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VIBRATION CONTROL SYSTEM WITH DIGITALLY ADJUSTABLE ELECTROMAGNETIC DAMPING AND STIFFNESS

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ABSTRACT

In this paper, we present an active vibration control system for wind tunnel testing of high-rise buildings. This system was designed with digitally adjustable electromagnetic damping and stiffness, consisting of a sensing module, control centre, power amplifier, and shaking table. The control system was built on the basis of the principle that electrified coils, in a stable magnetic field, generate an electromagnetic force proportional to the current carried. Aeroelastic models of high-rise buildings can be fixed on the shaking table for wind tunnel testing under conditions of different wind speeds and wind directions. The damping and stiffness can be adjusted digitally and a real-time data acquisition and control algorithms were applied to ensure the reliability of the real-time control. This system is well suited for the control of damping and stiffness for vibration frequencies lower than 50 Hz. The control results agreed well with theoretical predictions. The system reported here is beneficial to the study of dynamic behaviour of structures under wind loads.

KEYWORDS

Actuator, electromagnetic force, vibration control, damping, stiffness.

INTRODUCTION

The average height of buildings is continually increasing with the introduction of new materials and new technologies, resulting in the reduction of their natural resonant frequencies. With low resonant frequencies, tall buildings, especially those in typhoon-prone regions, are increasingly sensitive to wind loads. Therefore, the fluid–structure interaction (FSI) related to wind loads is an important consideration in the design of tall buildings (Kareem 1992; Chai et al. 1997). The aerodynamic damping caused by wind loads is deeply rooted in FSI and may be negative in some cases, which is potentially hazardous to structures (Gerges et al. 2003; Gu et al. 2004). Aerodynamic damping is complicated with respect to both formation mechanisms and impact analysis. It is mainly investigated by wind tunnel testing of rigid models and aeroelastic models (Kwok et al. 1995; Zhou et al. 2003). The low natural frequencies of rigid models lead to poor performance in wind tunnel testing, and the aeroelastic models are complex and difficult to build because they need to be in accordance with the dynamic characteristics of archetypes. The required damping and stiffness for wind tunnel testing of aeroelastic models cannot be accurately and efficiently adjusted with conventional experimental
techniques. Oil dampers and replaceable springs are common tools utilised for regulating damping and stiffness of aeroelastic models, respectively. On account of poor accuracy of damping regulation and the frequent need to change springs in these methods, the experimental process tends to be cumbersome and less reliable (Taniike et al 1988; Xu et al.1993; Gerges et al. 2003). For this purpose, this paper describes a vibration control system of 2 degrees of freedom (DOF), using electromagnetic technologies. The damping and stiffness of the proposed system can be tuned digitally. The system provides a reliable and convenient method to obtain the variation of aerodynamic damping under different wind conditions.

**SYSTEM STRUCTURE**

This system consists of a sensing module, a control centre with data acquisition, a power amplifier and a shaking table supported by four actuators. The sensing module includes four eddy current displacement sensors (MICRO-EPSILON, eddyNCDT3010). The structure of the 2-DOF shaking table is shown in Fig. 1 and Fig. 2.

![Figure 1. Structure of the shaking table](image1)

![Figure 2. Picture of the shaking table](image2)

Aeroelastic models that are installed on the pillar, which is fixed on the shaking table, can rotate around their X-axis and Y-axis along with the shaking table. Four electromagnetic actuators are attached to the table through connecting springs which provide initial stiffness for the vibration system. Four eddy current sensors are placed under the table to measure the displacement of the four corners of the shaking table. Two actuators at the opposite corners of the table act in synchronisation with each other to adjust damping and stiffness of the corresponding DOF.

This system can serve the needs of most high-rise buildings from 300 m to 600 m in wind tunnel testing of aeroelastic models. The range of motion for each actuator is ±0.5 mm. Each actuator generates a maximum damping force of 12.50 N and a maximum elastic force of 37.50 N for the installed aeroelastic model.

**DESIGN OF ACTUATORS**

**Simplified Model of the Vibration System**

The motions of each DOF are identical and are shown in Fig. 3. The aeroelastic model on the shaking table vibrates under wind loads in wind tunnel testing, and this can be regarded as a reciprocating motion. Due to its rather small displacement (±0.5 mm), the pendulum motion can be further reduced to simple harmonic vibration. The torsional stiffness $K$ of the total shaking components, which is also known as the minimum-order modal stiffness, can be expressed as
\[ K = I \times \omega^2 , \quad (1) \]

where \( I \) is the moment of inertia of the combined shaking components including the shaking table, the pillar and the aeroelastic model, and \( \omega \) is the angular frequency of vibration.

The torsional stiffness \( K \) can also be calculated with the individual stiffness \( k \) generated by either actuator, when a unit angular rotation is exerted, i.e., \( \theta = 1 \), as shown in Fig. 3,

\[ K = 2k \times (\theta \times B) \times B , \quad (2) \]

where \( B \) is the horizontal distance from the centre of the actuator to the rotation axis of the table. With Eqs. 1 and 2, the elastic force provided by each actuator follows

\[ F_e = kA = \frac{I\omega^2 A}{2B^2} , \quad (3) \]

where \( A \) is the displacement of the point where the force is generated by the actuator.

![Figure 3. Motion of one DOF on the vibration table](image)

The damping force \( F_c \) generated by each actuator can be described as

\[ F_c = -c \cdot v , \quad (4) \]

where \( c \) is the damping coefficient and \( v \) denotes the linear velocity of the point acting by the output force from the actuator to the shaking table. Here, \( v \) can be obtained through the derivation of measured displacement at the corner of the shaking table.

In this paper, a linear control algorithm is applied to adjust the damping and stiffness. In each actuator, two sets of coils, i.e., damping coils and stiffness coils, are employed to generate damping force and elastic force, which are proportional respectively to the linear velocity and vibration displacement at the corresponding point. Numerical simulation methods were used to estimate the size of the actuator and the number of turns of the coils. Correction factors were employed in the algorithm to ensure that all the four actuators have the identical output. Specifically, the actuator can create electromagnetic forces of 12.50 N and 37.50 N from damping coils and stiffness coils, respectively, with the current of 0.7 A.

**Control Scheme**

The flow chart of this vibration control system is presented in Fig. 4. Four eddy current displacement sensors are used to measure the vibration displacement of the shaking table. The processed signals are sent to the control center. Velocity data is acquired by the derivation of displacement signals. The linear control algorithm sends out control signals through 8 channels ranging from -10V to +10V. Here, 4 channels correspond to damping control and the other 4 channels correspond to the stiffness control. The power amplifier (200W, 1A, 8 Channels) was devised to drive the damping and stiffness coils with the control signals. The required forces for damping and stiffness were generated through the use of electrified coils around permanent magnets.
RESULTS AND DISCUSSION

Free vibration tests were conducted to examine the capacity of controlling damping and stiffness, as shown in Fig. 5. An initial displacement was given by applying a normal load at one corner of the shaking table without the building models. After releasing the load, we measured and recorded the displacement at this corner. The measured displacement was analysed through a logarithmic decrement method and Fast Fourier Transform (FFT) method to recognise the damping factor and natural frequency. The time evolutions of displacement under different damping output levels are presented in Fig. 6.
Figure 6. Displacement under different damping output levels: (a) without output of damping; (b) 30% of full-scale damping output; (c) 60% of full-scale output; (d) 90% of full-scale output

It can be clearly seen that the decay time shows a dramatic downward trend from around 16 sec to less than 2 sec when 90% of the damping output over the full range is applied. The system performance under controlled damping and stiffness with different output levels is presented in Fig. 7. All the average values of damping ratios and natural frequencies are obtained from over 10 release tests. When the output of the system is zero, the structural damping and the natural frequency of the total shaking components are 0.0176 Hz and 12.01 Hz, respectively. We can see that the experimental results are highly consistent with the theoretical values for both damping ratio and natural frequency. This system can also achieve negative damping and negative stiffness, which is useful in the wind tunnel testing of aeroelastic models. The structural damping of the proposed system for aeroelastic models can be extended to be below 0.1% (Xu et al.1993; Gerges et al. 2003). By utilising negative damping and stiffness available through this system, the requirement for having large ranges of system stiffness and damping can be considerably reduced for constructing aeroelastic models.

Figure 7. Damping and stiffness control results of different output of controlling variables

It is found that this system exhibits improved performance in terms of repeatability and stability, when the output levels of damping control are between 40% and 60% of their full range. While for the low output levels, the values of standard deviation are higher compared with the cases with the output range from 40% to 60%. The limitations of the resolution of data acquisition and the amplifier are probably the main cause of the underperformance in the regime of low output levels, where only very small forces are provided to control the damping.
With increasing output level (over 60%), this system shows less stability in stiffness control. With the displacement signals after FFT, we found that the frequencies of most noises were close to 120 Hz, which is similar to the natural frequency (121.17 Hz) of the shaking table. High levels of output are likely to cause the vibration of the shaking table, which disturbs the displacement measurement. Another possible explanation is that the inductance of coils results in a phase difference between electrified current in coils and control signals. The phase difference could be ignored for the cases of low level output, while the impact of the phase difference increases as the vibration frequencies rise.

CONCLUSIONS

This paper presents a vibration control system with adjustable electromagnetic damping and stiffness for studying aerodynamic damping of high-rise buildings. We took advantage of the force generated as the result of current in an electrified coil in the presence of a magnetic field to vary damping and stiffness, including negative values. The maximum damping force of 12.50 N and the maximum elastic force of 37.50 N can be produced by single electromagnetic actuator. The results agree well with the theoretical predictions for this control system. The system proposed performs well in controlling damping and stiffness when the vibration frequency is less than 50 Hz. Further development should include extending this vibration system to more general vibrating structures.

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