Computer Simulation Model of a Lift Car Assembly with an Active Tuned Mass Damper

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Abstract. In engineering systems a Passive Tuned Mass Damper (a secondary mass – spring - damper combination) is often used to reduce vibrations of a primary structure (main mass). In an Active Tuned Mass Damper (ATMD) arrangement vibrations of the main mass are attenuated when the secondary mass (referred to as an active mass) is actively controlled. The ATMD system is equipped with a controller, sensors and an actuator. The attenuation is achieved by the application of control force determined by a suitable feedback control algorithm. In this paper the ATMD method is considered to attenuate resonance vertical vibrations of a lift car assembly – suspension rope system during the lift travel, when the frequency of harmonic excitation acting upon the car assembly becomes near its natural frequency. A mathematical model with the optimal feedback gain calculated using Linear–Quadratic Regulator control law is developed. Then, a case study is presented in which computer simulation is carried out. The simulation results are discussed and the effectiveness of an active tuned mass damper system is demonstrated for a given set of lift system parameters.

1 INTRODUCTION

Excessive vibrations in a lift system compromise car ride quality and may lead to wear, fatigue, malfunctioning, failure and structural damage of the installation. The underlying causes of vibration are varied, including poorly aligned joints and imperfections of guide rails, eccentric pulleys and sheaves, systematic resonance in the electronic control system, and gear and motor generated vibrations [1]. In high-rise applications lifts are subject to extreme loading conditions. High-rise buildings sway at low frequencies and large amplitudes due to adverse wind conditions and the load resulting from the building sway excites the lift system. This results in large vibratory motions of lift ropes [2,3].

Vibration suppression (reduction) can be achieved through *passive*, *semi-active* and / or *active* control methods. In passive control the aim is to develop a design of the system in which amplitudes of vibration are limited through an optimal choice of mass, stiffness and damping characteristics. However, often the desired level of vibration reduction cannot be obtained by passive methods and in order to achieve high performance of the system active vibration control (AVC) strategies must be applied [4]. In active vibration control a set of actuators with external power supply is used to provide a force to the system in order to limit vibration amplitudes. In this approach a set of sensors and a suitable control algorithm (feedback/ feedforward) are used to determine the control force to be applied. For example, in lift systems an active vibration damper can be applied under the lift car, fitted between the floor and sling, to suppress its vertical vibrations [5].

In resonance conditions (when a structure is acted upon by a force whose frequency coincides with its natural frequency) vibration attenuation can be introduced by the application of an auxiliary spring – damper - mass combination (a dynamic vibration absorber) attached to the main structure (primary mass). The best vibration control effects are then achieved when the mass – spring –

damper parameters are optimally tuned. Thus, this implementation the absorber device is referred to as a *tuned mass damper* (TMD).

In order for the TMD to be effective the harmonic excitation should be well known and its frequency should not deviate from its constant value. If the driving frequency drift occurs or there are changes in the TMD characteristics, the tuning condition will not be satisfied and the primary mass will experience some vibration. Furthermore, the driving frequency might be shifted to one of the natural frequencies of the combined primary – secondary mass assembly and the system will be driven to resonance and potentially fail. In order to address these issues a semi-active or active TMD device can be applied.

In this paper the concept of active tuned mass damper (ATMD) is discussed in the context of lift applications. The principle of operation and an ATMD is explained and then its operation is demonstrated through a case study involving a lift car-suspension model subjected to resonance vibrations.

2 ACTIVE TUNED MASS DAMPER

Fig. 1 shows a schematic diagram of a structure equipped with an ATMD system. Vibrations x_1 of the main mass m_1, acted upon by an excitation force *f(t)*, are attenuated by the application of an actively controlled auxiliary mass m_2. The ATMD system is equipped with a controller C_o, sensors s_1, s_2 (typically accelerometers) and an actuator providing a control force *u(t)*. The active attenuation is achieved by the application of control force *u* determined by a suitable feedback control algorithm.

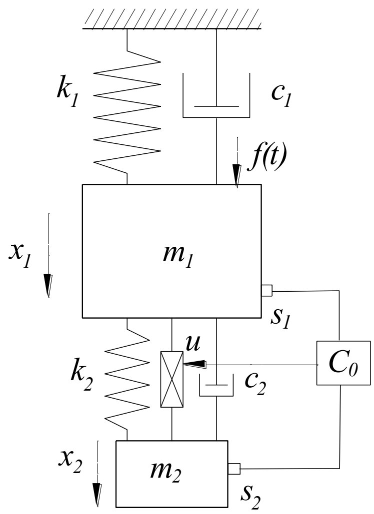

Figure 1 Schematic diagram of a structure equipped with an ATMD system

By introducing the state variable vector $\mathbf{x} = \left[x_1, x_2, \ldots\right]^T$, where x_1 and x_2 represent the absolute displacements of the main mass and the auxiliary mass, respectively, and the overdot denotes differentiation with respect to time *t*, the equations describing the dynamics of system can be written as [6]

$$(t) + \mathbf{B}_u u(t) + \mathbf{B}_f f(t) \qquad (1)$$

where the matrices **A** (the state matrix), $\mathbf{B}_u$ (the input matrix) and $\mathbf{B}_f$ are defined as follows

$$\mathbf{A}=\begin{bmatrix}\mathbf{0}_{2x2} & \mathbf{I}_{2x2}\\ -\tilde{\mathbf{K}} & -\tilde{\mathbf{C}}\end{bmatrix};\ \mathbf{B}_u=\begin{bmatrix}0 & 0 & -\frac{1}{m_1} & \frac{1}{m_2}\end{bmatrix}^T;\mathbf{B}_f=\begin{bmatrix}0 & 0 & \frac{1}{m_1} & 0\end{bmatrix}^T \tag{2}$$

with the mass-normalized stiffness and damping matrices $\tilde{\mathbf{K}}$ and $\tilde{\mathbf{C}}$ defined as

$$\tilde{\mathbf{K}}=\begin{bmatrix}\frac{k_1+k_2}{m_1} & -\frac{k_2}{m_1}\\ -\frac{k_2}{m_2} & \frac{k_2}{m_2}\end{bmatrix};\tilde{\mathbf{C}}=\begin{bmatrix}\frac{c_1+c_2}{m_1} & -\frac{c_2}{m_1}\\ -\frac{c_2}{m_2} & \frac{c_2}{m_2}\end{bmatrix} \tag{3}$$

where c_1, c_2 and k_1, k_2 are the coefficients of damping and stiffness, respectively. In a state feedback approach the control force is determined as $u(t)=-\mathbf{Gx}$, where $\mathbf{G}=\begin{bmatrix}g_1 & g_2 & g_3 & g_4\end{bmatrix}$ is a gain vector. The output equation is then given as $\mathbf{y}(t)=\mathbf{Cx}(t)$, where the constant output matrix is $\mathbf{C}=\begin{bmatrix}1 & 0 & 0 & 0\end{bmatrix}$. The closed-loop control system can then be represented by the block diagram shown in Fig. 2.

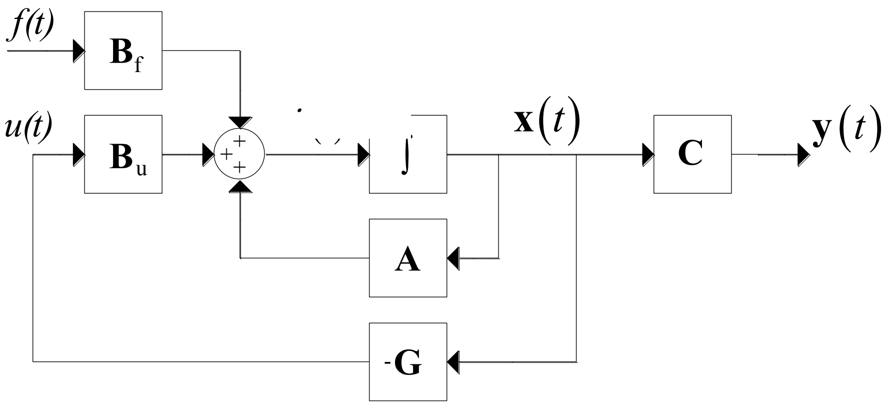

Figure 2 Closed-loop control system.

The most effective and widely used technique to determine the gain vector $\mathbf{G}$ and to obtain an asymptotically stable control system is the optimal Linear Quadratic (LQ) regulator [4]. This technique involves minimizing the cost functional (the quadratic performance index)

$$J=\int_0^\infty\left(\mathbf{x}^T\mathbf{Q}\mathbf{x}+Ru^2\right)dt \tag{4}$$

in order to determine the control u, where $\mathbf{Q}=diag\begin{bmatrix}q_1 & q_2 & q_3 & q_4\end{bmatrix}$ is the state cost matrix (with time invariant weights), and R is the control force weight. According to the LQ theory the optimum control law is then expressed as:

$$u(t)=-R^{-1}\mathbf{B}_u^T\mathbf{P}\mathbf{x}=-\mathbf{Gx} \tag{5}$$

where $\mathbf{P}$ is the solution of the Algebraic Riccati Equation (ARE, [4]).

3 LIFT MODEL

A lift car assembly – suspension rope model is depicted in Fig. 3. The combined mass of the assembly, denoted as M, is suspended on ropes of length L and mass per unit length m_r each (see

Fig. 3a). Fig. 3b shows a single-degree-of-freedom (SDOF) vibration model representing the fundamental vertical (bounce) mode with the overall motion denoted as x_c. In this mode both the car and sling move in phase and the effective modal mass is then determined using the kinetic energy expression corresponding to the vibration mode. The equivalent (effective) mass is given then as $M_e = M + n_r m_r L/3$, where n_r is the number of ropes. The flexibility of ropes is represented by a spring of effective coefficient of stiffness given as $k_e = n_r EA/L$, where EA is the product of modulus of elasticity and cross-sectional area of the ropes. Damping in this model is represented by a dashpot damper of the effective coefficient of viscous friction c_e.

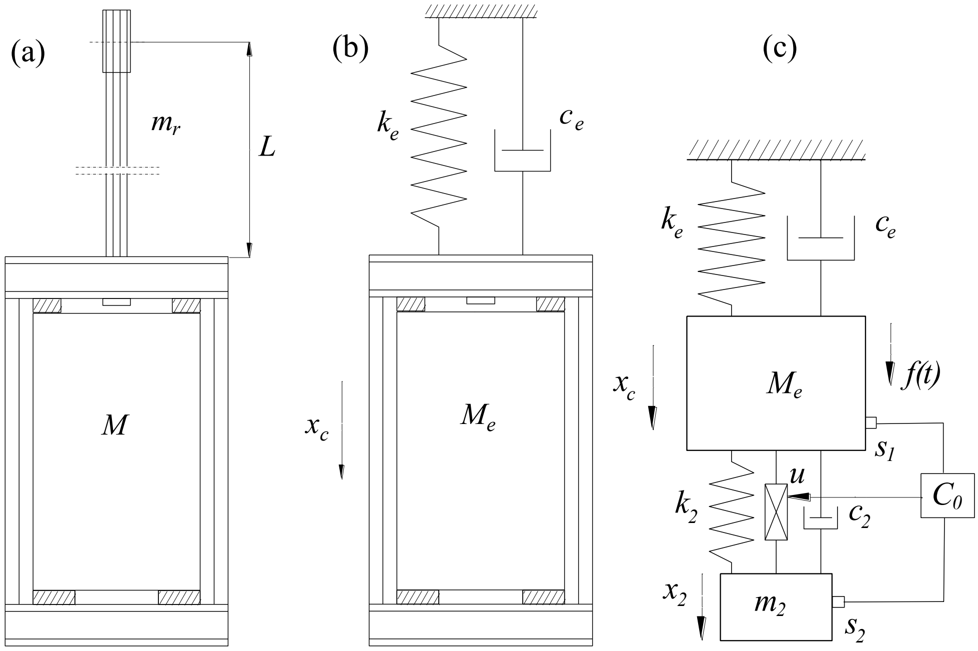

Figure 3 Lift car – suspension rope model.

Following what has been discussed above the application of ATMD can be considered to reduce vibrations of the car assembly. In schematic diagram shown in Fig. 3c an actively controlled auxiliary mass is fitted under the sling to implement the ATMD strategy.

4 NUMERICAL SIMULATION: CASE STUDY

The performance of an ATMD is demonstrated through a numerical simulation experiment. In the simulation the lift travels at rated speed V = 2 m/s. The car - sling mass is M = 1400 kg and the assembly is suspended on n_r = 4 steel wire ropes in 1:1 configuration. The ropes are of modulus of elasticity $E = 0.85 \times 10^5$ N/mm^2, mass per unit length m_r = 0.66 kg/m and effective area A = 69 mm^2 each. A scenario in which the car is subjected to harmonic excitation *f(t)* of frequency 3.7 Hz is considered in the test. The frequency of excitation becomes tuned to the natural frequency of car – suspension system during the lift travel when the length of the suspension ropes L is 30 m (see Fig. 4a). This results in resonance at the time instant of about 11.25 s and without application any active mitigation measures the car will suffer from excessive vibrations (with peak-to-peak displacements of over 3.4 mm, see the resonance region identified in Fig. 4b).

However, if ATMD is used and tuned according to possible resonance scenarios vibrations can be substantially reduced. In order to mitigate the effects of resonance in the above scenario, the lift performance is simulated when the car assembly is fitted with an ATMD system with moving mass m_2 = 71 kg equipped with an actuator capable of providing the maximum force of about 50 N, dictated by LQR algorithm. The damping ratio of the car – suspension system is assumed to be $\zeta_e = 5\%$ and the optimal value of damping ratio of the ATMD system is determined as $\zeta_2 = \sqrt{\frac{3\mu}{8(1+\mu)}}$, where $\mu = {m_2}/{M_e}$. The coefficient k_2 is determined as $k_2 = \frac{m_2}{1+\mu}\sqrt{\frac{k_e}{M_e}}$ [7]. The performance of the system is illustrated in Fig. 5.

(a) (b)

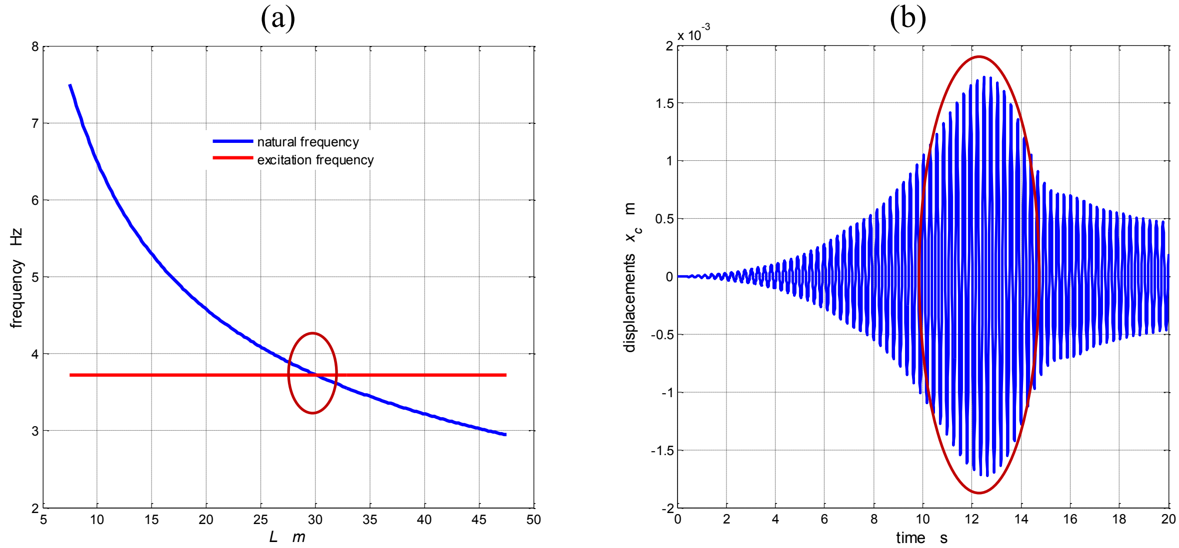

Figure 4 Lift resonance.

(a) (b)

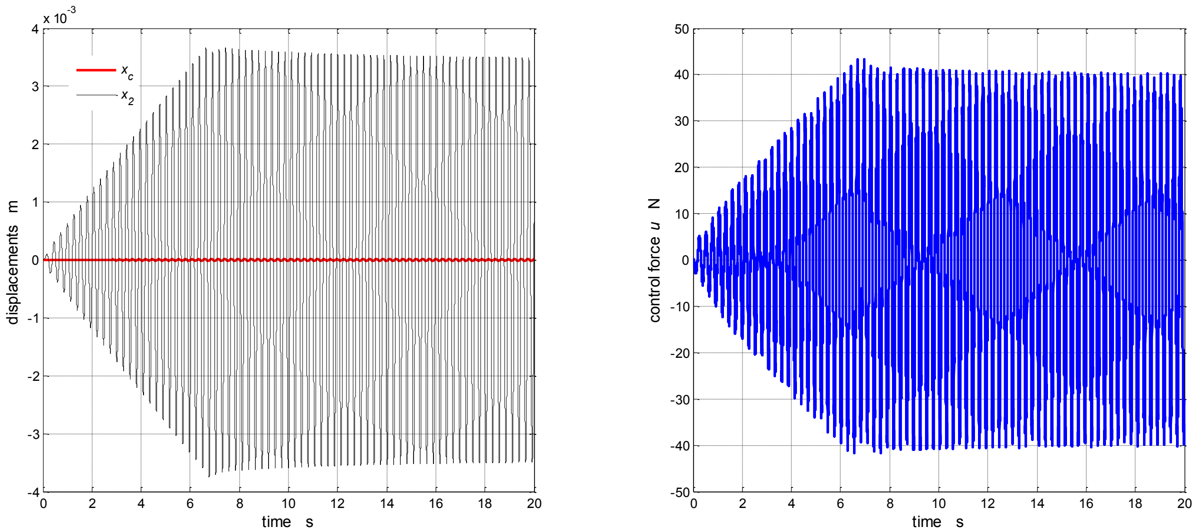

Figure 5 (a) Displacements of the car (red line), active mass (black line) (b) control force.

Fig. 5(a) shows that with the actuator providing a control force of t magnitude about 40 kN and the active mass peak-to-peak displacements of about 7 mm, the car vibrations can be eliminated. The results of numerical experiments will be illustrated with co-simulation and visualization using a model developed in a multibody system dynamics software environment and Matlab/ Simulink.

5 CONCLUSION

In a lift installation an adverse situation arises when one of the time-varying natural frequencies of the car – suspension rope system becomes near the frequency of a periodic excitation existing in the system. This results in a passage through resonance. In such a case the lift car will not vibrate throughout its travel, but will pass through a resonant vibration at some particular stage in the travel. Passive vibration isolation techniques are often applied to mitigate the effects of resonance. However, active vibration control methods can be used to control adverse dynamic behaviour of a lift. For example, resonance vibrations of a lift car can be attenuated by the application of a suitable ATMD system, as demonstrated by the results of numerical experiment carried out for a given set of lift system parameters.

REFERENCES

[1] S. Kaczmarczyk, "Vibration Problems in Lift and Escalator Systems: Analysis Techniques and Mitigation Strategies". The 3rd Symposium on Lift and Escalator Technologies, Northampton 26-27 September 2013.
[2] G.R. Strakosch, *The Vertical Transportation Handbook*. John Wiley & Sons Ltd., New York (1998).
[3] S. Kaczmarczyk, "The Nonstationary, Nonlinear Dynamic Interactions in Slender Continua Deployed in High-rise Vertical Transportation Systems in the Modern Built Environment". *Journal of Physics: Conference Series,* Vol. 382, (2012).
[4] A. Preumont, K. Seto, *Active Control of Structures.* John Wiley & Sons Ltd., New Delhi (2008)
[5] K. Funai et al., "The Development of Active Vibration Dampers for Super High-Speed Elevators". *The Proceedings of Elevcon 2004*, Istanbul, Turkey, 81-89.
[6] J. Wilkie, M. Johnson, R. Katebi, *Control Engineering: and Introductory Course.* Palgrave Macmillan, (2002).
[7] S.G. Kelly, *Fundamentals of Mechanical Vibrations*. McGraw-Hill, Inc., New York (1993).

BIOGRAPHICAL DETAILS

Stefan Kaczmarczyk is Professor of Applied Mechanics at the University of Northampton. His expertise is in the area of applied dynamics and vibration with particular applications to vertical transportation and material handling systems. He has been involved in collaborative research with a number of national and international partners and has an extensive track record in consulting and research in vertical transportation and lift engineering. He has published over 90 journal and international conference papers in this field.

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