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Online control of critical speed vibrations of a single-span rotor by a rotor dynamic vibration absorber at different installation positions †

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Abstract

The serious vibration of rotors around the critical speed is a problem in rotor systems. To overcome this problem, a single-span two-disk rotor bench was built to simulate the starting process of a rotor. A new Rotor dynamic vibration absorber (RDVA) was designed and installed in the middle of the rotor. A on-off control method based on speed was applied to control the on-off position of the electromagnet in RDVA. Therefore, the natural frequencies between two selected values could be changed. The principles for the vibration control of the rotor system were studied. The vibration suppression performance of an RDVA is a function of its location. The location of the RDVA is subject to several constraints due to the compact structure of the rotor system. As a result, RDVA cannot always be installed at the optimal location of vibration suppression. Accordingly, a study was performed to observe the effect of RDVA location on the vibration suppression performance. Results showed that installing RDVA with on-off control between the two disks not only suppressed the violent vibrations of the rotor at critical speed during the starting process but also avoided the two resonance peaks generated by the traditional absorber. RDVA maintained the vibration of the rotor at a low level in the entire speed range. Furthermore, the vibrations of the rotor system decreased by 20 % when RDVA was installed near the rotor support.

Keywords: Rotor dynamic vibration absorber; Critical speed; On-off control; Single-span rotor; Installation position

1. Introduction

Rotating machinery often causes vibration during operation because of unbalanced mass. A high-speed rotor, whose operating speed is usually above the first-order critical speed, must pass the critical speed. In this case, the rotor has strong resonance, which affects its stable operation. Rotor balancing is commonly used to reduce critical vibrations. By balancing the machine, the unbalanced mass is reduced. Rotor balancing with a balancing machine requires the removal and transportation of the rotor. These operations involve a large amount of time, manpower, and cost. Field balancing of the rotor can also be performed, but it requires repeated on and off switching of the unit; this repeated switching results in severe economic loss [1-4]. An automatic balancing device is also used to control unbalanced rotor vibrations. The unbalanced mass of the rotor of fans and other rotating machinery gradually increases during operation due to scaling, corrosion, or other reasons, all of which cause violent vibration. An online automatic balancing device can be utilized to balance the rotor during its operation [5-9]. Electromagnetic and hydraulic automatic balancing devices are currently used. However, the structure of an automatic balancing device is complex. Furthermore, it is mostly used only in the machining field of grinders and machining centers because of its limited balancing ability and working conditions [10-14].

The application of a Dynamic vibration absorber (DVA) is an effective method for vibration control and suppressing structural vibrations [15-17]. A DVA generally consists of a mass, spring, and damper. This device connects the subsystem to the main system through elastic elements and generates inertial force from relative motion. The force created acts on the main system and suppresses its vibrations. Considering its excellent vibration control effect, DVA is widely used, especially in the control of structural vibrations caused by wind load or earthquakes [18]. The application of DVA for vibration control in rotating machinery has been widely studied. Nakano [19] investigated the effect of multiple dynamic absorbers on the chatter generated in end milling operations by performing a stability analysis and cutting test. Arrigan [20] used a semi-active tuning mass damper to effectively control the blade beat vibrations of a wind turbine. Campos [21] used a rotating vibration absorber to reduce lateral vibration during the spinning of a washing machine with a capacity of 8 kg. The results showed a significant reduction in the lateral vibra-

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tion of the machine basket when the absorber was used. The disadvantages of DVA are as follows: a new resonance peak is generated before or after the natural frequency and the effective damping band is narrow [22]. These drawbacks limit the wide application of DVA. Moreover, the location of DVA is subject to several constraints due to the compact structure of the rotor system. As a result, DVA cannot always be installed at the optimal location of vibration suppression. Therefore, the vibration suppression performance of DVA when installed in other positions in the rotor should be studied.

To overcome the inadequacies of rotor balancing techniques and DVA, a single-span two-disk rotor bench was established in this study, and a novel Rotor dynamic vibration absorber (RDVA) was designed and applied to the rotor vibration control system. The on-off control method based on speed was applied to study the principles of single-span rotor vibration control as the rotor passed through the critical speed during the starting process. The RDVA was installed in different positions in the rotor, and the effect of RDVA location on rotor vibration control performance was studied.

2. Control principle and structure of RDVA

2.1 Control principle of RDVA

The rotor system was discretized into a system with multiple degrees of freedom. The dynamical equation of the rotor system under the effect of unbalanced force can be expressed as [23]

$$\mathbf{M}\ddot{\mathbf{x}} + \mathbf{C}\dot{\mathbf{x}} + \mathbf{K}\mathbf{x} = \mathbf{F}e^{j\omega t}, \tag{1}$$

where $\boldsymbol{M}$, $\boldsymbol{C}$ and $\boldsymbol{K}$ are the mass, damping, and stiffness matrices of the rotor system, respectively. $\boldsymbol{C} = \boldsymbol{C}_0 + \boldsymbol{C}_\mathrm{C}$, $\boldsymbol{K} = \boldsymbol{K}_0 + \boldsymbol{K}_\mathrm{K}$, $\boldsymbol{C}_0$ is the transition damping matrix. $\boldsymbol{C}_\mathrm{C}$ is the entire coupled damping matrix. $\boldsymbol{K}_0$ is the transition stiffness matrix, and $\boldsymbol{K}_\mathrm{K}$ is the entire coupled stiffness matrix. $\ddot{\mathbf{x}}$, $\dot{\mathbf{x}}$ and $\boldsymbol{x}$ are the acceleration, velocity, and displacement matrices of the main rotor system, respectively. $\boldsymbol{F}$ is the unbalanced force matrix of the rotor. ω is the frequency of the exciting force. The response of the rotor under steady-state unbalanced excitation is

$$\mathbf{x} = \mathbf{X}e^{j\omega t}, \tag{2}$$

where $\boldsymbol{X}$ is the amplitude matrix of the rotor. Substituting Eq. (2) into Eq. (1) results in

$$(-\omega^2\mathbf{M} + j\omega\mathbf{C} + \mathbf{K})\mathbf{X} = \mathbf{F}. \tag{3}$$

Modal matrix $\boldsymbol{\Phi}$, which consists of inherent vectors originating from the analytical intrinsic values, is as follows:

$$\Phi = [\mathbf{X}_1 \quad \mathbf{X}_2 \quad \cdots \quad \mathbf{X}_n] = \begin{bmatrix} \varphi_{11} & \varphi_{12} & \cdots & \varphi_{1n} \\ \varphi_{21} & \varphi_{22} & \cdots & \varphi_{2n} \\ \vdots & \vdots & \ddots & \vdots \\ \varphi_{n1} & \varphi_{n2} & \cdots & \varphi_{nn} \end{bmatrix}. \tag{4}$$

This modal matrix was used for coordinate transformation between the physical and modal coordinate systems.

$$\mathbf{X} = \Phi\delta. \tag{5}$$

Substituting Eq. (5) into Eq. (3) and multiplying both sides with $\boldsymbol{\Phi}^\mathrm{T}$ results in

$$(\Phi^\mathrm{T}\mathbf{G}\Phi)\delta = \Phi^\mathrm{T}\mathbf{F}, \tag{6}$$

where

$$\mathbf{\Phi}^\mathrm{T}\mathbf{G}\mathbf{\Phi} = -\omega^2\mathbf{\Phi}^\mathrm{T}\mathbf{M}\mathbf{\Phi} + j\omega\mathbf{\Phi}^\mathrm{T}\mathbf{C}\mathbf{\Phi} + \mathbf{\Phi}^\mathrm{T}\mathbf{K}\mathbf{\Phi}.$$

When the vibration absorber acts on the rotor, the control force vector of DVA is $\boldsymbol{F}_d$. Therefore, the dynamic equation of the rotor system with the dynamic vibration absorber device can be divided as

$$(\Phi^\mathrm{T}\mathbf{G}\Phi)\delta = \Phi^\mathrm{T}(\mathbf{F} - \mathbf{F}_\mathrm{d}). \tag{7}$$

The force F_d of DVA that acts on the rotor can be expressed as

$$F_d = \frac{-m\omega^2(k + jc\omega)}{k - m\omega^2 + jc\omega}\delta_i, \tag{8}$$

where m, k, and c are the mass, stiffness, and damping coefficients, respectively, of DVA.

Therefore, the transfer function of the rotor system with DVA can be expressed as follows:

$$G(\omega) = [1 - (\frac{\omega}{\omega_n})^2 + j2\xi_n(\frac{\omega}{\omega_n})]\cdot\frac{1}{K_i} / \{(\frac{\omega}{\Omega_i})^2(\frac{\omega}{\omega_n})^2 - [(\frac{\omega}{\omega_n})^2 + (\frac{\omega}{\Omega_i})^2(1+\mu)] + 1 + j2\zeta_n(\frac{\omega}{\omega_n})[1 - (\frac{\omega}{\Omega_i})^2(1+\mu)]\}, \tag{9}$$

where

$$\Omega_i = \sqrt{\frac{K_i}{M_i}}, \quad \omega_n = \sqrt{\frac{k}{m}}, \quad \zeta_n = \frac{c_n}{2m\omega_n}.$$

Ω_i, ω_n and ζ_n are the natural frequencies of the rotor and DVA and the damping ratio of DVA, respectively. μ is the mass ratio between the absorber and rotor system. The steady-state response of the rotor is

$$\mathbf{X} = \sum_{i=1}^{n}\Phi_i^\mathrm{T}\mathbf{F}\Phi_i G_i(\omega). \tag{10}$$

Fig. 1 shows the vibration control effect of DVA on the rotor at different damping ratios of DVA when the mass ratio

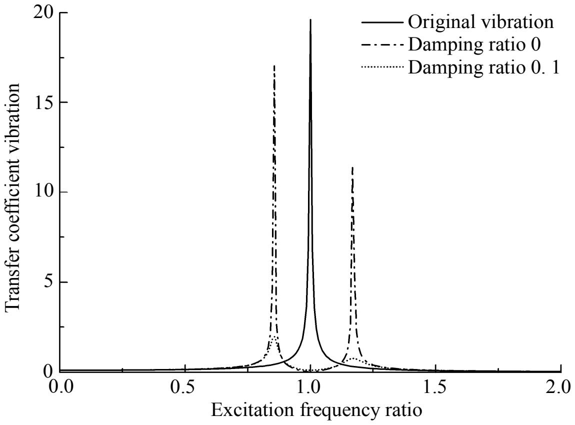

Fig. 1. Vibration control effect of DVA under different damping ratios.

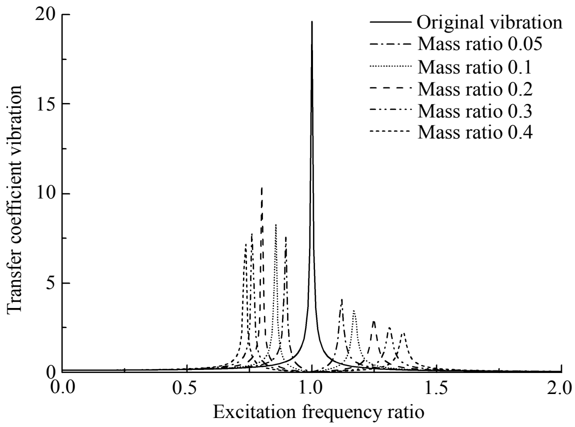

Fig. 2. Vibration control effect of DVA under different damping ratios.

is 0.1. When the natural frequency of DVA and the frequency of rotor excitation force are equal, the inertial force from the relative movement acting on the rotor and the vibration amplitude of the rotor system decrease substantially. However, two new resonance peaks of resonance frequencies appear simultaneously before and after the natural frequency. The amplitudes of the peaks are small when the damping ratio is large. The frequencies of the two peaks do not change.

Fig. 2 shows the vibration control effect of DVA on the rotor at different mass ratios when the damping ratio of DVA is 0.01. With the increase in mass ratio, the frequency interval of the two new peaks becomes large, and the frequency bandwidth of vibration control widens. However, a large mass ratio would increase the weight load of the rotor.

The traditional dynamic vibration absorber is likely to increase the rotor vibrations when it deviates from the tuning frequency and would generate two new resonance peaks. Therefore, the traditional dynamic vibration absorber is unsuitable for rotor vibration control.

To overcome the disadvantages of the traditional dynamic vibration absorber, a new type of RDVA was designed to control the vibrations of a single-span rotor system.

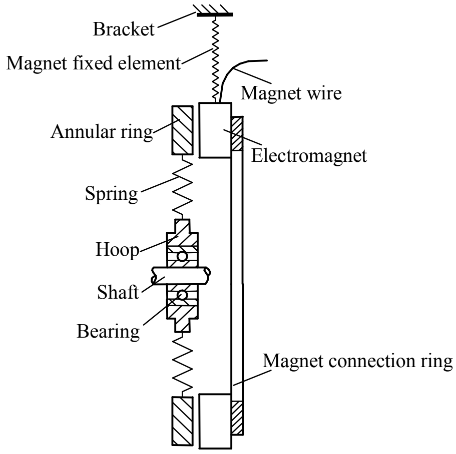

Fig. 3. Structure of RDVA.

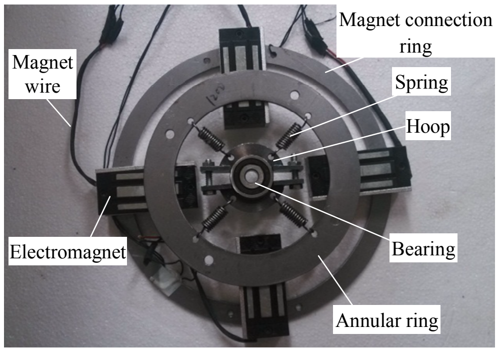

Fig. 4. Photo of RDVA.

2.2 Structure of RDVA

The designed RDVA was based on the structure of the rotor test bench and critical vibration characteristics (shown in Figs. 3 and 4). The mass ratio of RDVA was 0.1. RDVA consisted of connection, spring, and mass units. The connection unit consisted of a connecting bearing with adjustment for axial positioning and a hoop. The hoop was fitted over the bearing outer ring to transmit the rotor vibrations. However, the hoop itself did not rotate. The spring unit consisted of four springs with the same stiffness distributed uniformly on the circumference. The mass unit consisted of the less massive annular ring (m_0) and electromagnet suspended from the bracket (m_1). The electromagnet picked up the annular ring when it was energized, and it disengaged from the annular ring when the power was turned off.

2.3 Vibration control principle of RDVA

The RDVA effectively supressed the rotor vibrations when the natural frequency of RDVA and the frequency of exciting force were equal. The natural frequency of RDVA depended on the mass of the mass unit and the stiffness of the spring unit. The frequency of RDVA is

$$f = \frac{1}{2\pi}\sqrt{\frac{k}{m}}, \tag{11}$$

where m is the mass of the mass unit and k is the stiffness of the spring unit.

In this study, the spring stiffness of RDVA remained unchanged. The natural frequency was changed by changing the mass of the mass unit. The mass of the mass unit was m_0 and increased to $m_0 + m_1$ by controlling the on-off status of the electromagnet. The natural frequency of the RDVA with the annular ring of a mass of m_0 is given by $f_0 = \sqrt{k/m_0}/(2\pi)$. To make the two resonance peaks of the frequencies generated by the RDVA higher than the rotor operating speed, f_0 was maintained higher than the frequency corresponding to the rotor operating speed. Thus, in the entire range of rotor operating speeds, this RDVA was prevented from generating two new resonance peaks to increase rotor vibration. When the annular ring m_0 picked up the electromagnet m_1 in the rotor critical resonance region, the natural frequency of RDVA is given by $f_1 = \sqrt{k/(m_0 + m_1)}/(2\pi)$. f_1 remained consistent with the frequency corresponding to the rotor critical speed.

During the operation, the rotor vibration was small when the rotor speed was in the noncritical speed resonance region. The electromagent was under power outage and disengaged the annular ring. The RDVA consisting of an annular ring had no vibration control effect on the rotor and did not generate new resonance peaks to increase the rotor vibration. The rotor vibration was large when the rotor speed was in the critical speed resonance region. The on-off control method based on speed was applied to control the on-off status of the electromagnet; thus, the electromagnet picked up the annular ring, and the RDVA exerted a vibration control effect. The on-off control method based on speed for RDVA supressed the rotor vibration at the critical speed and prevented RDVA from generating two new resonance peaks.

3. Experimental studies on single-span rotor vibration control

3.1 Single-span rotor bench

Figs. 5 and 6 show the single-span two-disk rotor bench. The rotor is belt driven. The speed of the drive motor was adjusted by a speed controller, and the rotor speed was varied from 0 rpm to 1800 rpm. The diameter of the rotor was 15 mm, and the bearing span was 600 mm. The rotor had two disks. The weight of each disk was 6.89 kg, the diameter of the disk was 200 mm, and the thickness of the disk was 30 mm.

The experimental test system consisted of an LC-8000 multichannel vibration monitoring fault diagnostic system equipped with eight input channels and a specialized vibration signal processing acquisition board. The measuring support frame for horizontal vibration acceleration was selected for this experiment.

Table 1. Structural parameters of RDVA.

Annular ring mass (m_0) (g)	180
Electromagnet mass (m_1) (g)	1250
Frequency of m_0 and m_1 (Hz)	20
Frequency of m_0 (Hz)	56.4
Single spring stiffness (N/mm)	7.98

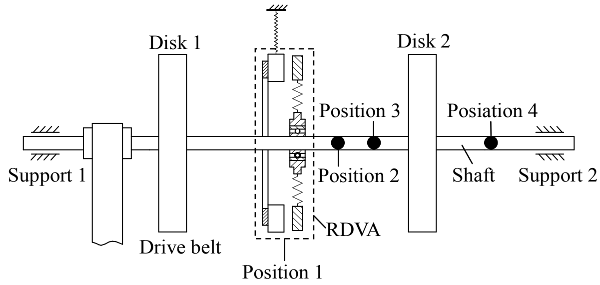

Fig. 5. Structure of the single-span rotor bench.

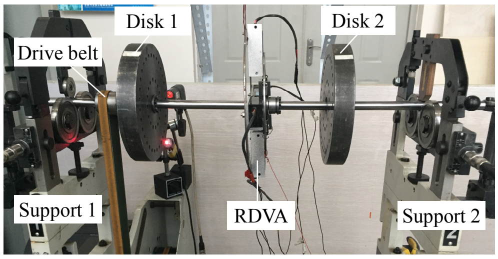

Fig. 6. Photo of the single-span rotor bench.

3.2 Experimental study on the principle of single-span rotor vibration control based on RDVA

When the single-span rotor's speed was increased from 0 rpm to 1800 rpm, the critical speed was 1196 rpm. The vibration of the rotor rapidly increased because of the unbalanced mass at the critical speed in the starting process. A new version of RDVA was designed and installed on the rotor to study the principles of rotor vibration control and develop methods to suppress violent rotor vibrations at the critical speed. Moreover, the vibration suppression performance of the RDVA when the RDVA was installed in different positions in the rotor was compared.

3.2.1 Parameters of RDVA

To suppress the vibration of the rotor at the critical speed under on-off conditions, an appropriate mass unit that consisted of an annular ring and electromagnets was selected. The stiffness of the spring could be obtained from the natural frequency (Eq. (11)) of RDVA. The structural parameters of RDVA are shown in Table 1.

First, the annular ring of RDVA was installed on the rotor with bearings, hoops, and springs. When the electromagnet was not energized, the natural frequency of the RDVA consisting of an annular ring only was calculated with Eq. (11) as 56.4 Hz. The corresponding speed was 3384 rpm. The speed range of the first peak of the two resonance peaks generated by the absorber was 3200 rpm to 3300 rpm. This speed was much higher than the maximum rotor speed of 1800 rpm. Therefore, the RDVA consisting of an annular ring only installed on the rotor did not generate new resonance peaks to increase the vibration of the rotor.

Based on these results, the on-off control method based on speed was applied to control the on-off condition of the electromagnet using a controllable property of RDVA in the rotor critical resonance region to suppress violent rotor vibration at the critical speed. The electromagnet was not energized in the noncritical resonance region; therefore, the RDVA did not increase nor inhibit rotor vibration.

3.2.2 Experimental study on the rotor with and without on-off control

The effect of RDVA with or without on-off control on rotor vibration control was investigated. Without on-off control corresponded to the condition of the electromagnet absorbing and holding the annular ring during the speeding up of the rotor. With on-off control corresponded to the condition of the electromagnet picking up an annular ring only in the rotor critical resonance region.

When the rotor was equipped with the RDVA without on-off control, the electromagnet always absorbed an annular ring. The natural frequency of RDVA was consistent with the rotor critical speed corresponding frequency. The violent rotor vibration was effectively suppressed in the critical resonance region from 1080 rpm to 1260 rpm, as shown in Fig. 7. However, in the speed range of 900 rpm to 1020 rpm, the rotor system generated a new peak (11.2 m/s^2), and a second peak (27.2 m/s^2) was generated in the speed range of 1480 rpm to 1590 rpm. Therefore, the electromagnet always absorbing an annular ring generated two new peaks.

When the rotor was equipped with RDVA with on-off control, the electromagnet in RDVA absorbed and held the annular ring in the resonant critical speed region of 1080 rpm to 1260 rpm. In this speed range, the natural frequency of RDVA was close to or consistent with the frequency corresponding to the rotor speed, and the rotor critical resonance region was within the effective vibration control bandwidth of RDVA. The rotor critical vibration was effectively suppressed by the on-off control. The maximum amplitude at the measuring point decreased from 30.7 m/s^2 to 3.5 m/s^2. This decrease amounted to 88.6 %. The electromagnet disengaged the annular ring in the noncritical areas.

Therefore, the RDVA with on-off control suppressed the violent vibration of the single-span rotor in the critical resonant region during the speeding up process and avoided the generation of two new resonance peaks. The vibration of the single-span rotor remained at a low level in the entire process of speeding up, as shown in Fig. 7.

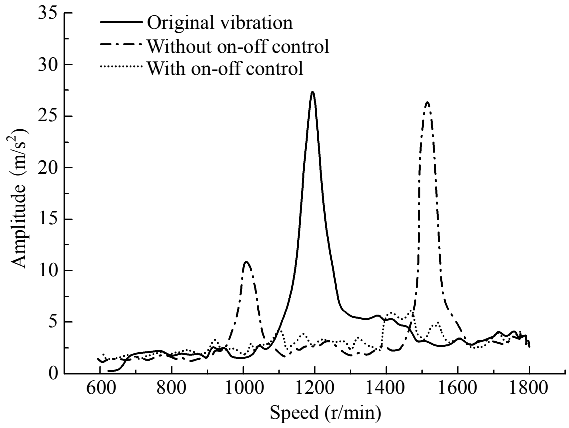

Fig. 7. Vibration of the single-span rotor during speeding up.

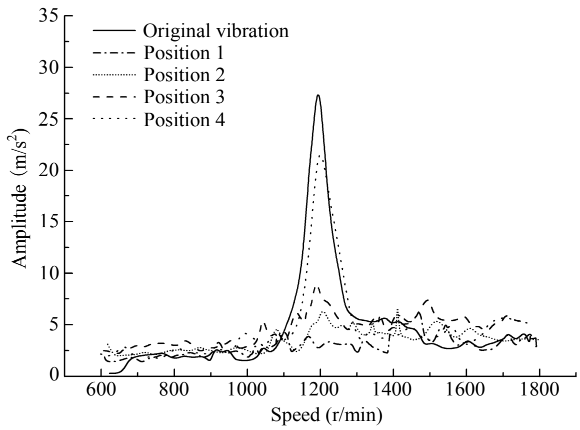

Fig. 8. Vibration of the rotor at different locations of RDVA.

3.2.3 Experimental study on the installation positions of the RDVA for the rotor system

The vibration suppression effect was considerable when the RDVA was installed in the middle of the rotor. However, the actual structure of the rotor system was compact, and the installation space was limited. As a result, the installation of RDVA may be impossible at position 1 (indicated in Fig. 5). An alternative position needs to be selected. Therefore, the RDVA was installed at the three other positions, namely, (i) between the two disks at 50 mm (Position 2) from one disk, (ii) between the two disks at 100 mm (Position 3) from one disk, and (iii) near the rotor support (Position 4), as shown in Fig. 5. The effect of the installation position of the RDVA on the vibration suppression performance was studied. The vibration data are presented in Fig. 8.

When the RDVA was installed at position 2, the rotor vibration amplitude in the critical resonance region decreased by 77.9 % to a value of 6.9 m/s^2. Similarly, when the RDVA was installed at position 3, the amplitude decreased by 67.1 % to 10.1 m/s^2. The reduction in amplitude was lower than the one corresponding to the RDVA installation at position 1, but the magnitude of vibration was significantly reduced compared with the original vibrations. Hence, the vibration of the single-span two-disk rotor system was suppressed effectively by installing the RDVA between the two disks. When the RDVA was near the support (Position 4), the magnitude of vibration was 24.3 m/s^2, which corresponded to a 20.8 % reduction in the amplitude of vibration. In addition, the vibration suppression performance was reduced when the position of the RDVA was close to the support.

4. Conclusions

(1) The on-off control method based on speed can control the on-off position of the electromagnet in RDVA to change the natural frequencies of RDVA between two selected values.

(2) The RDVA with on-off control can eliminate the two amplitude peaks of traditional DVA, thereby reducing the negative effects of such devices on rotor vibration control.

(3) The RDVA effectively suppressed the violent vibration of the single-span rotor at speed during the starting process, such that the single-span rotor vibration remained at a low level.

(4) The vibration of the single-span two-disk rotor system was suppressed effectively by installing an RDVA between the two disks. The vibration suppression performance was reduced when the position of the RDVA was close to the support. However, the vibrations of the rotor system still underwent a reduction of around 20 %.

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Nomenclature

$\boldsymbol{M}$: Mass matrix of the rotor system
$\boldsymbol{C}$: Damping matrix of the rotor system
$\boldsymbol{K}$: Stiffness matrix of the rotor system
$\boldsymbol{C}_0$: Transition damping matrix of the rotor system
$\boldsymbol{C}_\mathrm{C}$: Entire coupled damping matrix of the rotor system
$\boldsymbol{K}_0$: Transition stiffness matrix of the rotor system
$\boldsymbol{K}_\mathrm{K}$: Entire coupled stiffness matrix of the rotor system
$\ddot{\mathbf{x}}$: Acceleration matrix of the rotor system
$\dot{\mathbf{x}}$: Velocity matrix of the rotor system
$\boldsymbol{x}$: Displacement matrix of the rotor system
$\boldsymbol{F}$: Unbalanced force matrix of the rotor
ω : Frequency of exciting force
$\boldsymbol{X}$: Amplitude matrix of the rotor
$\boldsymbol{\Phi}$: Modal matrix
$\boldsymbol{F_d}$: Control force vector of DVA
$G(\omega)$: Transfer function
Ω_i : Natural frequency of the rotor
ω_n : Natural frequency of DVA
ζ_n : Damping ratio of DVA
μ : Mass ratio between the absorber and rotor system

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