Torsional Vibration Modeling and Simulation of Printing Cylinder in Printing Machine

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Abstract. Printing cylinder torsional vibration is one of the main vibration forms of printing machine, which directly restricts the increase of the printing speed. The impression cylinder is the core part of the printing press. Based on the D'Alembert's principle, an impression cylinder torsional vibration mechanics model based on concentration-distribution quality was established and compared with the CAE simulation results to verify the accuracy of the mathematical model. Conclusion: With the change of printing pressure and printing speed, the influence of excitation force on torsional vibration will also change. The results provide a reliable model for the active control of torsional vibration of printing cylinders.

1. Instruction

The torsional vibration of the impression cylinder not only affects the time when the impression cylinder enters the steady state process, but also brings the impact of angular displacement to the cylinder and affects the printing quality.

The research of torsional vibration control in printing machinery field is still in the stage of theoretical modeling and experiment. Buck and others\cite{1,2} put forward that sheet-fed offset press uses continuous gear transmission chain to drive all the rollers synchronously, which is easy to produce torsional vibration, and discussed the nonlinear vibration phenomenon. Martin\cite{3} established a flexible multi-body dynamics model to simulate the nonlinear effect of printing press. Wang\cite{4} studied the relationship between the fault of double screen printing and transmission accuracy and put forward a way to control printing quality by detecting the working condition of transmission unit. Bartlett\cite{5} studied the vibration and torsional behavior of high speed printing cylinder. Hermanski\cite{6} studied the dynamics of the printing roller rotor, analyzed the vibration characteristics of the printing roller, and proposed a suppression strategy. Masahiro\cite{7} and Daniel\cite{8} used a built-in vibration absorber in offset press to rapidly reduce the peak of instantaneous vibration after the cylinder is impacted into the gap. Cai\cite{9} studied the torsional vibration problem of the cylinder, actively controlled the system to largely suppress the torsional vibration and achieved a system's fast and vibration-free response. Wang\cite{10}, Cheng\cite{11} and Liu\cite{12} studied the vibration and cylinder dynamics of printing press.

This paper takes the impression cylinder of printing press as the research object, a discrete torsional vibration model of the concentrated-distributed mass of the impression cylinder is established, and the simulation experiment is carried out.
2. Establishment of a Discrete Model of Concentrated-Distributed Mass for Printing Machine Impression Cylinder

2.1 Printing machine impression cylinder description
Taking a certain type of printing machine entity as a research object, the structure diagram of the printing machine impression cylinder is established, as shown in Fig.1. In the figure, 1 is a motor; 2 and 4 are synchronous belt wheels, 3 is synchronous belt; 5, 6 and 9 are meshing helical gears; 7 represents impression cylinder; 8 and 11 are the opening and closing cams of the impression cylinder and the blanket cylinder respectively; 10 represents a blanket roller.

![Figure 1. Schematic diagram of printing machine impression cylinder structure](image1.png)

![Figure 2. Impression cylinder distribution mass model](image2.png)

2.2 Establishment of impression cylinder distribution mass model
In Fig.2, the distribution mass model of the impression cylinder is established. In order to simplify the calculation, each rotation step of the cylinder is used as a unit, and the natural frequency is calculated by applying the Holzer transfer matrix method. The state vectors of each element are represented by a column matrix which includes two elements: rotation angle $\phi$ and torque $M$. The angular displacement of a certain section is expressed as $\phi(x,t)$.

The kinetic energy $W(t)$ and potential energy $V(t)$ of torsional vibration of cylinder segmented element are as follows respectively:

$$W(t) = \frac{1}{2} \int_0^L \rho J \left( \frac{\partial \phi(x,t)}{\partial t} \right)^2 dx - \frac{1}{2} \int_0^L \rho \sum_{i=1}^n \phi_i(x) \dot{q_i}(t)^2 dx$$

$$V(t) = \frac{1}{2} \int_0^L G J \left( \frac{\partial \phi(x,t)}{\partial x} \right)^2 dx = \frac{1}{2} \int_0^L G J \sum_{i=1}^n \phi_i(x) \dot{q_i}(t)^2 dx$$

Where $\rho J$ is the moment of inertia per unit length of the cylindrical unit, $GJ$ is the torsional stiffness of the cylindrical element, $\phi(x)$ is the modes of each order of the impression cylinder element; and $F(t)$ is the corresponding generalized coordinates.

Therefore, by substituting equations (1) and (2) into the Lagrangian equation, we can obtain the torsional vibration equations for each segment of the impression cylinder as:

$$[J] \ddot{q} + [K] q = M$$

Where $[J]$ and $[K]$ is the generalized torsional inertia matrix and generalized stiffness matrix respectively, $M$ is generalized torque vector, $q$ is generalized coordinate vector.

The discrete values of moment of inertia and rotational stiffness of the impression cylinder are obtained by SolidWorks software as shown in table 1 and table 2:
Table 1. Discrete values of moment of inertia of impression cylinder

| Number | Moment of Inertia/ kg • m² | Number | Moment of Inertia/ kg • m² |
|--------|---------------------------|--------|---------------------------|
| $J_{11}$ | 0.032                     | $J_{55}$ | 56.22                     |
| $J_{22}$ | 0.056                     | $J_{66}$ | 12.48                     |
| $J_{33}$ | 0.048                     | $J_{77}$ | 0.048                     |
| $J_{44}$ | 12.48                     | $J_{88}$ | 0.056                     |

Table 2. Discrete values of torsional stiffness of impression cylinder

| Number | Torsional Stiffness $10^5$ N • m/ rad | Number | Torsional Stiffness $10^5$ N • m/ rad |
|--------|----------------------------------------|--------|----------------------------------------|
| $K_{11}$ | 4.02                                   | $K_{55}$ | 16.62                                   |
| $K_{22}$ | 8.88                                   | $K_{66}$ | 39.74                                   |
| $K_{33}$ | 27.8                                   | $K_{77}$ | 27.8                                    |
| $K_{44}$ | 39.74                                   | $K_{88}$ | 8.88                                    |

2.3 Establishment of a model for the centralization-distribution mass of the impression cylinder

In Fig. 3, where $J_m$ is the torque of the motor; $J_1$ and $J_2$ are the moment of inertia of the small helical gear and the large helical gear respectively; $[J]$ is the matrix of the impression cylinder the moment of inertia; $K_i$ is the torsional stiffness of the shaft for the motor; $[K]$ is the torsional stiffness matrix of the impression cylinder; $M_e$ is the driving moment of the motor, $M_i$ is the impression cylinder gap excitation torque, $M_c$ is the cam opening and closing mechanism excitation torque. According to the principle of energy conservation, the rotational inertia and torsional stiffness of the gear pair are equivalently transformed. The equivalent diagram is shown in Fig. 4:

\[ M = J \beta = J \frac{d\omega}{dt} = J \frac{d^2\phi}{dt^2} \]  

(4)

In the formula, where $\omega$ is angular acceleration; $t$ is rotation time; $\phi$ is angular displacement. The moment of inertia after conversion can be obtained as follows:

\[ J = \left( \frac{n_2}{n_1} \right)^2 J_1 = i^2 J_1 \]  

(5)

In the formula, where $i$ is the gear transmission ratio.
Taking the rotational speed of the active system as the benchmark, \( J = 1.2 \text{kg} \cdot \text{m}^2 \) was obtained.

### 3. Establishment mathematical model of torsional vibration of printing cylinder

According to the simplified model of Fig.4, the torsional vibration model of cylinder system is established by applying the Newton’s second law, as shown in Fig.5.

\[
\begin{bmatrix}
M_e \\
\phi_m \\
\phi_1
\end{bmatrix}
= \begin{bmatrix}
J_n \\
J_1 \\
\sum J
\end{bmatrix}
\begin{bmatrix}
k_1 \\
[k] \\
[k]
\end{bmatrix}
\begin{bmatrix}
\phi_m \\
\phi_1 \\
\phi
\end{bmatrix}
\]

Figure 5. torsional vibration model of printing machine cylinder system

In Fig.5, where \( J_m \) is torque of motor, \( J_1 \) is equivalent moment of inertia of helical gear pair and pulley, \( J_2 \) is the moment of inertia of impression cylinder, \( K_1 \) is the torsional stiffness of shaft for motor, \([K] \) is the torsional stiffness of impression cylinder. \( M_e \) is the load disturbance torque, and \( M_i = M_i + M_e \), where \( M_i \) is the impression cylinder gap excitation moment and \( M_e \) is the excitation moment of cam tooth opening and closing mechanism.

The differential equations of motion are listed as follows:

\[
\begin{align*}
-J_m \ddot{\phi}_m + k_1 (\dot{\phi}_m - \dot{\phi}_1) &= M_a \\
J_1 \ddot{\phi}_1 + k_1 (\dot{\phi}_m - \dot{\phi}_1) + k_2 (\phi_1 - [\phi]) &= 0 \\
\sum J \ddot{\phi} + k_2 (\phi_1 - [\phi]) &= M_e
\end{align*}
\]

The corresponding dynamic mathematical model in matrix form is obtained, In the undamped state, the system torsional vibration differential equation:

\[
\begin{bmatrix}
\sum J \ddot{\phi}_2 \\
\phi
\end{bmatrix} + [K_z][\phi] = [M_e]
\]

In the formula, the matrix is the \( n \times n \) order matrix, where \([J_z] \) is the moment of inertia matrix, \([K_z] \) is the torsional stiffness matrix, \([\phi] \) is the angular displacement vector and \([M_e] \) is the external load vector of the system.

### 4. Simulation analysis of torsional vibration of impression cylinder of printing press

#### 4.1 Establishment of ANSYS simulation model

Modal analysis is used to determine the vibration characteristics of structures or members. The main structural parameters of the impression cylinder are shown in Fig.2. Modal analysis consists of the following steps:

1. Modeling mainly includes the definition of parameters and establishing geometric model. The material of the impression cylinder is HT250 high grade cast iron with a modulus of elasticity of 210GPa, a density of 7800kg/m³, and a Poisson's ratio of 0.27. When performing modal analysis, the entity is selected as the SOLID element.

2. According to the established distributed mass model, meshed, and the first 6 modes is analyzed. The freeform meshing of the impression cylinder is selected, and the number of divided units is 16807 and the number of nodes is 29672.

3. Apply constraints and solve, which mainly includes defining loads, boundary conditions, specifying loading process settings, and solving.

#### 4.2 Simulation results of torsional Vibration Mode of impression cylinder body

By applying the Block Lanczos method in ANSYS, the natural frequencies of the first 6 torsional vibrations of the cylinder body (shown in table 3) and the natural vibration modes are obtained (shown in Fig.6). Compared with the calculated results of the natural frequency, the accuracy of the dynamic model and the mathematical model is verified.
Combined with the natural frequency values of table 3, it can be seen from Fig.6 that in the same working state, and the higher order torsional vibration has a large number of nodes, so the cylinder body is not easy to be excited. Therefore, the torsional frequency of the cylinder is mainly considered, which is close to the natural frequency of the first order torsional vibration.

### 4.3 Analysis of ADAMS simulation results

The ADAMS simulation is carried out, which was based on the interference moment and the specific numerical value of the motor input speed. By setting the observation point, the output of the angular velocity on the surface of the impression cylinder can be obtained; at the same time, using the built mathematical model for MATLAB simulation. The results of the two simulations are shown in Fig.7 and Fig.8.

![Figure 7. Curves of angular velocity of impression cylinder at 8000s/h rises to 10000s/h](image)

It can be seen from Fig.7 and Fig.8 that a large peak occurs at the beginning of the speed change. The reason is that during the speed increasing process, the motor is in the acceleration phase, causing it to have some motion coupling with the helical gear pair for a certain period of time. During the two speed-up intervals, the angular velocity of the torsional vibration of the impression cylinder fluctuates within a certain range and eventually stabilizes with time. Table 4 shows the change of the rotation period of the impression cylinder and the cylinder angular velocity during the rotation.

![Figure 8. Curves of angular velocity of impression cylinder at 10000s/h rises to 12000s/h](image)
Table 4. Numerical simulation and model simulation results of impression cylinder

| Speed Increase Interval (r/h) | Numerical Simulation | Model Simulation |
|------------------------------|----------------------|------------------|
|                              | Max Amplitude Oscillation Time | Max Amplitude Oscillation Time |
| 8000s/h Speed up to 10000s/h | 2700 0.5             | 2700 0.55        |
| 10000s/h Speed up to 12000s/h| 2800 0.7             | 2800 0.80        |

From table 4, it can be found that the maximum amplitude and oscillation time of the simulation results basically match in the same speed increasing interval. In different speed-up intervals, with the increase of basic speed, the results of simulation under both operating conditions will gradually increase, and the maximum amplitude of the oscillation and the cycle of oscillation will increase. This is because the increase in the rotational speed causes various torsional vibration excitation forces in the impression cylinder to increase. Therefore, in order to increase the operating speed of the printing machine, it is necessary to suppress the torsional vibration so that the angular velocity amplitude and oscillation time at a high speed are stable within a range where the influence on the system is small.

5. Conclusion
A torsional vibration model of the impression cylinder of the printing machine was established, and the natural modal parameters of the impression cylinder body were obtained through the finite element simulation analysis of the torsional vibration. The impression cylinder torsion vibration analysis of the printing press cylinder is performed during the speed changing. The paper verified the accuracy of the established mathematical model and established a reliable foundation for the study of torsional vibration control of the impression cylinder. The change in the operating speed of the printing machine is more pronounced than the torsional vibration of the impression cylinder when it is stable, and it needs to be controlled.

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