Research on the Beat Vibration of Active Magnetic Bearings

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Abstract

Beat vibration is a kind of special vibration which sometimes arise in the active magnetic bearing. This vibration can smear the quality of magnetic bearings seriously. Based on the analysis of the synthesis mechanism of the beat vibration, this paper investigated the origin of the beat vibration in the magnetic bearing. According to the special electrical and machinery structure of the magnetic bearing, the source and the formation of the two sub-vibrations (the vibration with the same frequency of the rotate speed of the rotor and the vibration with the same frequency of the rotating magnetic field in induction motor) which synthesize the beat vibration were analyzed. Finally, method of inhibition of the beat vibration is given.

1 Introduction

Magnetic bearings are widely used in the field of precise machining, turbine machinery, super clean room and so on, because of a series of its advantages[1-3]. In most of its applications, high suspension accuracy is needed. However, in the practical application of magnetic bearings, the beat vibration of the rotor is sometimes observed.

Beat vibration is a vibration with periodically varying amplitude, which has a sinusoidal envelope. The existence of the beat vibration seriously deteriorates the magnetic bearing suspension accuracy, sometimes accompanied by the emergence of periodic noise[4-5]. To explore the mechanism of beat vibration in magnetic bearings and take measures to suppress and eliminate it is an important issue in the high-precision application field of magnetic bearings. Some scholars have researched on the asynchronous motor vibration phenomenon [6-9], but due to the special electrical and mechanical structure of magnetic bearings, the traditional beat vibration theory cannot explain the beat phenomenon in the magnetic levitation bearings. Therefore, researching on the magnetic bearing beat vibration has important theoretical and practical significance[10-11] .This paper firstly discusses the synthesis mechanism of beat vibrations, and then focus on the generation of the two sub-vibrations which compose the beat vibration. Finally the methods of inhibition of the vibration are given.

2 Generation principle of the beat vibration in magnetic bearings

The magnetic bearing rotor was controlled by 5 pairs of magnetic bearings and suspended at the set position, driven by the embedded asynchronous motor to achieve the desired high speed[12-13]. Theoretical analysis and experiments show that there are two prominent sub-vibrations with a close vibration frequency when beat vibrations occur[14]: one is the vibration with the same frequency of the rotor rotation speed and the other with the same frequency of the

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Supported by National Natural Science Foundation of China (50775129), Natural Science Foundation of Shandong province(Y2008F44), Shandong Province Excellent Young Scientists Award Fund(BS2011ZZ009).

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motor rotating magnetic field. When the rotor is suspended and motor operates at 350Hz, rotor vibration signal collected is shown in Figure 1. Do FFT analysis to obtain its spectrum, which is shown in Figure2. Figure 2 (a) is the 0 - 500Hz frequency spectrum and Fig 2 (b) is the 345 - 355Hz local frequency spectrum.

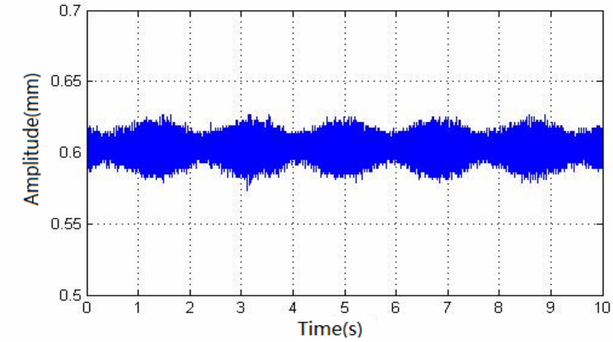


Figure 1: Beat vibration of the magnetic bearing rotor

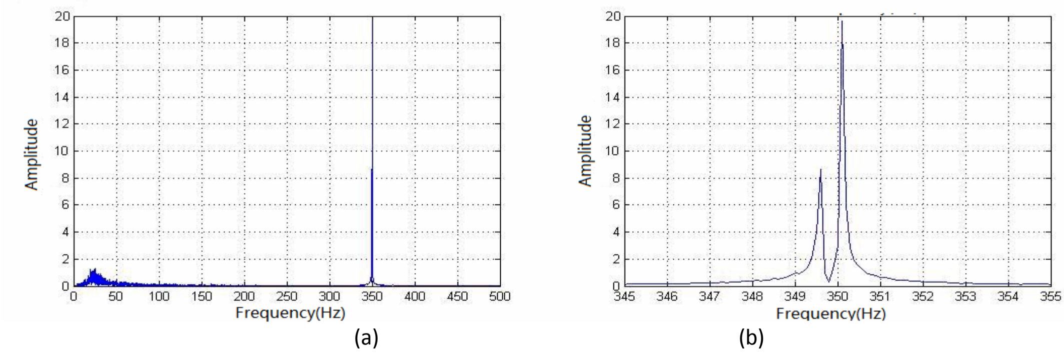


Figure.2 Frequency spectrum of the magnetic bearing vibration

In Figure 2, there are two obvious peaks in the rotor vibration spectrum at about 350Hz. One peak with the amplitude of A_1 at 350Hz corresponds to the inverter set frequency ω_1, i.e. the rotating frequency of the rotating magnetic field in the motor . Another peak with the amplitude of A_2 corresponds to the induction motor rotor rotation frequency ω_2. Electric motor theory shows that these two frequencies are very close to each other and the rotation frequency of the rotating magnetic field ω_1 is greater than the rotation frequency of the rotor ω_2. Select the appropriate time reference point to let the two vibrations with the opposite initial phase, then the two vibration equations are:

$$\begin{aligned} V_1 &= A_1 \cos(\omega_1 t - \theta) \\ V_2 &= A_2 \cos(\omega_2 t + \theta) \end{aligned} \tag{1}$$

The superposition of these two vibrations is:

$$\begin{aligned} V &= A_1 \cos(\omega_1 t - \theta) + A_2 \cos(\omega_2 t + \theta) \\ &= \cos(\frac{\omega_1 - \omega_2}{2} t - \theta)\left[(A_1 + A_2)\cos\frac{\omega_1 + \omega_2}{2} t \right] \\ &+ \sin(\frac{\omega_1 - \omega_2}{2} t - \theta)\left[-(A_1 - A_2)\sin\frac{\omega_1 + \omega_2}{2} t \right] \end{aligned} \tag{2}$$

In formula (2), the synthesized vibration V has two components, in the first component $\cos(\frac{\omega_1 - \omega_2}{2} t - \theta)\left[(A_1 + A_2)\cos\frac{\omega_1 + \omega_2}{2} t \right]$, $(A_1 + A_2)\cos\frac{\omega_1 + \omega_2}{2} t$ is high frequency vibration, whose amplitude is the sum of the amplitude of the two sub-vibrations of V_1 and V_2, and the frequency is half of the sum of the frequency of V_1 and V_2. $\cos(\frac{\omega_1-\omega_2}{2}t-\theta)\left[(A_1+A_2)\cos\frac{\omega_1+\omega_2}{2}t\right]$ is the high-frequency vibration envelope. This high frequency vibration with envelope is shown in the upper waveform in Figure 3. In the second component $\sin(\frac{\omega_1-\omega_2}{2}t-\theta)\left[-(A_1-A_2)\sin\frac{\omega_1+\omega_2}{2}t\right]$, $-(A_1-A_2)\sin\frac{\omega_1+\omega_2}{2}t$ is the high frequency vibration, whose amplitude is the difference of the amplitudes of the two sub-frequency of V_1 and V_2, and frequency is the half of the sum of the frequency of V_1 and V_2. $\sin(\frac{\omega_1-\omega_2}{2}t-\theta)\left[-(A_1-A_2)\sin\frac{\omega_1+\omega_2}{2}t\right]$ is the high-frequency vibration envelope. This high frequency vibration with envelope is shown in the middle waveform in Fig3. The phase difference of the two sub-vibration's envelope is 90°, and their synthesis vibration V shown as below waveform in Figure 3. The frequency of V is the half of the sum of the frequencies of V_1 and V_2, the amplitude of V changed cyclically, the maximum amplitude is the sum of the amplitude of V_1 and V_2, and the minimum amplitude is the difference of the amplitude of V_1 and V_2, the amplitude change period is 2 times the frequency of the envelope line, namely, the difference frequency difference of V_1 and V_2.

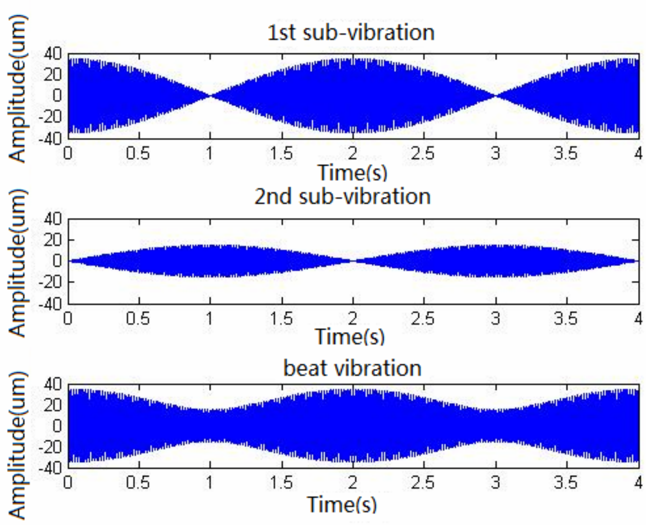


Figure.3 Waveform of the synthesis mechanism of beat vibration

In Figure 2, two peaks correspond to the frequencies of 350Hz and 349.45Hz. The superposition of two time domain vibration corresponding to these two peaks is the beat vibration shown in Figure 1. Beating Vibration envelope line frequency f_b is (350-349.45) / 2 = 0.275Hz, and period T_b is 1/0.275 = 3.636s. Beat vibration period $T_p = 0.5T_b$ = 1.818s.

3 Analysis of the source of the beat vibration

3.1. The formation of the vibration with the same frequency of rotating magnetic field

In asynchronous motor, as long as the rotor has eccentric relative to the stator, there will be the unilateral magnetic force. Set a uniform air gap δ of the motor, at a moment rotor has eccentricity e, take any pair of polar in a diameter of the motor, the angle of the eccentric direction and the diameter is θ. The unilateral magnetic force of the imbalance in the pole is[15]:

$$F_{M\theta} = \frac{2A_P B_\delta^2 e \cos\theta}{\mu_0 \delta} \tag{3}$$

The projection of the magnetic force in the direction of the eccentric is $F_{M\theta}\cos\theta$. If the motor has a total of p pare of the pole, the total unbalanced magnetic force in eccentric direction is:

$$F_M = p\left[\frac{1}{\pi}\int_{-\frac{\pi}{2}}^{\frac{\pi}{2}} F_{M\theta}\cos\theta d\theta\right] = \frac{pA_P B_\delta^2 e}{\mu_0 \delta} \tag{4}$$

In the process of a turn revolution in the air gap magnetic field , static eccentric rotor's (eccentric position unchanged) unilateral unbalanced magnetic force alternates twice , which results in the rotor vibration with the frequency of 2 times of the rotating magnetic field, which was mentioned in lots of literatures. But in magnetic bearings, the position of the rotor can be moved , then the vibration mode of the rotor will change .

Set the air gap rotating magnetic field flux density $B_\delta = B\sin\omega t$, using the PD control in magnetic bearings, combing the unilateral magnetic force and magnetic bearings control force, the joint force of the rotor (ignoring gravity) is :

$$\begin{aligned} F &= \frac{pA_P e B^2}{2\mu_0\delta}(1-\cos 2\omega t) - K_P e - K_D \dot{e} \\ &= K_M e - K_M e\cos 2\omega t - K_P e - K_D \dot{e} \end{aligned} \tag{5}$$

Where

$$K_M = \frac{pA_P B^2}{2\mu_0\delta} \tag{6}$$

Set the rotor equivalent mass m, according to Newton's second law of motion $F = ma = m\ddot{e}$ and substitutes it into equation (5)

$$m\ddot{e} = K_M e - K_M e\cos 2\omega t - K_P e - K_D \dot{e} \tag{7}$$

i.e.:

$$m\ddot{e} + K_D\dot{e} + (K_P - K_M + K_M\cos 2\omega t)e = 0 \tag{8}$$

Formula (8) is a time-varying parameters, second-order differential equation, its analytical solution cannot be obtained, now Runge-Kutta method is used to obtain its numerical solution.

In PD control algorithm, take the stiffness of K_P with natural stiffness, damping K_D with natural damping, substitutes magnetic bearings parameters into equation (8) , set the initial speed of the rotor with 15m/sec , initial displacements 0 , rotor vibration displacement curve can be obtained in Figure 4 (a), where the curve 1 is the asynchronous motor rotating magnetic field curve , curve 2 is the rotor vibration . In this figure, rotor vibration with the same frequency of the rotating magnetic field can be clearly seen. Because of the rotor's inertia, the phase rotor vibration lags after the phase of the rotating magnetic field.

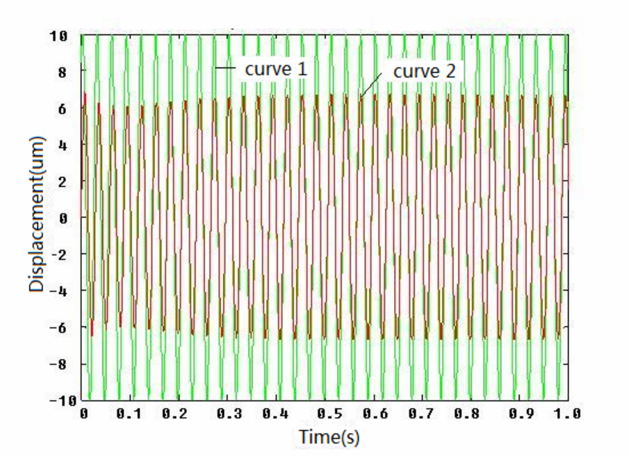


Figure.4 Rotating magnetic field frequency vibration waveform obtained using Runge-Kutta algorithm

Set the eccentric displacement of the rotor $e = S\sin(\omega t)$, air gap rotating magnetic field $B_\delta = B\sin(\omega t+\theta)$, where $\theta \le 90^\circ$ is the phase difference between the rotating magnetic filed and the rotor vibration. Then the rotor's unilateral magnetic force is:

$$\begin{aligned} F_M &= \frac{pA_P B_\delta^2 e}{\mu_0\delta} = \frac{pA_P}{\mu_0\delta}B^2\sin^2(\omega t+\theta)S\sin(\omega t) \\ &= \frac{pA_P}{2\mu_0\delta}B^2 S\sin(\omega t)[1-\cos(2\omega t+2\theta)] \end{aligned} \tag{9}$$

Speed of rotor vibration is:

$$v = \frac{de}{dt} = S\omega\cos(\omega t) \tag{10}$$

Power of the unilateral magnetic force acting on the rotor is:

$$\begin{aligned} p_M &= F_M v \\ &= \frac{pA_P}{2\mu_0\delta}B^2 S\sin(\omega t)[1-\cos(2\omega t+2\theta)]S\omega\cos(\omega t) \\ &= \frac{pA_P B^2 S^2\omega}{4\mu_0\delta}[\sin(2\omega t)-\sin(2\omega t)\cos(2\omega t+2\theta)] \end{aligned} \tag{11}$$

When the magnetic field rotates a round, the power of unilateral magnetic force acting the rotor is:

$$\begin{aligned} W_M &= \frac{pA_P B^2 S^2\omega}{4\mu_0\delta}\int_0^{\frac{2\pi}{\omega}}[\sin(2\omega t)-\sin(2\omega t)\cos(2\omega t+2\theta)]dt \\ &= \frac{pA_P B^2 S^2\omega}{4\mu_0\delta}\int_0^{\frac{2\pi}{\omega}}\sin(2\theta)dt \\ &= \frac{pA_P B^2 S^2\pi}{2\mu_0\delta}\sin(2\theta) \end{aligned} \tag{12}$$

The negative work of damping force acting on the rotor is:

$$\begin{aligned} W_P &= -\int_0^{\frac{2\pi}{\omega}}\left|K_D v v\right|dt = -4\int_0^{\frac{\pi}{2\omega}}K_D S^2\omega^2\cos^2(\omega t)dt \\ &= -K_D\pi\omega S^2 \end{aligned} \tag{13}$$

In formula (12) , because of $W_M = \frac{pA_P B^2 S^2\pi}{2\mu_0\delta}\sin(2\theta) > 0$, so as long as the vibration with the same frequency of rotating magnetic field generates, it will gradually set up automatically .

From Figure 4 can be found that, in the initial stage of the vibration with the same frequency of rotating magnetic field, the rotor vibration phase lag behind the rotating magnetic field a large scale, the work done by the unilateral magnetic force on the rotor is larger, so the same frequency vibration has been strengthened. With the passage of time, the phase difference becomes smaller and smaller, so that the same frequency of vibration can be maintained.

3.2. Generation of the vibration with the same frequency of Rotor rotation speed

The reason of the vibration with the same frequency of rotor rotation speed is that rotor itself has eccentric, i.e., there is a deviation from the geometric position on the rotor geometric center W and the rotor center of mass S , also known as the eccentricity.

Set the rotation angle speed of the rotor frequency as Ω , eccentricity as e, then the motion equations of the rotor geometric center W is:

$$\begin{bmatrix} \ddot{x}_w \\ \ddot{y}_w \end{bmatrix} + \begin{bmatrix} \omega^2 & 0 \\ 0 & \omega^2 \end{bmatrix} \begin{bmatrix} x_w \\ y_w \end{bmatrix} = \begin{bmatrix} e\Omega^2 \cos\Omega t \\ e\Omega^2 \sin\Omega t \end{bmatrix}, \omega = \sqrt{\frac{k}{m}} \tag{14}$$

Where k and m are the stiffness of magnetic bearings and the stiffness of the rotor equivalent quality respectively.

The solution of the equation is:

$$\begin{aligned} x_w(t) &= C\cos\Omega t \\ y_w(t) &= S\sin\Omega t \end{aligned} \tag{15}$$

Substituted it into equation (14) , the amplitude of the vibration is:

$$C(\Omega) = S(\Omega) = e\frac{\Omega^2}{\omega^2 - \Omega^2} \tag{16}$$

The rotor center of gravity W motion in a circular trajectory as the angular velocity Ω , trajectory radius is:

$$r_w(\Omega) = \sqrt{x_w^2 + y_w^2} = e\frac{\Omega^2}{\omega^2 - \Omega^2} \tag{17}$$

4 Measures to suppress beat vibrations

Beat vibration is generated by the composition of the two sub-vibrations; therefore, in order to suppress the beat vibration, the two sub-vibration must be inhibited.
(1) According to formula (17) , trying to do the good balancing of the rotor and reducing the rotor eccentricity can proportionally reduce the vibration of the same frequency with rotation frequency. In addition, when the speed of the rotor pass over the critical speed, improving the speed of the rotor can also reduce the same frequency vibration.
(2) According to formula (13) , increasing the speed of the rotor can increase the negative work done by the damping of the rotor , thereby suppressing the vibration with the same frequency of rotating magnetic field.
(3) According to formula (8) , the dynamic effects of the unilateral magnetic force is actually dynamically change the magnetic bearing stiffness, in the case of the stability of the magnetic levitation , increasing the stiffness coefficient in the control parameters can effectively inhibit the vibration with the same frequency of rotation magnetic field.
(4) Design multi-channel PID controller, isolate two vibration signals which caused the beat vibration from the vibration signal, and inhibit them particularly.

5 Experimental result

According to the measures to suppress beat vibration in the previous section, the following relevant experiments are done[16-18].

(1) Change the inverter frequency to observe the change of the beat vibration spectrum:
The converter frequency gradually increased from 200Hz to 400Hz, the data and curve of two sub-vibration spectrum peaks versus frequency are shown in Table 1 and Figure 5 (a).
It can be seen from Figure 5 (a), two sub-vibrations descend with the speed increase of the rotor, particularly the vibration with the same frequency of rotating magnetic field. It can verify that the increasing speed is a viable method to reduce beat vibrations.

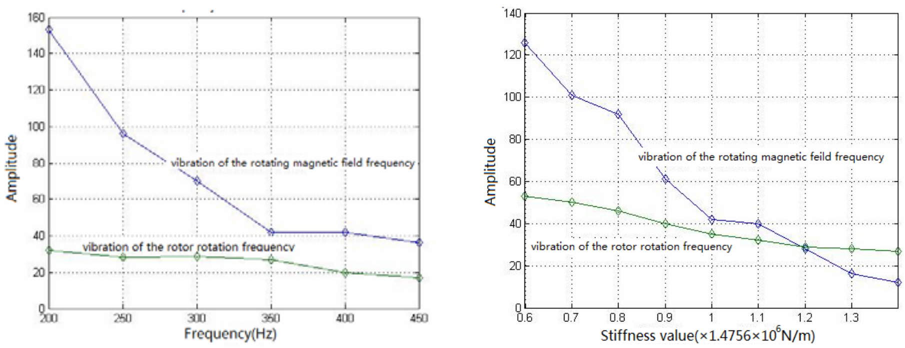
(2) change control stiffness to observe the change of beat vibrations:
Gradually increase the control stiffness coefficient from 0.6 times the natural stiffness up to 1.4 times the natural stiffness, the data and the curve of the two sub-vibration spectrum peak changes with stiffness are shown in Table 2 and Figure 5 (b).

frequency	The spectrum peak value at the rotating magnetic field frequency	The spectrum peak value at the rotor rotation frequency
200	153.0	32.0
250	96.3	28.3
300	69.8	28.6
350	42.0	27.0
400	42.0	19.8
450	36.3	17.2

Table.1 The change of the two vibration's frequency spectrum with converter frequency

control stiffness (×natural stiffness)	The spectrum peak value at the rotating magnetic field frequency	The spectrum peak value at the rotor rotation frequency
0.6	126.0	53.3
0.7	101.2	50.0
0.8	92.4	46.3
0.9	61.6	40.2
1.0	42.3	34.9
1.1	40.4	31.8
1.2	28.6	29.0
1.3	16.0	28.1
1.4	12.1	27.0

Table.2 The change of the two vibration's frequency spectrum with stiffness value of the controller



(a) with rotate speed; (b) with stiffness value of the controller

Figure.5 The change of the two vibration's frequency spectrum

It can be seen in Figure 5 (b), under the premise of ensuring the stability of the system, the beat vibration can be significantly inhibited by increasing the stiffness coefficient of the controller.

(3) Multi-channel PID to suppress the vibration signal causing the beat vibration:

Magnetic Bearing is controlled by conventional PID control algorithm[20-21], the rotor speed is set to 20000rpm (inverter frequency of 333Hz).Figure 6 (a) shows the rotor beat vibration curve at the x and y direction at the place of the former radial bearing, where the beat vibration phenomenon is very obvious. Figure 6 (c) shows the frequency spectrum of the x direction vibration near the set frequency of the inverter, there are two distinct peaks, one is at the set frequency 333Hz, the amplitude is nearly 35, the frequency is equal to the motor's rotating magnetic field frequency; another is the frequency of rotation of the rotor, approximately 332.5Hz, the vibration amplitude is about 22.

Then using multi-channel PID control algorithm in magnetic bearings, the rotor speed is the same as 20000rpm[22]. Figure 6 (b) shows rotor beat vibration curve at the x and y directions at the place of the former radial bearing, the beat vibration phenomenon has become insignificant. Figure 6 (d) shows the frequency spectrum of the x direction vibration near the set frequency of the inverter, which still has two peaks, one is the set frequency of the inverter 333Hz, the amplitude is reduced to 6; another is the frequency of rotation of the rotor, about 332.6Hz, the vibration spectrum amplitude is reduced to about 4.Visible, the beat vibration can be effectively suppressed by multi-channel PID control.

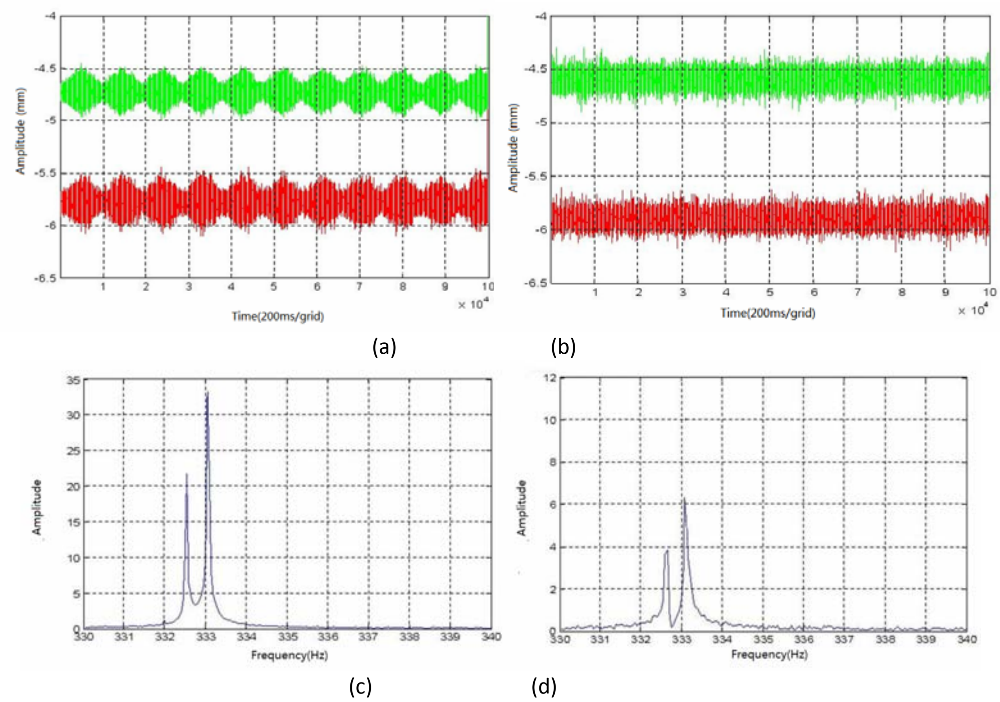


Figure.6 Vibration waveform and frequency spectrum of rotor with conventional and multi-channel PID algorithm

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