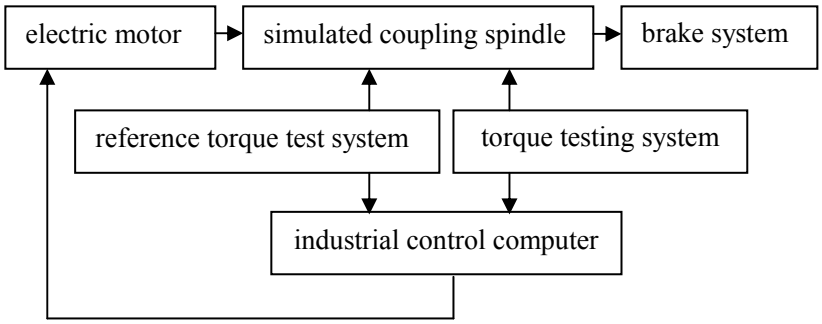


Fig. 6 The block diagram of simulation coupling spindle experimental platform

$$\begin{Bmatrix} \phi_m \\ \phi_1 \\ \phi_2 \end{Bmatrix} = \begin{Bmatrix} \frac{1}{I_m k_m s^2 + H_m s + K_m} & 0 & 0 \\ 0 & \frac{1}{I_1 k_1 s^2 + H_1 s + K_1} & 0 \\ 0 & 0 & \frac{1}{I_2 k_2 s^2 + H_2 s + K_2} \end{Bmatrix} \begin{Bmatrix} T_m \\ K_m \phi_m \\ K_1 \phi_1 + T_{Load} \end{Bmatrix} \tag{9}$$

Based on automatic control theory, each elastic shaft was a second-order transfer function. Torque of load was equal the sum that included the dynamic torque of motor and feedback torque of elastic shaft. So, the transfer function on control system of Fig. 6 were described as Fig. 7.

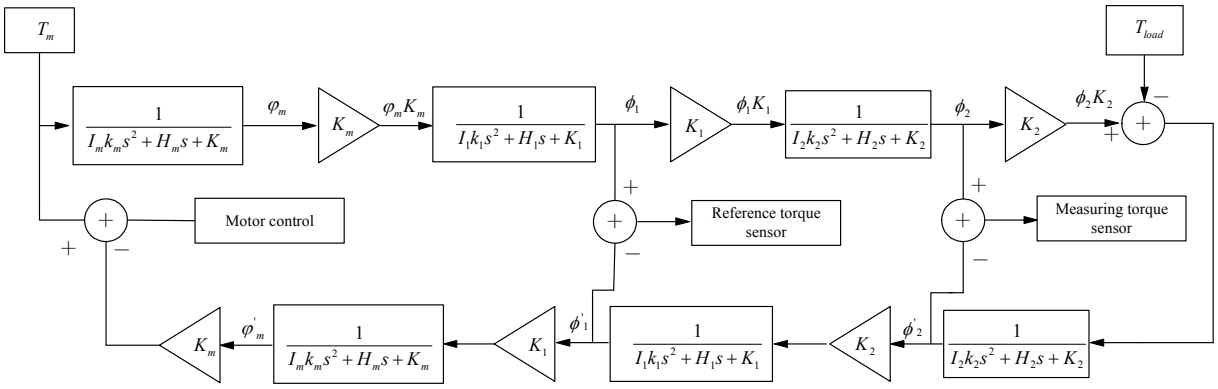


Fig. 7 Feedback contral diagram of simulation experimental platform

When the elastic shaft had be born with shocking, its torque signal was step load form. Feedback torque was returned into control module. Control commands were issued from the contral computer after completing data processing in order to reduce the amplitude of torsional vibration system.

According to the principle above, the electric motor rotating speed could be controlled by the actual torque which was feedback from elastic shafts end. Then the goal of suppression torsional vibration could be realized. The simulated compare chart of nipping steel and throwing steel on the rolling mill's main driving system using the reaction control in the previous or the following was shown as Fig. 8. From Fig. 8, it could be seen clearly that the torque fluctuation of elastic rotating shaft obviously reduced under the condition of reaction control, which explained that the torsional vibration had been restrained in some degree.

CONCLUSION

The research using systematical dynamic topology theory to establish the control model in order to suppress torsional vibration of rolling mill's main driving system were proved effectiveness. Modal control analyzed that small perturbation from torque fluctuation could induce system torsional vibration. Simulation of time domain include amplitude and phase response on reference change or load change, bode diagrams, and time domain simulations. Torque fluctuation reflects non-steady cyclical or quasi-cyclical characteristics on torsional vibration system. The result of simulations showed that the developing methods was feasible to simulate the rotation vibration response. The study acquired well consistency between the new methods and the other classical methods through the calculation example and simulation. Also the reconstruction control model from inversion of dynamic topology theory can easy be input and output for the control system to suppress torsional vibration. The result of the experiment showed that control tactics for suppress torsional vibration are correctness.

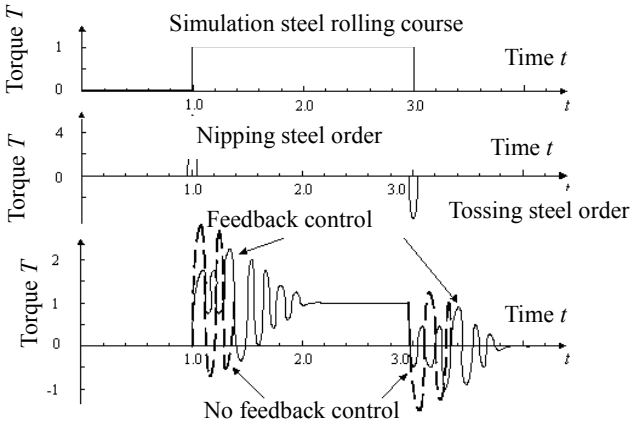


Fig.8 Torque change of elastic rotating shaft under feedback and no feedback control

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