An improved adaptive control method for active balancing control of rotor with time-delay

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Abstract: In a high speed rotor system, active balancing control is an important method to improve performance and robustness. The time-delay will significantly increase the stability of the balance control system. Based on the dynamic model of rotor system with time-delay, a strictly positive real system is built to satisfy the stability conditions of adaptive control system in this paper. Furthermore, a feed-forward gain adaptive controller is designed. To improve the balance control of rotor system under both variable speed and constant speed, an adaptive control method is proposed. The advantage of this method against traditional control strategy is validated by numerical and experimental results.

Keywords: high speed rotor, time-delay, active balancing, adaptive control

Classification: Circuits and modules for electronic instrumentation

References

[1] Xiang, M. and Wei, T: “Autobalancing of high-speed rotors suspended by magnetic bearings using LMS adaptive feedforward compensation,” Journal of Vibration and Control 20 (2014) 1428 (DOI: 10.1177/1077546313479990).

[2] Qiao X and Zhu C: “The active vibration attenuation of a built-in motorized milling spindle,” Journal of Vibration and Control 19 (2013) 2434 (DOI: 10.1177/1077546312456230).

[3] Pan X, Wu H Q and Gao J J: “New Liquid Transfer Active Balancing System Using Compressed Air for Grinding Machine,” Journal of Vibration and Acoustics–Transactions of the ASME 137 (2015) 011014 (DOI: 10.1115/1.4028507).

[4] Dyer S W, Ni J and Shi J: “Robust optimal influence-coefficient control of multiple-plane active rotor balancing systems,” Journal of Dynamic Systems, Measurement and Control, Transactions of the ASME 124 (2002) 41 (DOI: 10.1115/1.1435622).

[5] ZAPOMEL J, Ferfecki P and Forte P: “Analysis of the steady state unbalance...
response of rigid rotors on magnetorheological dampers: stability, force transmission and energy dissipation,” International Journal of Applied Mechanics 6 (2014) 1450022 (DOI: 10.1142/S1758825114500227).

[6] El-Saeidy F M A: “Dynamics of a Rigid Rotor Linear/Nonlinear Bearings System Subject to Rotating Unbalance and Base Excitations,” Journal of Applied Nonlinear Dynamics 16 (2010) 403 (DOI: 10.1177/1077546309103565).

[7] Tiwari R and Chougale A: “Identification of bearing dynamic parameters and unbalance states in a flexible rotor system fully levitated on active magnetic bearing,” Mechatronics 24 (2014) 274 (DOI: 10.1016/j.mechatronics.2014.02.010).

[8] Sun Y X and Xu J: “Experiments and analysis for a controlled mechanical absorber considering delay effect,” Journal of sound and Vibration 339 (2015) 25 (DOI: 10.1016/j.jsv.2014.11.005).

[9] Ren Y, Chen X and Cai Y: “Rotation modes stability analysis and phase compensation for magnetically suspended flywheel systems with cross feedback controller and time delay,” Mathematical Problems in Engineering (2016) (DOI: 10.1155/2016/3783740).

[10] Zhang Y F, Zhang S and Liu F X: “Motion analysis of a rotor supported by self-acting axial groove gas bearing system with double time delay,” Proceedings of the Institution of Mechanical Engineers, Part C: Journal of Mechanical Engineering Science 228 (2014) 2888 (DOI: 10.1177/0954406214523581).

[11] Chen Q, Liu G and Zheng S Q: “Suppression of imbalance vibration for AMBS controlled driveline system using double-loop structure,” Journal of Sound and Vibration 337 (2015) 1 (DOI: 10.1016/j.jsv.2014.09.047).

[12] C. Liu and G. Liu: “Field dynamic balancing for rigid rotor-AMB system in a magnetically suspended flywheel,” IEEE/ASME Trans. Mechatron. 21 (2016) 1140 (DOI: 10.1109/TMECH.2015.2495225).

[13] Ghazavi M R and Sun Q: “Bifurcation onset delay in magnetic bearing system by time varying stiffness,” Mechanical System and Signal Processing 90 (2017) 97 (DOI: 10.1016/j.ymspp.2016.12.016).

[14] Huan R H, Chen L X and Sun J Q: “Multi-objective optimal design of active vibration absorber with delayed feedback,” Journal of Sound and Vibration 339 (2015) 56 (DOI: 10.1016/j.jsv.2014.11.019).

[15] Hu Bing and FANG Zhichu: “Adaptive control of active balancing system for a fast speed-varying Jeffcott rotor with actuator time delay,” Journal of Shanghai Jiao tong University (Science) 12 (2008) 297 (DOI: 10.1007/s12204-008-0297-z).

[16] Ren BB, Ge SS and Lee TH: “Adaptive Neural Control for a Class of Nonlinear Systems With Uncertain Hysteresis Inputs and Time-Varying State Delays,” IEEE Transactions on Neural Networks 20 (2009) 1148 (DOI: 10.1109/TNN.2009.2016959).

[17] Juan Xu, Yang Zhao and Zhiyuan Jia: “Rotor dynamic balancing control method based on fuzzy auto-tuning single neuron PID,” IEICE Electronics Express 14 (2017) 20170130 (DOI: 10.1587/exlex.14.20170130).

[18] Carlucho L, De Paula M and Villar SA: “Incremental Q-learning strategy for adaptive PID control of mobile robots,” Expert Systems with Applications 80 (2017) 183 (DOI: 10.1016/j.eswa.2017.03.002).
1 Introduction

With high speed rotary machines, vibration control is increasingly important to efficiency and reliability of mechanical system [1]. The unbalance is the main cause of vibration in high speed rotor [2]. As the main approach to restrain the unbalance, active balancing control can effectively improve running accuracy and stability of the rotor.

At present, the active balancing theories for first-order critical speed of the rigid rotor in constant speed have been well developed [3]. However, there is scarcity of practical active balancing technology especially for high-speed rotor system. Dyer et al. [4] combined online estimation of the influence coefficient with adaptive control method of the flexible rotor and realized real-time active balance with the help of mass redistribution actuator. Unfortunately, this method merely reached the balance in constant speed. Zapomel et al. [5] studied unbalanced loading rotor dynamics with flexible support by means of computational modeling. These rotors were equipped with two short magnetic dampers to alleviate lateral vibrations. Yet it is not applicable for high-speed rotor system. Beltran-Carbajal et al. [6] proposed an active balance control scheme for a Jeffcott rotor with variable speed, which used a suspension with a linear electromechanical brake. In addition, the estimation scheme of disturbing signal caused by inherent eccentricity on a rotating mechanical system and the controlling means to track rotor speed were obtained. Tiwar and Chougale [7] came up with the identification algorithm for estimating the dynamic parameters of the active magnetic bearing and the rotor residual unbalance, which was completed with least squares fitting technique and the PID controller in frequency domain.

However, these two methods failed to take the time-delay [8] into consideration. For high-speed rotor system, even a microsecond time-delay can cause non-synchronization which will impact the stability of rotor system and dynamic response. The research on active balancing control of rotor system with variable time-delay has attracted wide academic attention. Ren et al. [9] investigated the effect of time-delay on the rotation mode stability of a magnetic bearing flywheel with strong gyroscopic effect and suggested using an efficient phase compensation method in the basis of cross channel. Zhang et al. [10] presented a dynamic response method in order to analyze the gas bearing-rotor nonlinear system including time-delay, which trajectory diagram, time series and phase diagram were conducive to analyze time-delay and feedback that were under the influence of system dynamic response triggered by control gains. But these two aforementioned methods were complicated and prone to generate larger computational errors.

To sum up, an ideal control method integrates the high balance efficiency and the stability, the later of which requires minimum time-delay. In order to obtain a superior active balancing control performance of high-speed rotor, this paper takes both the movement and the time-delay of actuator into consideration. Firstly, a dynamic model of rotor system with time-delay is built, and a filtering function is introduced to build a strictly positive control system. Secondly, a
feed-forward gain adaptive controller is successfully designed. Thirdly an improved adaptive control method is proposed, which can realize the unbalance control with time-delay with both variable speed and constant speed.

2 Materials and methods

2.1 System dynamic model with time-delay

The high-speed spindle 240XDJ10Y of CNC milling center is selected for example, and its structure is shown in Fig.1(a). The parts which are less relevant to the dynamics of the model are neglected in the model, including the front axle cavity, shell and bearing block. The system is modeled as a rigid rotating shaft and a bearings device with a rotor active balancing system, as shown in Fig.1(b).

![Fig.1](image1.png)

**Fig.1.** Model of high-speed spindle. (a) structure model, (b) equivalent model

The total mass of the rigid rotating shaft is m, which includes the original unbalance mass $m_1$ and corrected unbalance mass $m_2$ of actuator. Stiffness and damping factor of the bearings are K and C, respectively. The inertial coordinate system \( \{X, Y, Z\} \) was established with Z axis coincident with the center line of the bearings. The origin of relative coordinate system \( \{x, y, z\} \) coincides with geometric center of the rotor. And \( \beta(t) \) represents a rotation angle of the rotor in the relative coordinate system, as shown in Fig.2.

![Fig.2](image2.png)

**Fig.2.** Coordinate geometry model of the rotor system

In order to simplify the problem, angular acceleration \( \alpha \) and angular velocity \( \omega \) of the rotor are considered as constant values in the different working conditions. The initial rotation angle and initial angular velocity are zero, and rotation angles can be expressed as follows:

\[
\begin{align*}
\beta_1(t) &= \frac{1}{2} \alpha t^2 + \beta_0 = \frac{1}{2} \alpha t^2 \\
\beta_2(t) &= \omega t + \omega_0 = \omega t
\end{align*}
\]  \quad (1)
As shown in Fig.2, the coordinate of the rotor geometry center is \((r_x, r_y)\). The coordinates of original unbalance mass \(m_1\) and corrected unbalance mass \(m_2\) can be expressed as \((a_x, a_y)\) and \((b_x, b_y)\), respectively. Most of the actuators in the balancing system contain two corrected unbalance masses that can be driven by DC motors [11]. And the corrected unbalance value is a vector summation of the individual masses. Generally, the unbalance value can be expressed as mass-radius product or centripetal force [12,13]. In this paper, the mass-radius product is chosen. The original unbalance value \(p\) and the corrected unbalance value \(q\) are expressed as follows:

\[
p = m_1 a
\]

\[
q(t) = m_1 b(t)
\]  

(2)

In the Eqn.(2), \(a\) and \(b\) are positional vectors from center of rotor to the original unbalance mass and the corrected unbalance mass, respectively, with \(a = [a_x, a_y]^T\) and \(b = [b_x, b_y]^T\). Assuming stiffness of bearings are the same in all direction, the unbalance values are equal in amplitude but different in phase by 90 degrees. According to Newton's second law, the dynamic differential equations of rotor system with actuator's time-delay can be obtained:

\[
m \frac{d^2 r}{dt^2} + c \frac{dr}{dt} + kr = -\frac{d^2}{dt^2} \{\Omega [p + q(t - \tau)]\}
\]  

(3)

In the Eq.(3), \(\Omega = \begin{bmatrix} \cos(\beta) & -\sin(\beta) \\ \sin(\beta) & \cos(\beta) \end{bmatrix}\). The \(r\) is displacement of the rotor center in the coordinate systems \([X, Y, Z]\) and \(r = [r_x, r_y]^T\). According to Eq. (3), the differential operator \(s\) is introduced to derive \(r(t)\) as

\[
r(t) = \frac{-s^2}{m(s^2 + 2\varepsilon\omega_n s + \omega_n^2)} \{\Omega [p + q(t - \tau)]\}
\]  

(4)

Where \(\omega_n = \sqrt{\frac{\Omega}{m}}\) is natural frequency, and the \(\varepsilon = \frac{c}{\sqrt{4mk}}\) is damping ratio of system.

2.2 Active balancing control method

A single adaptive control method is not ideal because the rotor system has different vibration states under variable speed and constant speed. In this paper, an improved active balancing control method combining the traditional adaptive control method with the PID control method is proposed. According to different operating conditions of the rotor, different control methods are adopted. The inputs to the control system are vibration and speed signals from sensors and feedback signal of controller. The system recognizes the conditions of the rotor system through the speed signal, depending on which the controller can switch between two control methods. Finally, the controller outputs a control signal to
drive motors of the actuator. Two balance blocks are used to achieve balance of the rotor. A principle diagram of the improved active balancing control system is shown in Fig.3.

Fig.3. Improved active balancing control principle diagram

2.2.1 The design of adaptive controller

To achieve optimal stability, an adaptive control strategy is designed by using feed-forward gain adaptive control method [14]. The Eq. (4) is rearranged as follows:

\[ r(t) = G(s) \left[ - \frac{d}{dt} \{ \Omega[p + q(t - \tau)] \} \right] \]

\[ G(s) = \frac{s}{m(s^2 + 2\omega_n s + \omega_n^2)} \]  
(5)

Where \( G(s) \) is a linear transfer function of the rotor system. If the time-delay of actuator \( \tau = 0 \), the adaptive control strategy is designed as follows:

\[ \frac{dq}{dt} = q = -\lambda \frac{d}{dt} \Omega r = \lambda \frac{d}{dt} \Omega^* \]
(6)

Where the positive real number \( \lambda \) is a systematic influence coefficient, \( \Omega^* \) is the conjugate transpose matrix of \( \Omega \).

Hu Bing et al.[15] utilized step-by-step procedure and Ren B. et al.[16] used structural Lyapunov-Krasovskii functional method to prove that if the transfer function \( G(s) \) of rotor system is strictly positive real and the time-delay \( \tau \) is less than the boundary of system’s time-delay \( \tau' \), the adaptive control strategy proposed by this paper can satisfy stability conditions while \( \lim_{t \to \infty} \| r(t) \| = 0 \).

\( G(s) \) in Eqn.5 is not a strictly positive real transfer function. In order to satisfy the stability conditions above, a filter function is introduced to make the transformed transfer function \( G_r(s) \) strictly positive.

\[ r_f(t) = \frac{s + \delta}{s} r(t) = G_r(s) \left[ - \frac{d}{dt} \{ \Omega[p + q(t - \tau)] \} \right] \]  
(7)

Substituting Eqn. (5) into Eq. (7),

\[ G_r(s) = \frac{s + \delta}{m(s^2 + 2\omega_n s + \omega_n^2)} \]  
(8)
The \( r_f(t) \) is an output of the original system after adding the filter function and the \( \delta \) is a small positive number with the requirement \( 0 < \delta \leq \varepsilon \omega_n \).

The Eqn. (5) and Eq. (7) have exactly the same form, and \( G_r(s) \) in Eq. (7) is a strictly positive real transfer function. Therefore, the adaptive control strategy

\[
\frac{dq}{dt} = q = -\lambda \frac{d\beta}{dt} \Omega r = \lambda \frac{d}{dt} \Omega^* r_f
\]

satisfies the stability conditions and can be used in the active balancing control system of rotor. The principle diagram is shown in Fig.4.

![Fig.4. Adaptive control principle diagram](image)

### 2.2.2 The design of PID controller

Proportional, integral and differential control methods are called PID control method. It is widely used because of its simplicity, flexibility and stability [17,18]. Based on the initial system model of rotor, the schematic diagram of PID control system is shown in Fig.5.

![Fig.5. PID control principle diagram](image)

The \( \tau_0(t) \) is the expected vibration of the rotor system. The \( r(t) \) is the output vibration of the rotor system. And \( e(t) \) is the difference value between the expected vibration and the output vibration, \( e(t) = \tau_0(t) - r(t) \). These time constants, \( K_p, K_i, K_d \), are proportional coefficient, integral coefficient and differential coefficient of the control system, respectively. The PID control in time domain is expressed as follows:

\[
q(t) = K_p e(t) + K_i \int_0^t e(t)dt + K_d \frac{de(t)}{dt}
\]

Substituting Eq. (10) into Eq. (4), PID control of the rotor system is obtained in the following expression:

\[
r(t) = \frac{-s^2}{m(s^2 + 2\varepsilon \omega_n s + \omega_n^2)} \{\Omega^2 p + [K_p e(t) + K_i \int_0^t e(t)dt + K_d \frac{de(t)}{dt}]\}
\]
2.2.3 The design of control algorithm

Combining the two control methods above, the flow chart of the improved active balancing control algorithm designed in this paper is shown in Fig.6.

![Flow chart of control algorithm](image)

**Fig.6.** Flow chart of control algorithm

The detailed steps of the algorithm are stated as follows:

**Table 1.** The steps of the algorithm

| Algorithm: Improved adapting control method for active balancing control |
|---|
| **Input:** A speed signal \( \omega(t) \), A vibration signal \( r(t) \) |
| **Output:** Controlling signal of corrected unbalance \( q(t) \) |
| **Initialization:** Set expected vibration value \( r_0(t) \) and limited amplitude \( \varepsilon_0 \) |
| Detecting the initial signals \( \omega(t) \) and \( r(t) \), determine initial vibration and operating conditions. |
| **Loop:** While \( r(t) > r_0(t) \) do |
| If \( \omega(t) \) is a constant value, |
| Get error signal \( e(t) = r_0(t) - r(t) \), calculate controlling signal \( q(t) \) by the Eq.(10); |
| Else if \( \omega(t) \) is a time-variable value, |
| Get initial \( \Omega(t) \) and calculating \( q(t) \) through integral operation; |
| Else |
| \( q(t) = 0 \); |
| End |

3 Results and discussion

3.1 Simulation verification

In order to verify the method proposed in this paper, a Simulink simulation model of the rotor control system is built, as shown in Fig.7. It is assumed that the time-delay of the actuator is known. The values of constants are assumed as follows: mass \( m = 1 \text{kg} \), damping factor \( c = 10 \text{ (Ns/m)} \), stiffness \( K = 10^6 \text{ (N/m)} \), original unbalance value of the rotor \( p = 10 \text{ (g-mm)} \), parameter of output filter
\[ \delta = 0.01 \], positive real number \( \lambda = 0.01 \), sampling time in the process of numerical simulation \( \Delta T = 0.001s \), and the other initial values are 0.

![Simulation model of the improved active balancing control](image)

Different running conditions are designed to validate the control performance of the proposed control strategy. Condition 1: the rotor will rotate with a constant acceleration of \( 80r/s^2 \) from 0 to 3 seconds. Condition 2: the rotor will rotate at a variable speed around 12000 rpm from 3 to 6 seconds. Condition 3: the rotor will rotate at a constant speed of 14400 rpm from 6 to 10 seconds. The simulation results are shown in Fig.8 and Table II.

![Simulation results of the improved active balancing control](image)

Fig.8. Simulation results of the improved active balancing control. (a) the initial vibration without control, (b) the vibration with adaptive control \( (\tau=0.0001s) \), (c) the vibration with adaptive control \( (\tau=0.002s) \), (d) the vibration with improved active control \( (\tau=0.0001s) \)

The simulation data of Fig.8 and Table II are analyzed as follows. Under three
conditions, the vibration with adaptive control in Fig.8(b) and Fig.8(c) is significantly reduced compared to the initial vibration without control in Fig.8(a). However, the balance performance is not ideal in the condition of uniform rotational speed. It is also seen that adjusting time is longer and vibration tends to increase from 6 to 10 seconds in Fig.8(b) and Fig.8(c). Comparison between Fig.8(b) and Fig.8(c) shows that the balancing control performance is obviously reduced with the change of time-delay $\tau$. Under the same conditions, the rotor system even shows instability with the increase of $\tau$. Contrasting Fig.8(b) with Fig.8(d), the vibration with improved active balancing control method decreases rapidly and becomes stable in 0.5s from 6 to 10 seconds. The specific simulation values of final vibration and adjusting time are shown in Table II.

| Condition | Initial vibration / (mm) | Final vibration / (mm) | Adjusting time / (sec.) | Final vibration / (mm) | Adjusting time / (sec.) | Final vibration / (mm) | Adjusting time / (sec.) |
|-----------|--------------------------|------------------------|------------------------|------------------------|------------------------|------------------------|------------------------|
| 1         | 11.2                     | 0.82                   | 1.8                    | 1.18                   | 2.2                    | 0.82                   | 1.8                    |
| 2         | 8.3                      | 0.10                   | 0.2                    | 0.19                   | 0.4                    | 0.10                   | 0.2                    |
| 3         | 6.5                      | 0.78                   | 3                      | 1.02                   | 3                      | 0.18                   | 0.5                    |

### 3.2 Experimental verification

In order to further verify the control method proposed, a test bench of the rotor system is built as shown in Fig.9.

![Physical diagram of balancing control system test bench](image)

1. Oil-air lubrication system  
2. High-speed spindle  
3. Actuator  
4. Vibration sensor  

A speed sensor monitors the spindle operation. To obtain radial displacement of the spindle, a vibration sensor is mounted at the bearing block above the center of spindle. The actuator is mounted at the end of the spindle to provide corrected unbalance mass. The control system outputs a signal to drive the actuator for unbalanced correction. The controlling chip DSP TMS320F2812 is used in this experiment. The structural design is adopted to the software of controller. Signals processing, adaptive controlling and PID controlling are programmed using C
language.

In the experiment, both improved active balancing control and traditional adaptive control are performed for comparison purpose. Five sets of experiments are carried out on the same initial conditions for the control methods, and we collect data every 0.5 seconds. The experimental data are shown in Table III and Table IV. The averages of the vibration values are plotted in Fig.10 and Fig.11.

| Table III. Experimental data in the process of uniform rotational speed |
|---------------------------------------------------------------|
|                                        | The vibration under improved active balancing control method(mm) | The vibration under traditional adaptive control method(mm) |
| Time/ (sec.)          | 1 | 2 | 3 | 4 | 5 | Average values | 1 | 2 | 3 | 4 | 5 | Average values |
| 0                     | 30.0 | 29.9 | 30.0 | 30.1 | 30.0 | 30.1 | 29.9 | 30.0 | 30.0 | 30.0 | 30.0 |
| 1                     | 23.6 | 23.5 | 23.7 | 23.2 | 23.5 | 23.5 | 23.8 | 27.9 | 27.8 | 28.2 | 28.0 | 28.0 |
| 2                     | 19.2 | 18.2 | 17.2 | 18.0 | 17.4 | 18.0 | 27.9 | 27.7 | 27.4 | 27.8 | 28.0 | 27.8 |
| 3                     | 14.8 | 15.4 | 14.5 | 14.8 | 14.2 | 14.8 | 27.7 | 27.6 | 27.3 | 28.0 | 27.2 | 27.6 |
| 4                     | 12.8 | 13.3 | 13.0 | 12.8 | 13.1 | 13.0 | 25.9 | 25.7 | 25.4 | 26.2 | 25.8 | 25.8 |
| 5                     | 12.1 | 12.0 | 11.9 | 12.0 | 13.1 | 12.0 | 25.1 | 24.9 | 25.0 | 25.0 | 25.0 | 25.0 |
| 6                     | 11.6 | 12.0 | 11.9 | 11.8 | 11.6 | 11.9 | 23.8 | 23.5 | 24.6 | 23.9 | 23.7 | 23.9 |
| 7                     | 11.0 | 10.9 | 11.2 | 11.1 | 10.8 | 11.0 | 22.8 | 22.6 | 22.6 | 22.8 | 22.9 | 22.5 |
| 8                     | 10.2 | 9.9  | 10.0 | 10.1 | 9.8  | 10.0 | 21.8 | 21.3 | 21.2 | 21.6 | 21.1 | 21.4 |
| 9                     | 8.8  | 8.9  | 9.0  | 8.9  | 8.9  | 8.9  | 21.2 | 21.0 | 21.1 | 21.1 | 21.1 | 21.1 |
| 10                    | 8.1  | 8.2  | 8.2  | 8.2  | 8.3  | 8.2  | 20.0 | 19.7 | 19.6 | 19.7 | 20.0 | 19.8 |
| 11                    | 7.9  | 8.0  | 7.8  | 8.0  | 7.9  | 7.9  | 18.8 | 18.7 | 18.8 | 18.5 | 18.7 | 18.7 |
| 12                    | 7.5  | 7.7  | 7.9  | 7.7  | 7.7  | 7.7  | 17.0 | 17.0 | 16.7 | 16.8 | 17.0 | 16.9 |
| 13                    | 7.2  | 7.5  | 7.9  | 7.5  | 7.4  | 7.5  | 15.6 | 15.7 | 15.2 | 15.7 | 15.3 | 15.5 |
| 14                    | 7.6  | 7.3  | 7.5  | 7.3  | 6.8  | 7.3  | 14.7 | 14.8 | 14.3 | 14.8 | 14.4 | 14.6 |
| 15                    | 7.2  | 7.4  | 7.2  | 7.0  | 7.2  | 7.2  | 13.6 | 13.4 | 13.5 | 13.5 | 13.5 | 13.5 |

Fig.10. Vibration in the process of constant rotational speed

From Fig.10 it can be seen that the improved active balancing control method can reduce the vibration by 50% in about 3 seconds and the vibration tends to be stable in about 11 seconds, while the traditional adaptive control method can only reduce the vibration by 50% with a lower efficiency in about 14 seconds.

After the rotor system reaches a stable equilibrium, rotational speed is suddenly increased intentionally. The vibration changes are shown in Fig.11. The vibration can be restored to a relatively stable state within about 3 seconds with the use of the improved active balancing control method, while the traditional adaptive control method fails to reach stable state even in 13 seconds. It can be seen that the improved adaptive active balance control method can stabilize sudden vibration change with much better efficiency.
4 Conclusions

In this paper, a dynamic model of rotor system with time-delay is constructed, and the stability of the model is analyzed. Then, an improved active balancing control method is proposed, which combines adaptive controller with PID controller so that the vibration of high-speed rotor system can be controlled in different operating conditions. Numerical simulation and experiments are used to illustrate that the proposed control method has a better balance performance than the traditional adaptive control method. The results also show that time-delay factor will reduce balanced control efficiency and lead to instability of the rotor system.

Conflict of Interests

The authors declare that there is no conflict of interests regarding the publication of this paper.

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