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Design of Structural Parameters for Centrifugal Elevator Overspeed Governors

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Abstract: As an important part of overspeed and fail-safe protection for elevators, the centrifugal elevator overspeed governor is a device for limiting overspeed of elevator cars. This paper researches on the vibration of the centrifugal block, which plays a key role in the performance of this overspeed governor. By performing dynamics analysis on the centrifugal block, the differential equation on the vibration of the centrifugal block is established. Based on this, the paper performs simulation analysis on the influence of systematic parameters such as the speed of the overspeed governor sheave, the mass of centrifugal block, the turning radius of the centrifugal block, the position where the spring acts, and the stiffness of the centrifugal block spring, on the vibration of the centrifugal block, and finds out their specific influence relationship. *Copyright © 2014 IFSA Publishing, S. L.*

Keywords: Elevator overspeed governor, Centrifugal block, Dynamics analysis, Vibration differential equation, System parameter.

1. Introduction

Elevators are an indispensable transport for high-rise buildings, and carry thousands of passengers up and down every day. Therefore, safety is of primary concern [1-3].

All modern elevators are provided with a perfect safety protection system, including a series of electrical safety devices and mechanical safety devices. In the safety protection system of an elevator, it is the overspeed governor, safety gear and buffer that provide the final comprehensive safety and security control. That is to say, if for any reason the car is in a dangerous situation of overspeed or even falling during the operation of an elevator and all other safety devices are not functioning, the overspeed governor, safety gear and buffer will work to stop the car, so as to avoid any damage to passengers or the elevator [4-6].

The centrifugal overspeed governor is a very important safety device in modern elevators [7-9], while the centrifugal block plays a critical role in the performance of the overspeed governor. Research on this is rarely found in literature. Therefore, this paper plans to make simulation analysis on the influence of each system parameter on the vibration of the centrifugal block based on dynamics analysis on the centrifugal block and establishing the vibration differential equation of the centrifugal block.

2. System Structure and Mechanical Analysis

The structure of centrifugal overspeed governor is shown in Fig. 1 and Fig. 2. The rope sheave and brake disc of the overspeed governor can rotate on the ratchet shaft of the overspeed governor independently. On the rope sheave of the overspeed governor fix two centrifugal blocks which can rotate around their respective pins.

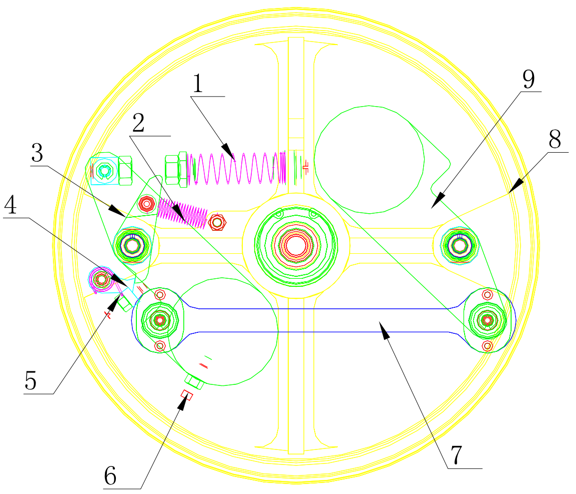

Fig. 1. Inside structure of the overspeed governor.
1 - Centrifugal block spring, 2 - Pawl springs, 3 – Pawl, 4 - Press claw, 5 - Torsional spring, 6 - Adjusting part, 7 - Connecting rod, 8 - Rope sheave, 9 - Centrifugal block.

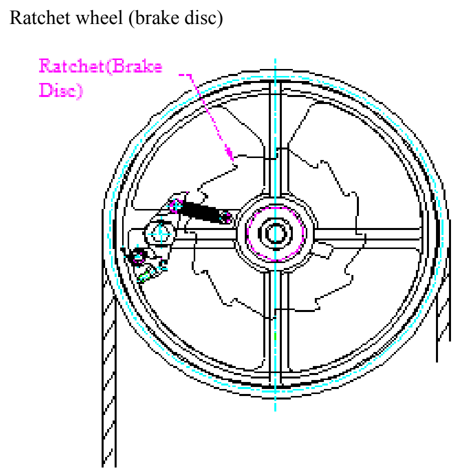

Fig. 2. Schematic Diagram of the position of the ratchet.

They are symmetrically distributed and connected to each other via connecting rod. The spring of the centrifugal block tensions the centrifugal block towards the center, and the size of the spring force can be adjusted with nut. The outer edge of the centrifugal block is provided with a press claw and pawl mechanism. Angled teeth are evenly distributed in the interior circular surface of the brake disc (ratchet). When the overspeed governor sheave is stationary and the centrifugal block is kept tensioned towards the center, a certain gap will be maintained between the centrifugal block and the sheave. When the car runs, the overspeed governor sheave is driven by the connecting rod and overspeed governor rope to rotate, and the centrifugal force acted on the centrifugal block enables the centrifugal block to swing outwards around the pin and maintain balanced with the spring force. The circumferential gap of the centrifugal block is reduced. The faster the speed of the overspeed governor sheave is, the bigger the centrifugal force acted on the centrifugal block and the smaller the circumferential gap. When the car reaches a certain overspeed, the centrifugal block swings outwards, resulting in the rotation of the press claw. The pawl is released, and the pawl is engaged with the angled teeth on the brake disc, and the car is forced to stop. By adjusting the pre-compression amount of the spring of the centrifugal block, the action speed of the overspeed governor can be adjusted.

3. Dynamics Analysis on the Centrifugal Block of the Overspeed Governor

According to the working principles of the mechanism, when the overspeed governor is moving, the centrifugal force is changing due to the change of the speed and the centrifugal radius. In addition, due to the presence of elastic elements, there is inevitably shock and vibration. Therefore, it is key to the design that determining reasonable system parameters to ensure movement stability of the system.

Take the centrifugal block as the research object, and the moving overspeed governor sheave as the reference system. Due to the action of centrifugal force, the centrifugal block will turn an angle of θ around the radius direction of the rotation center of the rope sheave and along its fixed axle. It has both kinetic energy and elastic potential energy of the spring (taking the friction of the revolute pair into account).

Based on the conservation of energy, and the generalized Lagrange equations [10], the following can be obtained:

$$\frac{d}{dt}\left(\frac{\partial T}{\partial \theta}\right)=\frac{\partial}{\partial \theta}(T-V), \quad (1)$$

where T is the kinetic energy of the centrifugal block, J; V is the Elastic potential energy of the spring; θ - angle the centrifugal block turns, rad, among which:

$$T=\frac{1}{2}J_1\dot{\theta}+\frac{1}{2}J_2\dot{\theta}, \quad (2)$$

$$V=\frac{1}{2}K_1\theta^2 R_X^2, \quad (3)$$

where J_1, J_2 are the rotational inertias of two centrifugal blocks, $kg \cdot m^2$; K_1 is the stiffness of the centrifugal block spring, N/m; R_x is the distance from the rotation center of the centrifugal block to the point where the spring acts on the centrifugal block, m.

The generalized force exerted on the system is as follows:

$$\frac{d}{dt}\left(\frac{\partial T}{\partial \dot{\theta}}\right) = F_0(t) - C_v \dot{\theta} R_x^2, \tag{4}$$

where C_v is the damping coefficient; $F_0(t)$ is the Force exerted by the press claw on the centrifugal block, N. Due to the existence of unbalanced force such as self-gravity of the press claw-pawl mechanism and the mass of the connecting rod of the centrifugal block mechanism in the circular motion in the vertical plane, sinusoidal variation will occur to F_0. Accordingly, assume this force is a sinusoidal quantity $F_0(t) = F_0 \sin(pt)$.

Substitute Formula (2), (3) and (4) into Formula (1), the following is obtained:

$$(J_1 + J_2)\frac{d^2\theta}{dt^2} + R_x^2 C_v \frac{d\theta}{dt} - R_x^2 K_1 \theta = F_0 \sin(pt) = F_0 e^{jpt} \tag{5}$$

Assume $A = J_1 + J_2 = 2J$, $B = R_x^2 C_v$, $C = R_x^2 K_1$ then

$$A\ddot{\theta} + B\dot{\theta} - C\theta = F_0 e^{jpt} \tag{6}$$

This equation is a linear non-homogeneous differential equation. It is a forced vibration under the simple harmonic vibration in the single Degree-of-freedom system. Use the complex amplitude of vibration to calculate its steady-state response.

Set the particular solution of the equation as, and substitute it into Formula (6) to get:

$$(Ap^2 + Bp - C)\theta_0 = F_0 \tag{7}$$

$$\theta_0 = \frac{F_0}{Ap^2 + Bp - C} \tag{8}$$

Then the natural frequency of the vibration system is obtained:

$$\omega^2 = \frac{C}{A} = \frac{R_x^2 K_1}{J_1 + J_2}, \tag{9}$$

where $J_1 = m_1 R_1^2$, $J_2 = m_2 R_2^2$, m_1, m_2 are the mass of two centrifugal blocks, kg; R_1, R_2 are the turning radius of two centrifugal blocks, m.

Damping ratio of the system $\xi = B/2\sqrt{AC} = R_x C_v / 2\sqrt{2JK_1}$; the ratio of exciting frequency to natural frequency $\Omega = p/\omega$; the displacement of the vibration system under action of the centrifugal force $\theta_{st} = \frac{F_0}{(m_1 + m_2)\omega^2}$. Then

$$\theta = \frac{F_0}{c\sqrt{(1-\Omega^2)^2 + 4\xi^2\Omega^2}} e^{-ja}, \tag{10}$$

where $\partial = \tan^{-1}\frac{2\xi\Omega}{1-\Omega^2}$, substitute the constant of the static equilibrium position into Formula (9), then the particular solution of the steady-state vibration equation can be obtained.

$$\theta(t) = \theta e^{jpt} = \frac{\theta_{st}}{\sqrt{(1-\Omega^2)^2 + 4\xi^2\Omega^2}} e^{-j(pt-\alpha)} \tag{11}$$

4. Simulation Analysis on the Vibration of Centrifugal Block

Based on the particular solution of steady-state vibration obtained from the above analysis on the system vibration differential equation, the simulation analysis on the influence of each relevant parameter on the vibration of the centrifugal block can be performed, and the system parameters which are helpful to the control can be identified. Simulation parameters: $R_1=R_2=0.15$ m, $R_x=0.1$ m, $K_1=0.01$ N/mm, $m_1=m_2=1$ kg.

4.1. Influence of the Speed of the Overspeed Governor Sheave

Change the speed of the overspeed governor sheave from 0.5 m/s to 3.5 m/s by an increment of 0.5 m/s. Compare the relationship between the amplitude, velocity and acceleration of the vibration of the centrifugal block and the time, as shown in Fig. 3 to Fig. 5.

As is found from the simulation results, the vibration characteristics are not uncertain when the speed of the overspeed governor sheave is 0.5 m/s. At this point, it is more difficult to achieve accurate control for the system. As the speed increases, the system vibration characteristics tend to be clear, indicating that the speed within a certain range makes it easier to realize control. As is seen from this, low speed is difficult for the control. Therefore in the following, the case of low speed, i.e. where the speed of the overspeed governor sheave is 0.5 m/s, is studied.

4.2. Influence of the Mass of the Centrifugal Block

Change the mass of the centrifugal block from 0.5 kg to 2.5 kg by an increment of 1.0 kg. Compare the relationship between the amplitude, velocity and acceleration of the vibration of the centrifugal block and the time, as shown in Fig. 6 to Fig. 8.

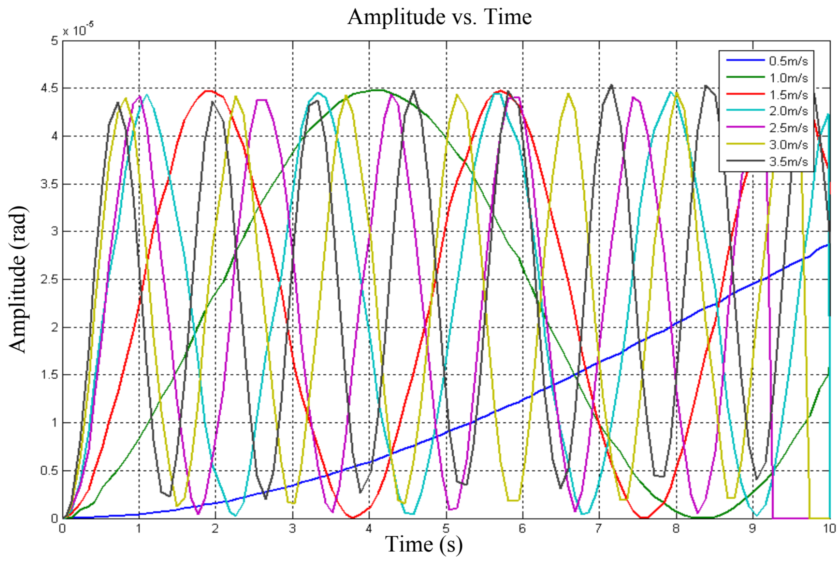

Fig. 3. Influence of the speed of the overspeed governor sheave on the amplitude of the centrifugal block.

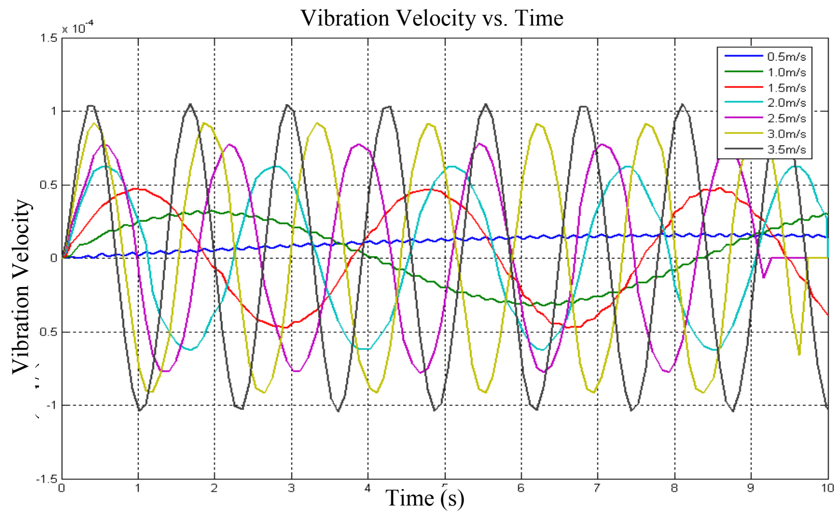

Fig. 4. Influence of the speed of the overspeed governor sheave on the vibration velocity of the centrifugal block.

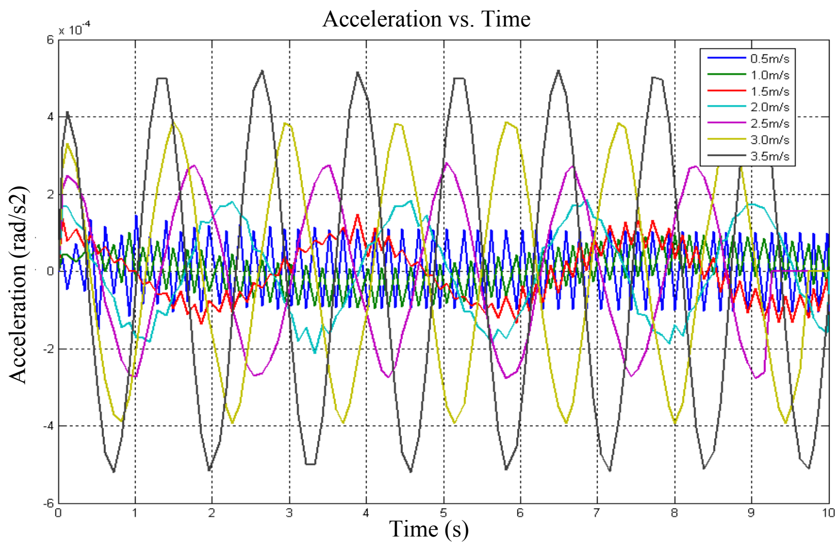

Fig. 5. Influence of the speed of the overspeed governor sheave on the vibration acceleration of the centrifugal block.

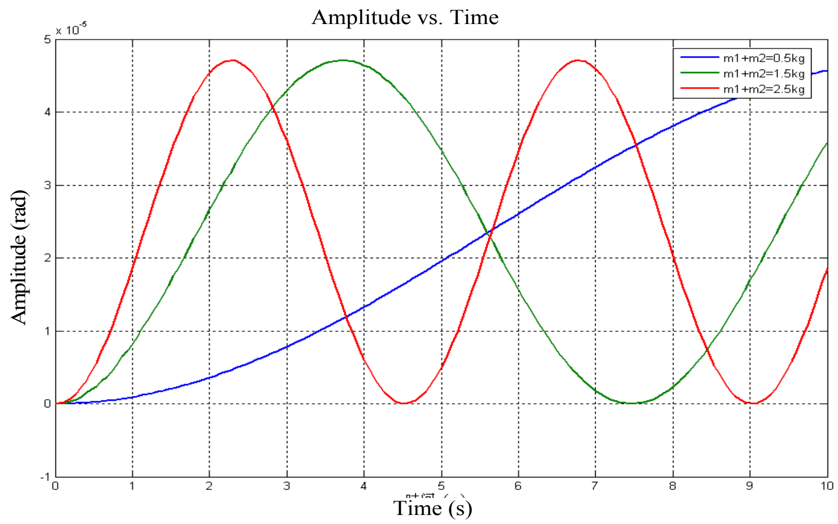

Fig. 6. Influence of the mass of the centrifugal block on the amplitude of the centrifugal block.

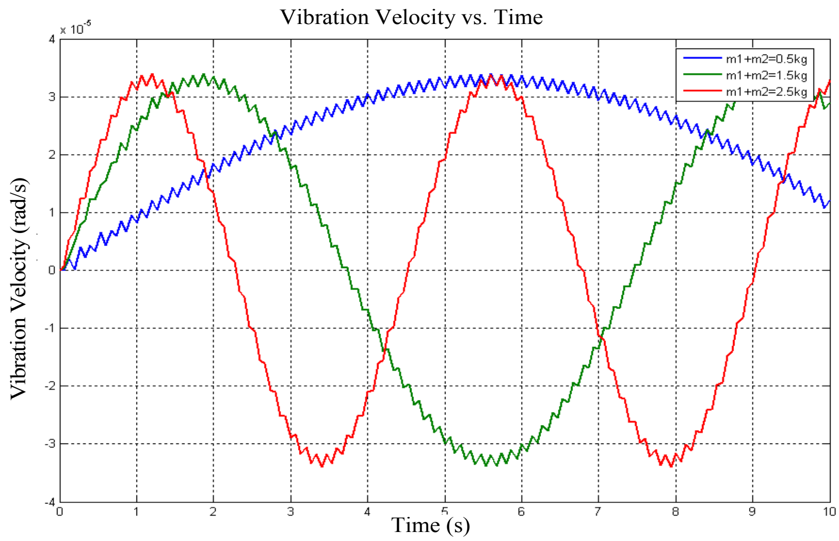

Fig. 7. Influence of the mass of the centrifugal block on the vibration velocity of the centrifugal block.

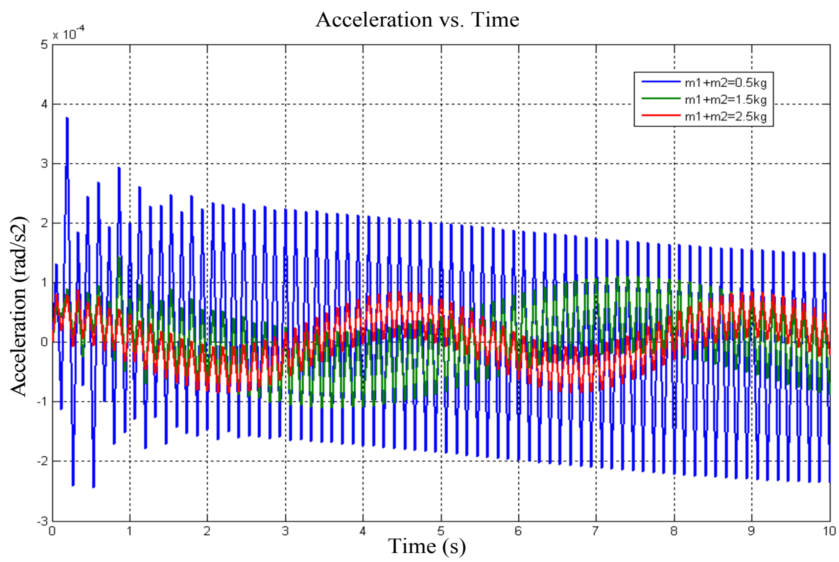

Fig. 8. Influence of the mass of the centrifugal block on the vibration acceleration of the centrifugal block.

As is found from the simulation results, the vibration characteristics tend to be uncertain when the mass of the centrifugal block is too small. Regular movement only occurs when the mass falls in a certain range (greater than 1.5 kg) to achieve stable control. Therefore, the mass should be about 2.5 kg.

4.3. Influence of the Turning Radius of the Centrifugal Block

Change the turning radius of the centrifugal block from 0.02 mm to 0.08 mm by an increment of 0.03 mm. Compare the relationship between the amplitude, velocity and acceleration of the vibration of the centrifugal block and the time, as shown in Fig. 9 to Fig. 11.

As is found from the simulation results, the turning radius of the centrifugal block has significant influence on the vibration of the system. In case of low speed (0.5 m/s) and when the parameter is 0.02 mm, the vibration characteristics of the system are uncertain, and it is difficult to realize accurate control. Therefore, such structural proportion should be avoided. Only when the turning radius of the centrifugal block reaches about 0.08 does system have certain vibration characteristics.

4.4. Influence of the Position Where the Spring Acts

Change the position where the spring acts from 0.02 mm to 0.05 mm by an increment of 0.01 mm. Compare the relationship between the amplitude, velocity and acceleration of the vibration of the centrifugal block and the time, as shown in Fig. 12 to Fig. 14.

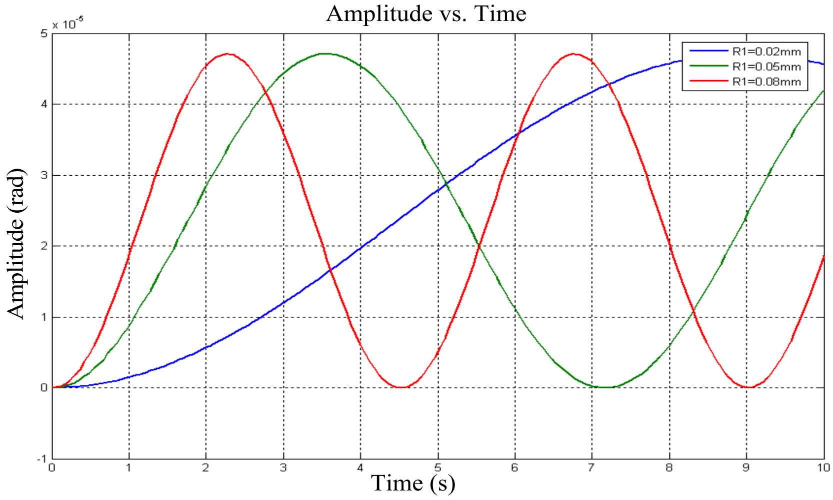

Fig. 9. Influence of the turning radius of the centrifugal block on the amplitude of the centrifugal block.

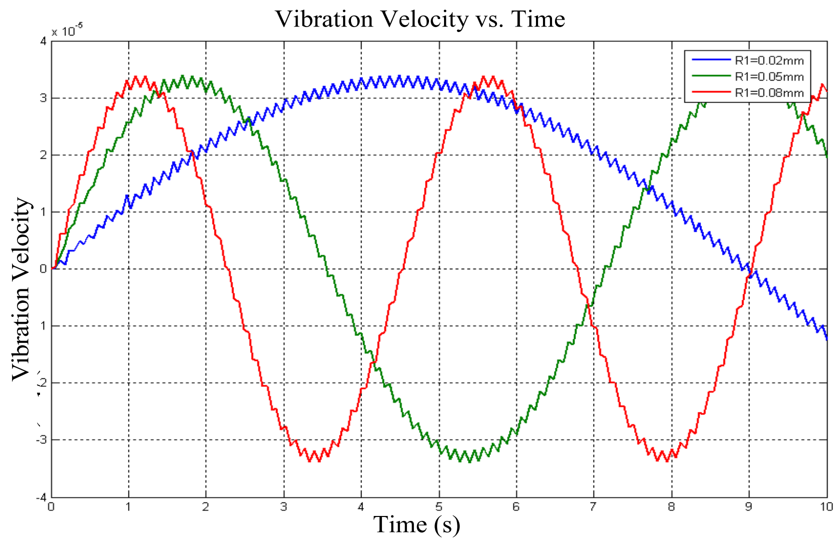

Fig. 10. Influence of the turning radius of the centrifugal block on the vibration velocity of the centrifugal block.

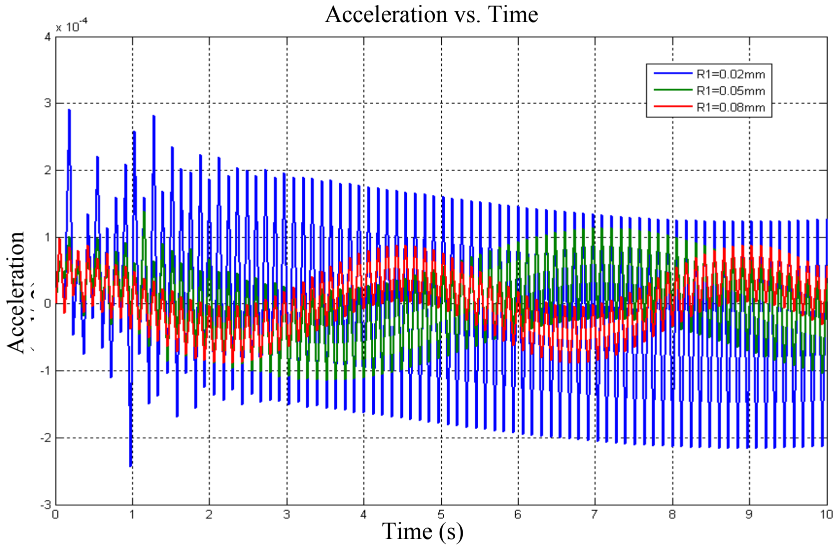

Fig. 11. Influence of the turning radius of the centrifugal block on the vibration acceleration of the centrifugal block.

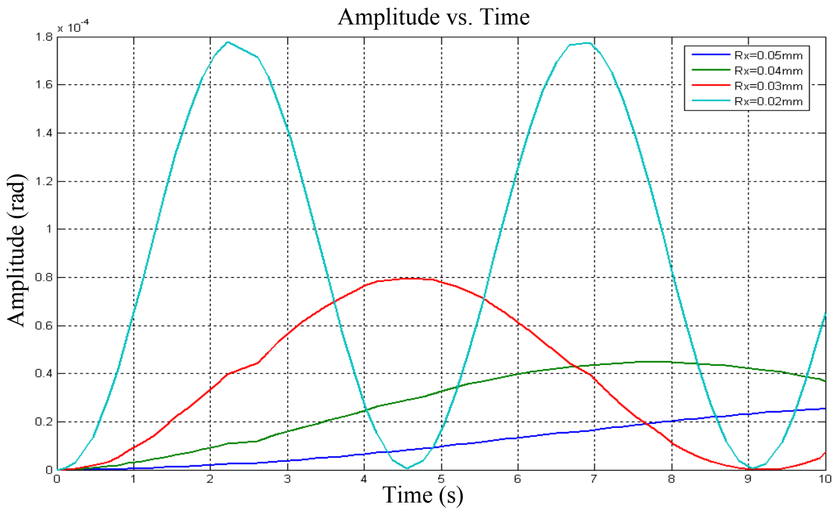

Fig. 12. Influence of the position where the spring acts on the amplitude of the centrifugal block.

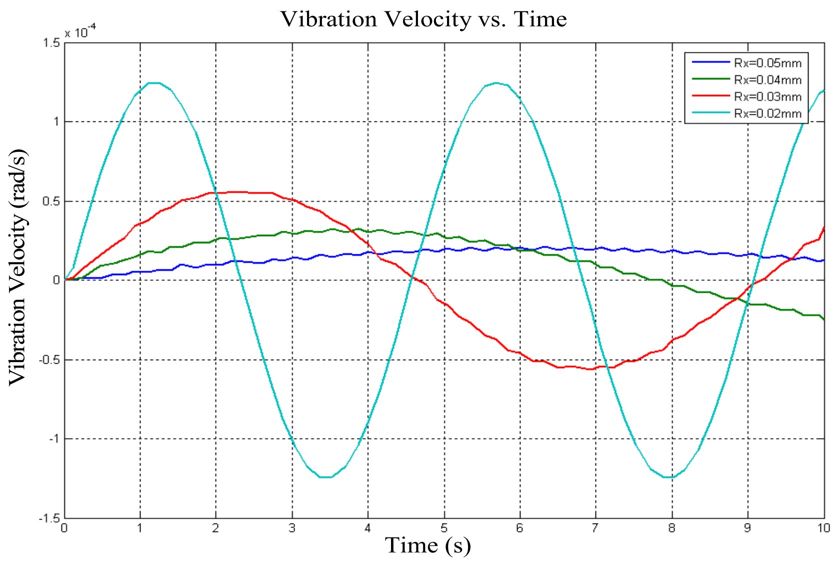

Fig. 13. Influence of the position where the spring acts on the vibration velocity of the centrifugal block.

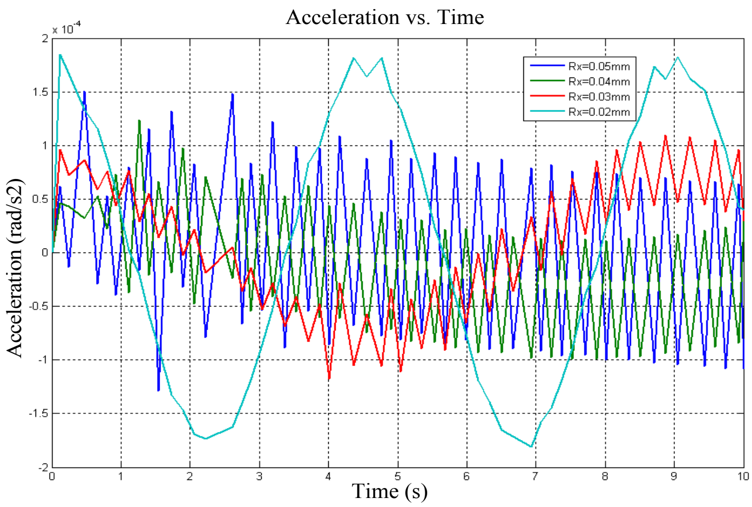

Fig. 14. Influence of the position where the spring acts on the vibration acceleration of the centrifugal block.

As is found from the simulation results, the position where the spring acts has significant influence on the vibration of the system. In design, such structural parameter should be controlled below 0.03 m before a reliable control can be obtained.

4.5. Influence of the Stiffness of the Centrifugal Block Spring

Change the stiffness of the centrifugal block spring from 0.0345 N/mm to 0.4354 N/mm by an increment of 0.1 N/mm. Compare the relationship between the amplitude, velocity and acceleration of the vibration of the centrifugal block and the time, as shown in Fig. 15.

As is found from the simulation results, the change of the stiffness of the centrifugal block spring directly influences the stability of the vibration of the system. When the spring stiffness is large, the vibration of the system is diverging. In case of low-speed operation, the stiffness of the centrifugal block spring should be controlled below 0.035 N/mm when it is designed based on the parameters given in this paper.

5. Conclusions

When the overspeed governor is moving, the centrifugal force acted on the centrifugal block is changing due to the change of the rotary speed of the overspeed governor sheave and the centrifugal radius of the centrifugal block. In addition, due to the presence of elastic elements, the centrifugal block is caused to vibrate. This paper makes simulation analysis on the influence of system parameters such as the speed of the overspeed governor sheave, the mass of the centrifugal block, the turning radius of the centrifugal block, the position where the spring acts, and the stiffness of the centrifugal block spring, on the vibration of the centrifugal block based on establishing the vibration differential equation of the centrifugal block.

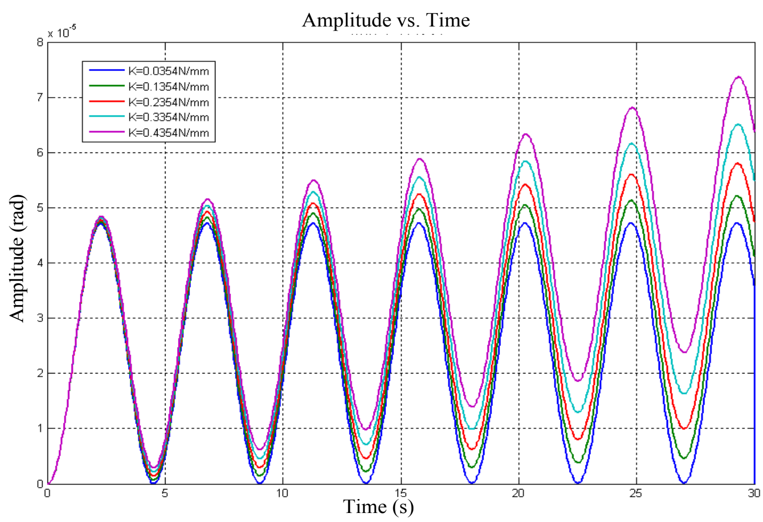

Fig. 15. Influence of the stiffness of the centrifugal block spring on the amplitude of the centrifugal block.

As is found from the simulation results, the low-speed operation of the overspeed governor sheave is the difficulty for the control. Since the speed is low, the vibration characteristics of the system tend to be uncertain. The stiffness of the centrifugal block spring is a key parameter affecting the vibration stability of the system. When the spring stiffness is large, the vibration of the system is diverging. The mass of the centrifugal block, the turning radius of the centrifugal block and the position where the spring acts all influence the vibration of the system, and they are parameters on which control should be focused in design. When the mass of the centrifugal block and the turning radius of the centrifugal block are too small and the position where the spring acts is too distant, the vibration characteristics of the system tend to be uncertain, resulting in that it is difficult to realize accurate control. Therefore such structural proportion should be avoided whenever possible in design.

References

[1]. Chen Jiasheng, Structure Principle, Installation and Maintenance of Elevators, *China Machine Press,* 2011, pp. 200-225.

[2]. Zhu Dewen, Zhang Zhendi, Li Dawei, Detailed Graphic Demonstration of Elevator Installation, *China Electric Power Press*, 2011, pp. 1.

[3]. Wei Kongping, Zhu Rong, Elevator Technology, *Chemical Industry Press*, 2006, pp. 7.

[4]. Zhu Changming, Safety Rules for the Construction and Installation of Electric Lifts, *Standards Press of China*, 2007, pp. 11.

[5]. Bo Yonggang, Method and Precaution for Site Verification of Overspeed Governors, *China Elevator*, 13, 2009, pp. 51-52.

[6]. He Deming, Xiao Weiping, Elevator Structure and Principle, *Sun Yat-Sen Universiey Press*, 2009, pp. 8.

[7]. Yong Huifeng, Jian Wuzhang, Yane Zhao, Modeling and Robust Control of Horizontal Vibrations for High-speed Elevator, *Journal of Vibration and Control*, 15, 9, 2009, pp. 1375–1396.

[8]. Jiang Jing, Zhang Xuesong, Variable Frequency Speed-regulation System of Elevator Using PLC Technology, in *Proceedings of the 3rd International Conference on Advanced Computer Control (ICACC'11),* 2011.

[9]. Jing Xin Deng, Xiao Ding Guo, Xiao Ke Chen, Zheng Mei Cheng, Simulation and Application on the Adaptive Fuzzy PI Control in Low-Rise Elevator Speed Regulation System, *Applied Mechanics and Materials*, 103, 2011, pp. 550-555.

[10]. Wu Zhen, Theoretical Mechanics (II), *Shanghai Jiaotong University Press*, 2011, pp. 334-342.