Active Vibration Control of an Elevator Car Using Two Rotary Actuators*

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Abstract

A new control device is proposed to isolate an elevator cabin from irregularities on a guide rail. The device consists of two rotary electric motors with disks and eccentric masses. One motor produces torque to rotate the eccentric mass according to the command signals from the controller for horizontal control, while the other rotates the mass to eliminate the unwanted vertical force. To design the optimal controller to reduce the horizontal vibration of the cabin, the dynamics of the cabin, including the proposed control device, are described by a linear equation. The performance of the proposed control system is examined through numerical simulations taking into consideration the nonlinear characteristics of the rotating masses, which are neglected in designing the controller, and through experiments with a miniature elevator cabin carrying the proposed actuator. The results indicate the proposed control device reduces the horizontal vibration without producing the unwanted vertical vibration.

Key words: Elevator, Vibration Control, Optimal Control, Vibration Control Device, Dynamic Absorber

1. Introduction

In recent years the travelling speeds of elevator cabins have increased significantly, and passengers have become much more aware of the ride quality they experience. Consequently the need to suppress vibrations of the cabin has become an important issue in elevator system design. Some research groups have focused on the analysis of the vibrations caused by the movement of the travelling rope⁽¹⁾, and discussed methods for suppressing these vibrations^{(2),(3)}. Active vibration control is a major method for suppressing the vibration. To isolate the cabin from horizontal vibrations caused by the roughness of the rails which guide the elevator cabin, a control system using a servo-motor^{(4),(5)}, an active mass damper system^{(6),(7)} and active roller guides⁽⁸⁾ were trialed. The performances of these systems were examined and some of them were already being used in practice. However, to make them more acceptable, there is a need to reduce their size so that the engineers do not have to change the design of the typical elevator cabin. This paper proposes a new device for active vibration control to isolate the horizontal vibrations resulting from irregularities of the rails.

*Received 27 July, 2010 (No. T1-08-0831)
Japanese Original : Trans. Jpn. Soc. Mech.
Eng., Vol.75, No.753, C (2009),
pp.1377-1382 (Received 19 Sep., 2008)
[DOI: 10.1299/jsdd.5.155]

The proposed control device consists of two rotary electric motors with eccentric masses. The two masses rotate in opposite directions at the same speed to produce horizontal forces without generating any unwanted vertical forces. Compared to a design using a linear actuator, we expected to make the device more compact⁽⁹⁾.

The equations of the motion are formulated assuming the main rope which suspends the elevator cabin is elastic, and the cabin is allowed to move in roll, horizontal and vertical directions. One of the motors produces control torque according to the command signal from the controller, while the other motor rotates to counteract any vertical forces. The controller is designed using an optimal control scheme to suppress both roll and horizontal vibration. The performance of the proposed system is examined both through numerical simulations taking into consideration the nonlinear characteristics of the rotating eccentric masses and through laboratory experiments using a miniature elevator cabin carrying the proposed control device.

2. Proposed Control Device

The proposed control device is shown in Fig.1. It is composed of two electric rotary actuators with identical eccentric masses m_d . Each mass is attached to a uniform rigid disk whose inertia moment around its center is I_d . By rotating the eccentric mass, the rotary actuator produces control force in the lateral direction to reduce vibrations of the cabin; however, it also produces unwanted vertical forces.

Fig. 1 The proposed vibration control device for the elevator cabin.

To eliminate the unwanted vertical forces the other actuator produces the counter vertical force. It is successful when the rotating angles ϕ_{d1} , ϕ_{d2} of the two masses keep the following relationship:

$$\phi_{d1} = \pi - \phi_{d2} \quad (1)$$

We use the new coordinate for the actuator angles θ_{d1} , θ_{d2} . While the coordinate for the first actuator is the same as mentioned above, the initial angle of the second actuator is set to be π and the positive direction is reversed, that is:

$$\theta_{d1} = \phi_{d1} \quad (2)$$

$$\theta_{d2} = \pi - \phi_{d2} \quad (3)$$

In this new coordinate system, the condition that cancels the vertical forces produced by the two actuators is given by:

$$\theta_{d1} = \theta_{d2} \quad (4)$$

3. Equation of Motion

Figure 2 shows the dynamic model of the elevator cabin with the proposed control device. The roughness of the guide disturbs the cabin. The displacements of the upper and

lower springs are expressed as x_{0u} and x_{0d} . When the lateral and vertical displacements and the angle of the center of the gravity of the cabin are defined as x_c , z_c , and θ_c , the positions of the eccentric masses (x_{d1} , z_{d1}) and (x_{d2} , z_{d2}) can be expressed as:

$$(x_{d1}, z_{d1}) = (x_c - l_{xd1} + l_d \theta_c - r_d \sin \theta_{d1}, \quad z_c - l_d - r_d \cos \theta_{d1}) \quad (5)$$

$$(x_{d2}, z_{d2}) = (x_c + l_{xd2} + l_d \theta_c - r_d \sin \theta_{d2}, \quad z_c - l_d + r_d \cos \theta_{d2}) \quad (6)$$

Then the total kinetic energy T and the potential energy P of the system are written as:

$$T = \frac{1}{2} \{ m_c (\dot{x}_c^2 + \dot{z}_c^2) + I_c \dot{\theta}_c^2 + I_d \dot{\theta}_{d1}^2 + I_d \dot{\theta}_{d2}^2 + m_d (\dot{x}_{d1}^2 + \dot{z}_{d1}^2) + m_d (\dot{x}_{d2}^2 + \dot{z}_{d2}^2) \} \quad (7)$$

$$P = \frac{1}{2} \{ k_u (x_c - l_1 \theta_c - x_{0u})^2 + k_d (x_c + l_2 \theta_c - x_{0d})^2 + k_r z_c^2 \} - m_d g r_d \cos \theta_{d1} + m_d g r_d \cos \theta_{d2} \quad (8)$$

The elevator cabin does not have dampers, but in practice it dissipates energy inside it. Assuming the forces dissipating energy are viscous damping forces, we introduce Rayleigh's dissipation function R , written as:

$$R = \frac{1}{2} \{ c_p \dot{x}_c^2 + c_c \dot{\theta}_c^2 + c_r \dot{z}_c^2 + c_{d1} \dot{\theta}_{d1}^2 + c_{d2} \dot{\theta}_{d2}^2 \} \quad (9)$$

From Lagrange's equation, the equation of motion of the system is obtained as shown in Eq. 10. In this equation, τ_1 and τ_2 are torques produced by the Actuators 1 and 2. The equation of motion represented in Eq. 10 includes nonlinear elements; however, it is easier to develop a linear equation in designing a controller. Assuming small displacements and angles, Eq. 10 reduces to a linear equation as shown in Eq. 11. From this equation, we can see the vertical motion has no relationship to any other motion, and therefore we can omit the second row.

Fig. 2 The elevator cabin with the proposed control device.

$$\begin{aligned}
 & \begin{bmatrix} m_c + 2m_d & 0 & 2m_d l_d & -m_d r_a \cos \theta_{d1} & -m_d r_a \cos \theta_{d2} \\ 0 & m_c + 2m_d & 0 & m_d r_a \sin \theta_{d1} & -m_d r_a \sin \theta_{d2} \\ 2m_d l_d & 0 & I_c + 2m_d l_d^2 & -m_d l_d r_a \cos \theta_{d1} & -m_d l_d r_a \cos \theta_{d2} \\ -m_d r_a \cos \theta_{d1} & m_d r_a \sin \theta_{d1} & -m_d l_d r_a \cos \theta_{d1} & m_d r_a^2 + I_d & 0 \\ -m_d r_a \cos \theta_{d2} & -m_d r_a \sin \theta_{d2} & -m_d l_d r_a \cos \theta_{d2} & 0 & m_d r_a^2 + I_d \end{bmatrix} \begin{bmatrix} \ddot{x}_c \\ \ddot{z}_c \\ \ddot{\theta}_c \\ \ddot{\theta}_{d1} \\ \ddot{\theta}_{d2} \end{bmatrix} \\
 & + \begin{bmatrix} k_u + k_d & 0 & -k_u l_1 + k_d l_2 & 0 & 0 \\ 0 & k_r & 0 & 0 & 0 \\ -k_u l_1 + k_d l_2 & 0 & l_1^2 k_u + l_2^2 k_d & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 \end{bmatrix} \begin{bmatrix} x_c \\ z_c \\ \theta_c \\ \theta_{d1} \\ \theta_{d2} \end{bmatrix} + \begin{bmatrix} c_x & 0 & 0 & 0 & 0 \\ 0 & c_r & 0 & 0 & 0 \\ 0 & 0 & c_\theta & 0 & 0 \\ 0 & 0 & 0 & c_d & 0 \\ 0 & 0 & 0 & 0 & c_d \end{bmatrix} \begin{bmatrix} \dot{x}_c \\ \dot{z}_c \\ \dot{\theta}_c \\ \dot{\theta}_{d1} \\ \dot{\theta}_{d2} \end{bmatrix} \\
 & + \begin{bmatrix} m_d r_a \dot{\theta}_{d1}^2 \sin \theta_{d1} + m_d r_a \dot{\theta}_{d2}^2 \sin \theta_{d2} \\ m_d r_a \dot{\theta}_{d1}^2 \cos \theta_{d1} - m_d r_a \dot{\theta}_{d2}^2 \cos \theta_{d2} \\ m_d l_d r_a \dot{\theta}_{d1}^2 \sin \theta_{d1} + m_d l_d r_a \dot{\theta}_{d2}^2 \sin \theta_{d2} \\ m_d g r_a \sin \theta_{d1} \\ -m_d g r_a \sin \theta_{d2} \end{bmatrix} = \begin{bmatrix} k_u & k_d \\ 0 & 0 \\ -l_1 k_u & l_2 k_d \\ 0 & 0 \\ 0 & 0 \end{bmatrix} \begin{bmatrix} x_{0u} \\ x_{0d} \end{bmatrix} + \begin{bmatrix} 0 & 0 \\ 0 & 0 \\ 1 & 0 \\ 0 & 1 \end{bmatrix} \begin{bmatrix} \tau_1 \\ \tau_2 \end{bmatrix} \quad (10)
 \end{aligned}$$

$$\begin{aligned}
 & \begin{bmatrix} m_c + 2m_d & 0 & 2m_d l_d & -m_d r_a & -m_d r_a \\ 0 & m_c + 2m_d & 0 & 0 & 0 \\ 2m_d l_d & 0 & I_c + 2m_d l_d^2 & -m_d l_d r_a & -m_d l_d r_a \\ -m_d r_a & 0 & -m_d l_d r_a & m_d r_a^2 + I_d & 0 \\ -m_d r_a & 0 & -m_d l_d r_a & 0 & m_d r_a^2 + I_d \end{bmatrix} \begin{bmatrix} \ddot{x}_c \\ \ddot{z}_c \\ \ddot{\theta}_c \\ \ddot{\theta}_{d1} \\ \ddot{\theta}_{d2} \end{bmatrix} + \begin{bmatrix} c_x & 0 & 0 & 0 & 0 \\ 0 & c_r & 0 & 0 & 0 \\ 0 & 0 & c_\theta & 0 & 0 \\ 0 & 0 & 0 & c_d & 0 \\ 0 & 0 & 0 & 0 & c_d \end{bmatrix} \begin{bmatrix} \dot{x}_c \\ \dot{z}_c \\ \dot{\theta}_c \\ \dot{\theta}_{d1} \\ \dot{\theta}_{d2} \end{bmatrix} \\
 & + \begin{bmatrix} k_u + k_d & 0 & -k_u l_1 + k_d l_2 & 0 & 0 \\ 0 & k_r & 0 & 0 & 0 \\ -k_u l_1 + k_d l_2 & 0 & l_1^2 k_u + l_2^2 k_d & 0 & 0 \\ 0 & 0 & 0 & m_d g r_a & 0 \\ 0 & 0 & 0 & 0 & -m_d g r_a \end{bmatrix} \begin{bmatrix} x_c \\ z_c \\ \theta_c \\ \theta_{d1} \\ \theta_{d2} \end{bmatrix} = \begin{bmatrix} k_u & k_d \\ 0 & 0 \\ -l_1 k_u & l_2 k_d \\ 0 & 0 \\ 0 & 0 \end{bmatrix} \begin{bmatrix} x_{0u} \\ x_{0d} \end{bmatrix} + \begin{bmatrix} 0 & 0 \\ 0 & 0 \\ 1 & 0 \\ 0 & 1 \end{bmatrix} \begin{bmatrix} \tau_1 \\ \tau_2 \end{bmatrix} \quad (11)
 \end{aligned}$$

To maintain $\theta_{d1} = \theta_{d2}$, the equations of eccentric masses (the equations of the fourth and fifth rows) need be identical. Then the torque τ_2 should be:

$$\tau_2 = \tau_1 - m_d g r_a \theta_{d1} - m_d g r_a \theta_{d2} \quad (12)$$

When this relationship is maintained, the fifth row can be omitted. It is not easy to produce such a torque, since a practical system includes modelling error. Therefore it is better to design a regulator such as a PD controller to control θ_{d2} to follow θ_{d1} . However the model used in the numerical simulation assumes an ideal system, for which there are no modelling errors, and there is complete control of θ_{d2} . This condition is identical for the torque produced by Actuator 2 as shown in Eq. 12. Then the equation of motion reduces to the following simple linear equation:

$$\begin{aligned}
 & \begin{bmatrix} m_c + 2m_d & 2m_d l_d & -2m_d r_a \\ 2m_d z_2 & I_c + 2m_d l_d^2 & -2m_d l_d r_a \\ -2m_d r_a & -2m_d z_2 r_a & 2m_d r_a^2 + 2I_d \end{bmatrix} \begin{bmatrix} \ddot{x}_c \\ \ddot{\theta}_c \\ \ddot{\theta}_{d1} \end{bmatrix} + \begin{bmatrix} c_x & 0 & 0 \\ 0 & c_r & 0 \\ 0 & 0 & 2c_d \end{bmatrix} \begin{bmatrix} \dot{x}_c \\ \dot{\theta}_c \\ \dot{\theta}_{d1} \end{bmatrix} \\
 & + \begin{bmatrix} k_u + k_d & -k_u l_1 + k_d l_2 & 0 \\ -k_u z_1 + k_d z_2 & z_1^2 k_u + z_2^2 k_d & 0 \\ 0 & 0 & 2m_d g r_a \end{bmatrix} \begin{bmatrix} x_c \\ \theta_c \\ \theta_{d1} \end{bmatrix} = \begin{bmatrix} k_u & k_d \\ -z_1 k_u & z_2 k_d \\ 0 & 0 \end{bmatrix} \begin{bmatrix} x_{0u} \\ x_{0d} \end{bmatrix} + \begin{bmatrix} 0 \\ 0 \\ 2 \end{bmatrix} \tau_1 \quad (13)
 \end{aligned}$$

The values of the parameters utilized in this study are shown in Table 1.

Table 1 Values of the parameters

Description	Symbols	Values
Damping coefficients	c_d	1.2 mNm/(rad/s)
	c_r, c_x	29.0, 41.1 N/(m/s)
	c_θ	3.58 Nm/(rad/s)
Gravity	g	9.8 m/s ²
Spring constants	k_d, k_r, k_u	1424, 13265, 1424 N/m
Masses	m_c, m_d	21, 0.2 kg
Radius	r_d	0.05m
Length	l_1, l_2, l_d	0.415, 0.230, 0.195m
Inertia moment	I_{yy}, I_d	1.38 kgm ² , 0.52 gm ²

4. Design of a Controller

An optimal control scheme is applied to the design of the feedback controller. The main purpose of the active control is to reduce horizontal and roll acceleration, therefore, with trial and error, the cost function is obtained as:

$$J = \int_0^\infty (X^T Q X + u^T R u) dt \quad (14)$$

where

$$Q = \begin{bmatrix} 4 \times 10^6 & 0 & 0 & 0 & 0 & 0 \\ 0 & 2.5 \times 10^5 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0.1 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0.1 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0.1 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0.1 \end{bmatrix} \quad (15)$$

$$R = [1] \quad (16)$$

$$X = [x_c \quad \theta_c \quad \theta_{d1} \quad \dot{x}_c \quad \dot{\theta}_c \quad \dot{\theta}_{d1}]^T \quad (17)$$

The values of matrixes Q and R are selected to reduce the amplitude of vibration to less than a half at the first mode. We define the in-phase disturbance as x_{0in} and the reverse-phase disturbance as x_{0re} which can be expressed by:

$$x_{0in} = \frac{x_{0u} + x_{0d}}{2} \quad (18)$$

$$x_{0re} = \frac{x_{0u} - x_{0d}}{2} \quad (19)$$

Figure 3 presents the frequency responses from in-phase or reverse-phase disturbances to lateral displacement or roll, where ‘with control’ indicates the elevator cabin for which the proposed control device is installed and operated, while ‘before install’ represents the cabin where the device is not installed. First and second mode natural frequencies are 1.7Hz and 2.3Hz. This control device is effective in suppressing the gains especially for the first mode, which is considered to have the most influence on the ride quality. Improving the performance for the second and the higher mode is our future topic.

Numerical simulations using the fourth order Runge-Kutta method are carried out with the cabin exposed to a sinusoidal in-phase disturbance whose amplitude is 0.4mm and frequency is 1.7Hz. In this simulation, the non-linear elements are considered, that is, the equation shown in Eq. 10 is used. Figure 4 shows the results. The amplitudes of the horizontal and rotating vibrations of the controlled system are reduced by approximately

one third and one half compared with those of the system without control. However, when the system is controlled, a vertical acceleration is produced, which is not found when the system is uncontrolled, but its amplitude is less than 10^{-7} m/s^2 , so that its effect is negligible.

Fig. 3 Frequency responses, which are the gains (a) from the in-phase disturbance to the lateral displacement, (b) from the in-phase disturbance to the roll, (c) from the reverse-phase disturbance to the lateral displacement, and (d) from the reverse-phase disturbance to the roll.

Fig. 4 The results of the numerical simulation under the same condition as the experiments, where (a) represents the lateral acceleration at the center of gravity, (b) represents the roll acceleration, (c) represents the vertical acceleration of the cabin and (d) represents the rotating angle of the disk of Actuator 1.

5. Experimental setup

Figures 5 and 6 show a photograph of the experimental setup and its schematic view. The miniature cabin with the proposed control device is inside the frame, which is shaken by the vibration exciter in the horizontal direction, and is allowed to move in the vertical and the horizontal directions and rotate around the center of gravity. There are four springs, which connect the upper and lower parts of the cabin horizontally to the frame, and an axis for rotation at the center of gravity, which is supported by four springs in the vertical direction.

Fig. 5 A photograph of the experimental setup.

Fig. 6 A schematic view of the experimental setup.

Two laser displacement sensors and two accelerometers measure the displacements and accelerations at the upper and the lower parts of the cabin. The horizontal and rotating angles of the cabin are estimated from the values measured by these sensors. Another displacement sensor and accelerometer are installed to measure the displacement, and the acceleration of the table of the exciter. The control device has two disks with weights, which are rotated by electric motors. Its rotating angles are measured by encoders installed on the motor axes. Measured data are transmitted to a digital signal processing unit and command signals are sent to two electric motors to control the vibration of the cabin. In the

experiments, the rotating angle of the second actuator was controlled with PID controller to keep the relationship shown in Eq. 4. We then applied the controller, which was designed using the reduced order model written in Eq. 13, to the experimental system.

6. Results

Figure 7 shows the results of the experiment when using the in-phase sinusoidal displacement whose amplitude is 0.4mm and frequency is 1.7Hz. In this figure ‘without control’ indicates the system where the proposed control device is installed but not operated; and the rotating disks are fixed. The lateral and the roll accelerations are reduced to less than the half of those without control. Theoretically vertical vibration is not caused by the lateral vibration in the system ‘without control’; therefore the measured acceleration is due to factors not considered in the equation of motion. The amplitudes of the vertical accelerations seem to be the same between the experiment and the numerical simulation. This indicates the proposed device does not cause the significant unwanted vertical vibration. These results show the proposed method is effective for suppression of vibration of the elevator cabin under these experimental conditions.

The absolute values of the experimental results are approximately half those of the numerical simulations. This is due to errors in the identification of the coefficient of internal damping. The internal damping forces mainly originate from the friction forces, therefore it is difficult to identify it as viscous damping. Since the ratio of the vibration amplitude of the controlled to the uncontrolled systems is similar in the experiment and the numerical simulation, it is concluded that the isolation performance can be examined using this model.

Fig. 7 Results of the experiment when the amplitude and the frequency of the in-phase sinusoidal displacement are 0.4mm and 1.7Hz; (a) represents the lateral acceleration at the center of gravity, (b) represents the roll acceleration, (c) represents the vertical acceleration of the cabin and (d) represents the rotating angle of the disk of the Actuator 1.

6. Conclusions

The authors proposed a rotary actuator system to suppress the vibration of the cabin of the elevator. The controller is designed using optimal control theory and its performance is examined through experiments and numerical simulations. In summary, the following observations are made:

1. The proposed control device reduces the vibration, especially in the first mode.
2. The proposed control device provides good isolation from the in-phase vibration in the first mode.
3. The proposed model for the numerical simulation is effective in examining the isolation performance through experiments and numerical simulation.
4. The proposed control device does not increase the vertical vibration.

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