The resonance conditions and application of passive and active control strategies in high-rise lifts to mitigate the effects of building sway

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Abstract. Tall buildings and high-rise structures are subjected to sway motions of large amplitude and low frequency due to the structural resonance conditions induced by wind loads and long-period seismic excitations. To mitigate the effects of resonance interactions on suspension and compensating ropes in a high-rise lift system the masses and geometry of the system can be adjusted to change the resonance frequencies to shift the resonance regions. However, in most cases the structural constraints and system design limitations do not leave much space for the possible changes to be effective. This paper revisits the system design parameters and how the simulation models and control strategies can be deployed to mitigate the effects of resonance conditions.

1 INTRODUCTION

In the modern high-rise lift installations traction drive systems are used. Strong wind conditions and earthquakes cause tall buildings to vibrate (sway) at low frequencies and large amplitudes which results in vibrations of car/ counterweight suspension ropes and compensating ropes. Under resonance conditions this results in complex dynamic interactions in the system [1,2].

Model-based control strategies can be developed to mitigate the effects of resonance interactions. In this paper, a simulation model to predict the dynamic responses taking place in a high-rise system is discussed and used in a case study to demonstrate and explain the interaction phenomena involved. It is then demonstrated how the design parameters can be optimised to minimize the effects of adverse dynamic responses, and possible control strategies are discussed.

2 SIMULATION MODEL

A schematic diagram of the dynamic model of the lift system is shown in Fig. 1. The modulus of elasticity, cross-sectional effective area and mass per unit length of the ropes are denoted as E_1, A_1, m_1 and E_2, A_2, m_2 for the compensating ropes and the suspension ropes, respectively. The compensating ropes are of length L_1 at the car side and the suspension ropes are of length L_2 at the counterweight side, respectively. The length of the suspension rope at the car side and the compensating rope at the counterweight side are denoted as L_3 and L_4, respectively. The lengths of suspension ropes and compensating cables are varying with the position of the car in the shaft (denoted by l_{car}). The masses and dynamic displacements of the car, counterweight and the compensating sheave assembly are represented by M_{car}, M_{cwt} and M_{comp}, q_1, q_2 and q_3, respectively. The compensating sheave rotational motion is represented by the angular coordinate θ and the second moment of inertia is I_{comp}. The compensating sheave assembly (CSA) is equipped with a hydraulic tie-down device. The building structure sway deformations, represented by the shape function $\Psi(\eta) = 3(\eta)^2 - 2(\eta)^3$, $\eta = z/Z_0$, results in harmonic motions $v_0(t)$ and $w_0(t)$ of frequency Ω_v and Ω_w, in the in-plane direction and out-of-plane direction, respectively.

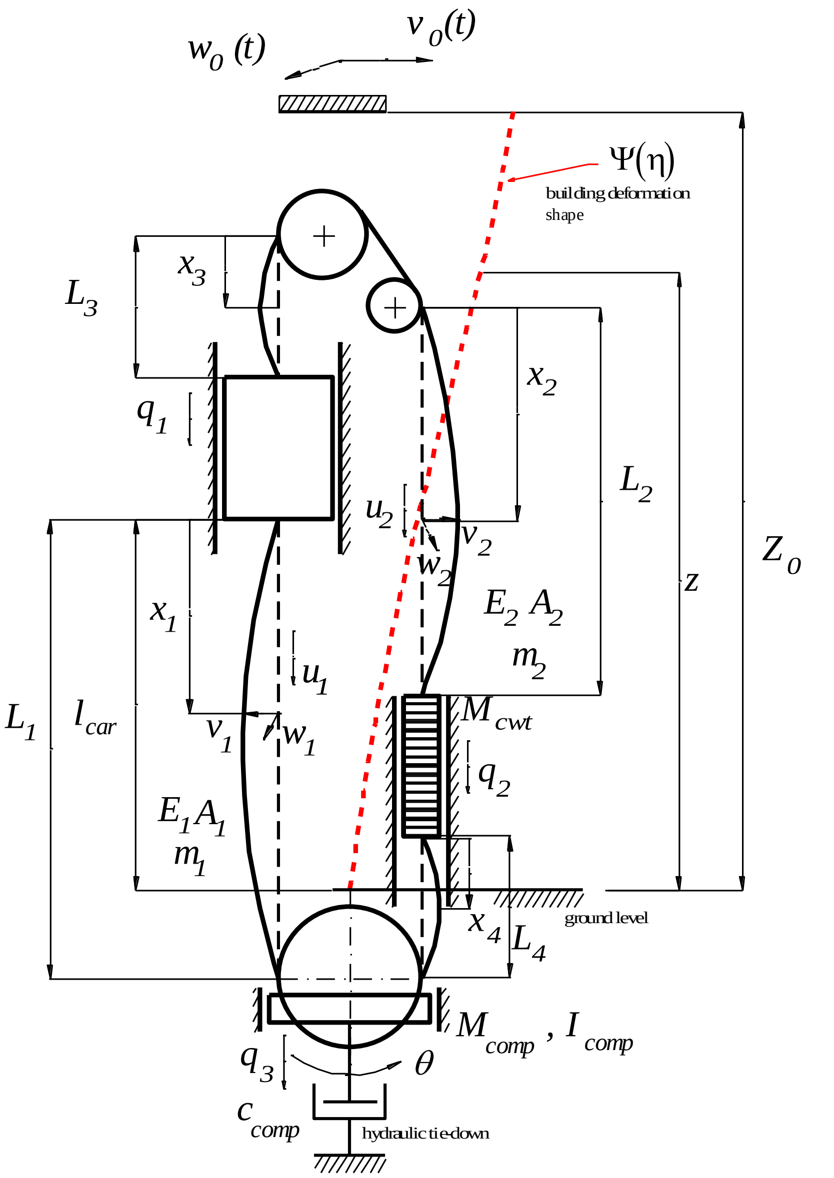

Figure 1: Model of a high-rise lift system with a hydraulic tie-down

The natural frequencies of the system change with the position of the car. An adverse situation arises when the building sways at its fundamental natural frequency which in turn is tuned to the natural frequency of the lift system, thus leading to resonance conditions. The resonance phenomena can be captured by the development of a suitable dynamic model. The model is represented by Eq. 1, where V, a represent the speed and acceleration/deceleration of the car, e_i denote the quasi-static axial strains in the ropes, $\bar{v}_i(x_i,t), \bar{w}_i(x_i,t)$, i = 1,2,…, 4, represent the dynamic displacements of the ropes, T_i, denote the rope quasi-static tension terms.

F_d is the damping force provided by the hydraulic tie-down of the constant damping coefficient c_{comp}, and α is a positive number $(0 < \alpha \leq 1)$. The longitudinal displacements of the compensating ropes and $x_1 = L_1$ and $x_4 = L_4$ at the car side and the counterweight side of the CSA are expressed in terms of the sheave vertical displacement and rotation as $u_1 = q_3 + R\theta$ and $u_4 = q_3 - R\theta$, respectively. Thus, the constrain relationship $2q_3 - u_1 - u_4 = 0$ is used in Eq. 1.

$$
\begin{aligned}
& m_i\bar{v}_{itt} - \left\{T_i - m_i\left[V^2 + (g - a_i)x_i\right] + E_iA_ie_i\right\}\bar{v}_{ixx} + m_ig\bar{v}_{ix} + 2m_iV\bar{v}_{ixt} = F_i^v\left[t, L_i(t)\right],\ i = 1,\ldots 4,\\
& m_i\bar{w}_{itt} - \left\{T_i - m_i\left[V^2 + (g - a_i)x_i\right] + E_iA_ie_i\right\}\bar{w}_{ixx} + m_ig\bar{w}_{ix} + 2m_iV\bar{w}_{ixt} = F_i^w\left[t, L_i(t)\right],\ i = 1,\ldots 4,\\
& M_{car}\ddot{q}_1 - E_1A_1e_1 + E_2A_2e_3 = 0;\\
& M_{cwt}\ddot{q}_2 - E_1A_1e_4 + E_2A_2e_2 = 0;\\
& M_{comp}\ddot{q}_3 + E_1A_1e_1 + E_1A_1e_4 + F_d = 0,\ F_d = c_{comp}\dot{q}_3\left|\dot{q}_3\right|^{\alpha-1}\\
& I_{c\,comp}\ddot{\theta} - RE_1A_1e_1 + RE_1A_1e_4 = 0,\\
& e_1 = \frac{1}{L_1(t)}\left[u_1(L_1,t) - q_1(t) + \frac{1}{2}\int_0^{L_1}\left(\bar{v}_{1x}^2 + \bar{w}_{1x}^2\right)dx_1 + \frac{\Psi_1^2}{2L_1(t)}\left(v_0^2 + w_0^2\right)\right],\\
& e_2 = \frac{1}{L_2(t)}\left[q_2(t) + \frac{1}{2}\int_0^{L_2}\left(\bar{v}_{2x}^2 + \bar{w}_{2x}^2\right)dx_2 + \frac{(\Psi_h - \Psi_2)^2}{2L_2(t)}\left(v_0^2 + w_0^2\right)\right],\\
& e_3 = \frac{1}{L_3(t)}\left[q_1(t) + \frac{1}{2}\int_0^{L_3}\left(\bar{v}_{3x}^2 + \bar{w}_{3x}^2\right)dx_3 + \frac{(\Psi_{car} - \Psi_{mach})^2}{2L_3(t)}\left(v_0^2 + w_0^2\right)\right],\\
& e_4 = \frac{1}{L_4(t)}\left[u_4(L_4,t) - q_{M2}(t) + \frac{1}{2}\int_0^{L4}\left(\bar{v}_{4x}^2 + \bar{w}_{4x}^2\right)dx_4 + \frac{\Psi_{cwt}^2}{2L_4(t)}\left(v_0^2 + w_0^2\right)\right],
\end{aligned} \tag{1}
$$

The system of Eq. 1 is discretized by using the Galerkin method [3] so that the resulting set of nonlinear ordinary differential equations (ODEs) can be simulated numerically.

3 NUMERICAL SIMULATION AND ANALYSIS

To demonstrate the dynamic behavior, the simulation is implemented for a high-rise installation roped *1:1* with the car of mass 5500 kg carrying rated load of 2620 kg. The travel height is H = 300 m and the installation is equipped with compensating ropes with a synthetic fiber core (SFC) of diameter 36 mm and mass per unit length m_{cr} = 4.76 kg/m each. The car and counterweight (balanced at 50%) are suspended on 9-stranded steel core ropes of diameter 19 mm and mass per unit length m_{sr} = 1.54 kg/m each [4]. The horizontal (bending mode) natural frequencies (eigenfrequencies) of the building structure are given as $\Omega_v = 0.1$ Hz in the in-plane direction and $\Omega_w = 0.15$ Hz in the out-of-plane direction, respectively

Fig. 2 shows the variation of the first two lateral natural frequencies (ω_1, ω_2) of the compensating ropes. The frequency curves are plotted against the position of the car in the shaft, with the in-plane and out-of-plane excitation frequencies represented by red horizontal lines, respectively. At the car side it is evident that when the car is approaching the top landing the in-plane excitation frequency is tuned to the first natural frequency of the ropes and the fundamental resonance takes place. Simultaneously the out-of-plane excitation frequency becomes close to the second natural frequency of the ropes which will activate the second resonance. Similar resonance effects are taking place with the car at the bottom landing, when the resonances occur at the counterweight side.

Fig. 3 shows the variation of the first two lateral natural frequencies of the suspension ropes at the car side. The plots demonstrate that when the car is at the bottom landing, the out-of-plane excitation frequency is tuned to the second natural frequency of the ropes, which results in the second mode resonance.

Another possibility of resonance interactions arises when the out-of-plane excitation frequency is tuned to the fundamental natural frequency of the counterweight suspension ropes, with the car positioned at the top landing (see Fig. 4).

The variations of the first four vertical mode natural frequencies ($\hat{\omega}_i$, $i = 1,\ldots,4$) are illustrated in Fig. 5. It is evident that those frequencies are much higher than the resonance frequencies of the building structure.

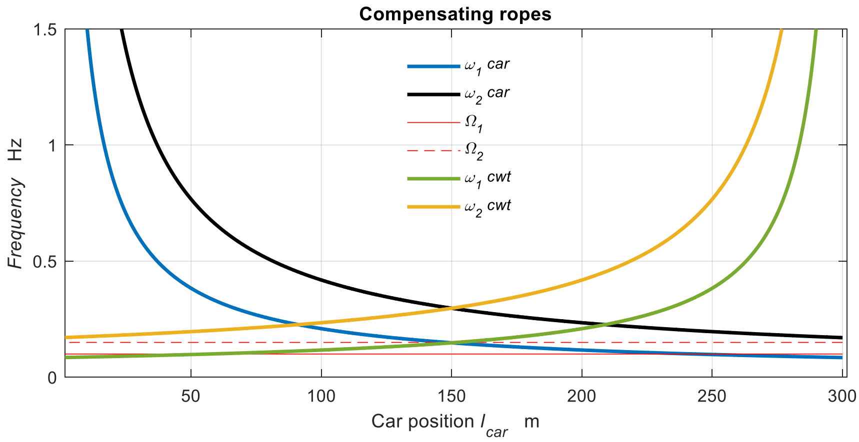

Figure 2: The natural frequencies of the compensating ropes

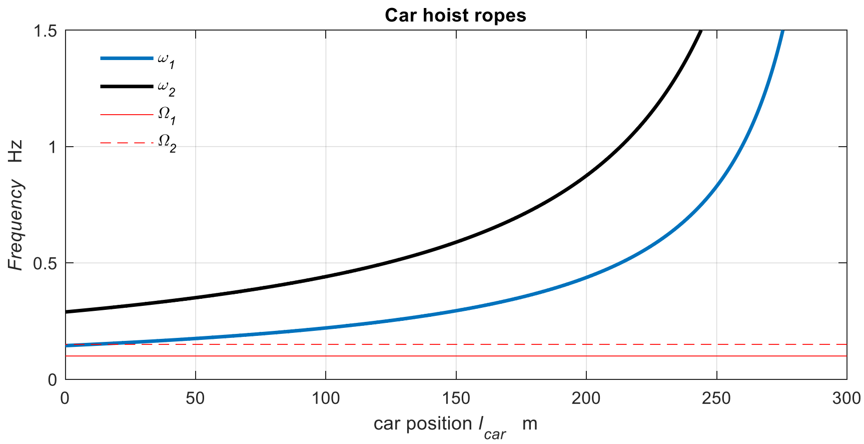

Figure 3: The natural frequencies of the suspension ropes at the car side

The effects of resonances that take place when the car is at the bottom landing are demonstrated by simulated records of the dynamic responses presented in Fig. 6 – 8. The building structure sways at the amplitudes of 0.9 m (in-plane) and 0.2 m (out-of-plane), respectively. The hydraulic tie-down damping force characteristic curve is shown in Fig. 9. The out-of-plane displacements of the car suspension ropes presented in Fig. 6 show the effects of the resonance when the out-of-plane building frequency (0.15 Hz) becomes near the fundamental frequency of the ropes. The in-plane displacements of the compensating ropes at the counterweight side presented in Fig. 7 illustrate the effects of the resonance condition when the in-of-plane building frequency (0.1 Hz) becomes near the fundamental frequency of the ropes. The FFT spectra shown in red in Fig. 6/7 demonstrate the resonance frequency tunings (0.15 Hz and 0.1 Hz, respectively). The lateral responses of the ropes are coupled with the vertical motions of the car, counterweigh and the CSA (see Fig. 8). The damping action of the hydraulic tie-down is evident from the subplot (c), where the displacements of the CSA are almost zero.

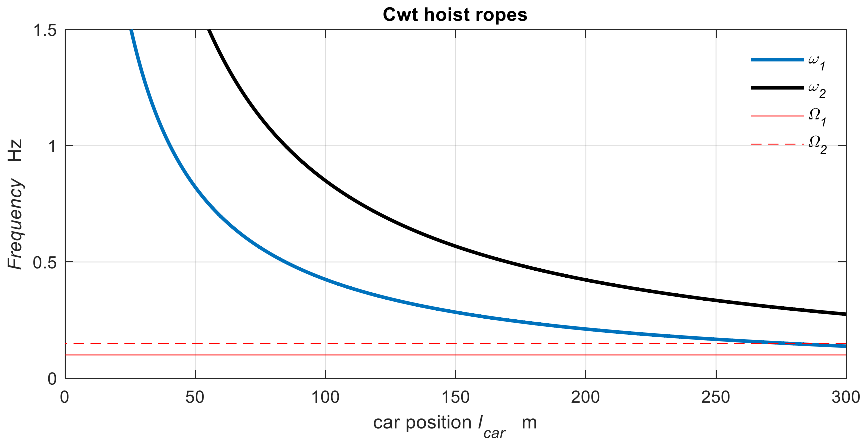

Figure 4: The natural frequencies of the suspension ropes at the cwt side

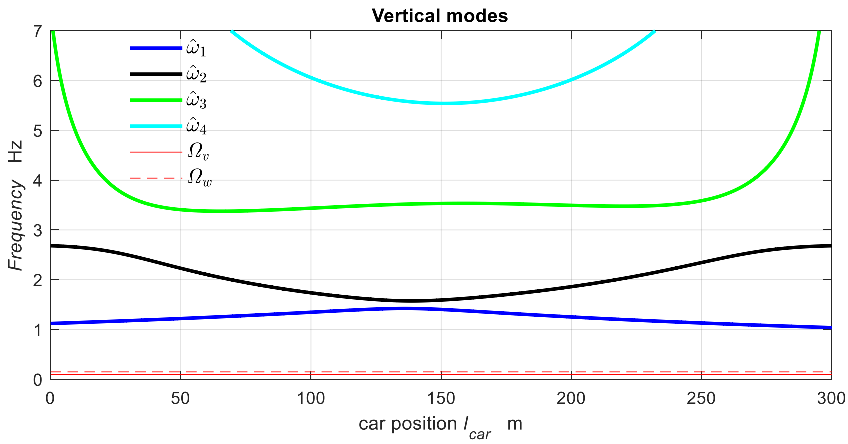

Figure 5: The natural frequencies: vertical modes

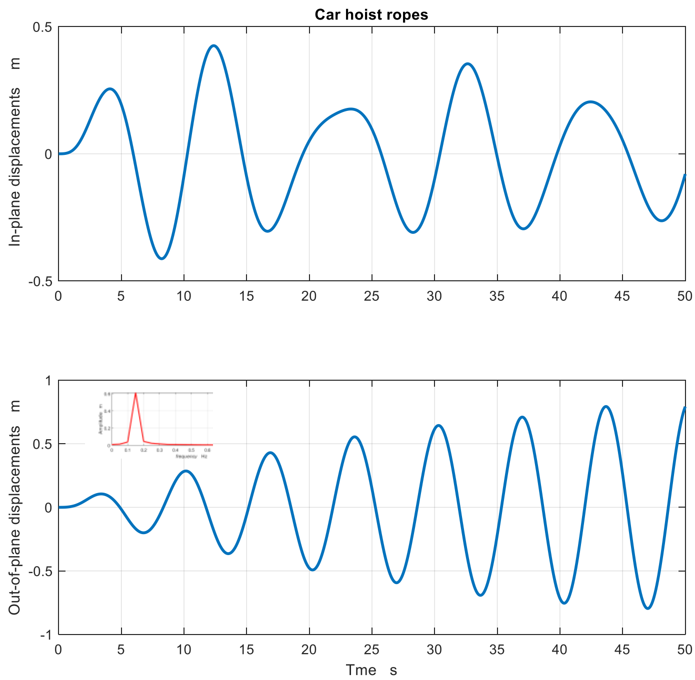

Figure 6: Displacements of the car suspension ropes at x_3 = 155 m (in-plane), 164 m (out-of-plane)

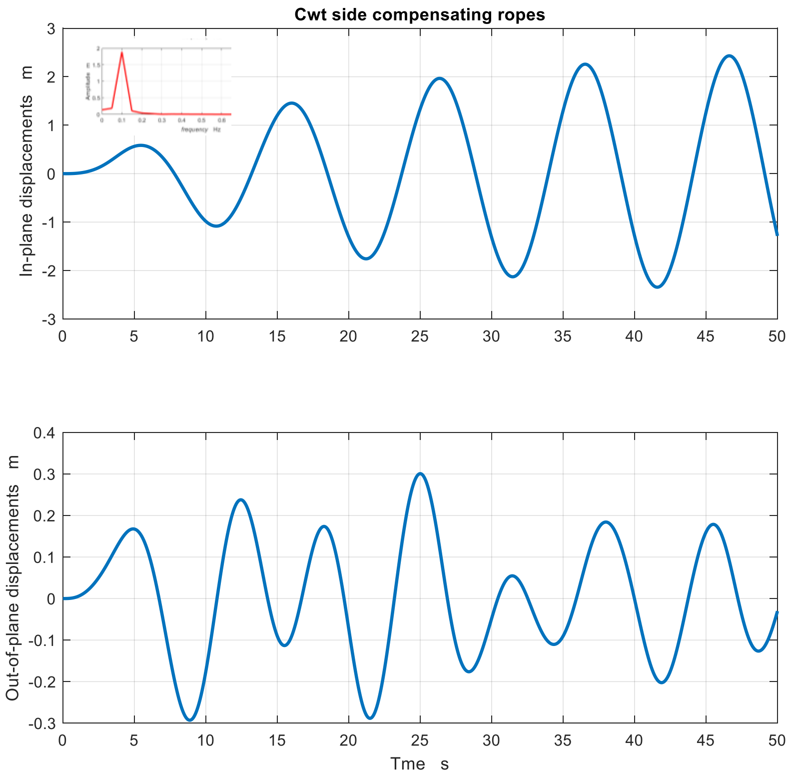

Figure 7: Displacements of the cwt compensating ropes at x_4 = 183 m (in-plane), 223 m (out-of-plane)

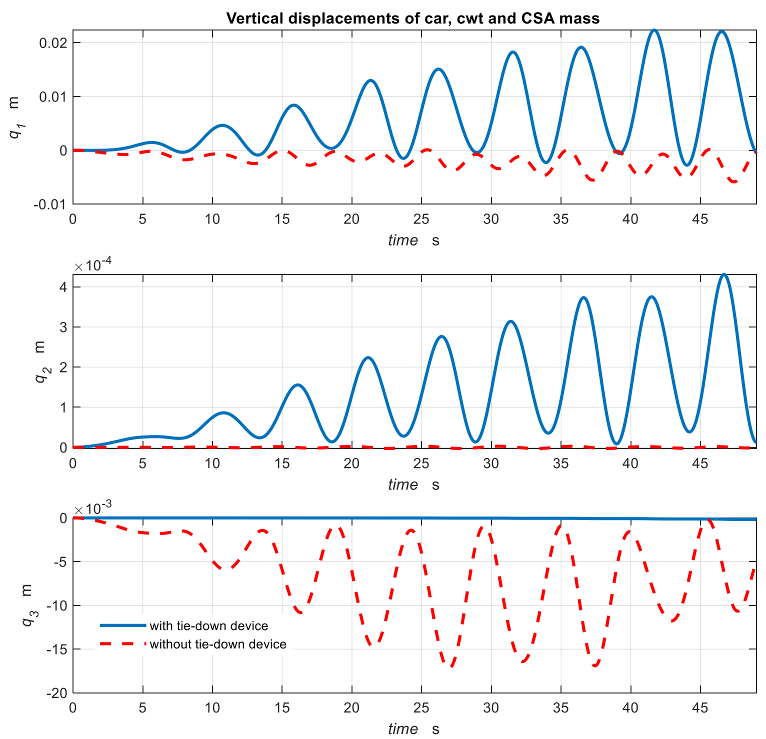

Figure 8: Vertical displacements of the car (q_1), counterweight (q_2) and compensating sheave (q_3) with and without hydraulic tie down

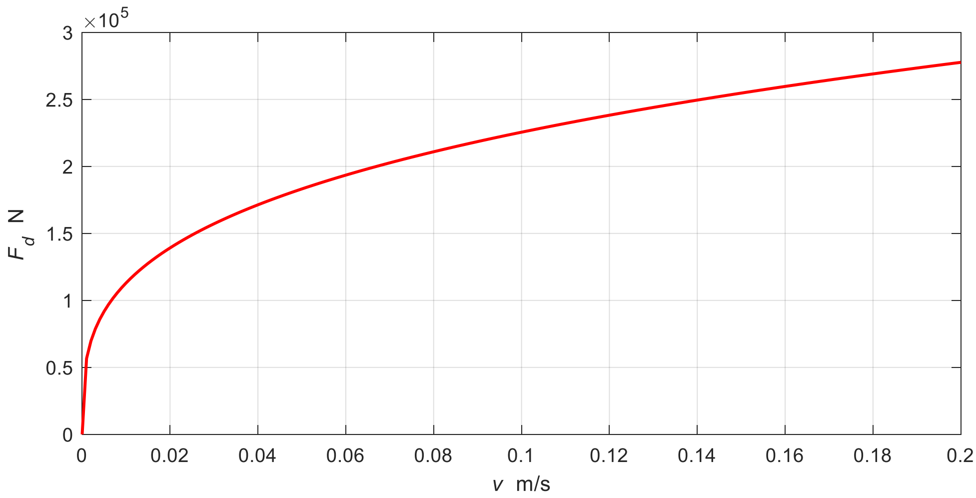

Figure 9: Hydraulic tie-down speed-force characteristic curve

4 DESIGN AND CONTROL STRATEGIES TO MITIGATE THE EFFECTS OF RESONANCE CONDITIONS

The modelling and simulation techniques are used to predict a range of dynamic interaction and resonance phenomena. This in turn informs the system design strategies. The application of a passive hydraulic tie-down device of suitable dynamic characteristics is effective in reducing the vertical motions of the CSA. However, this will not mitigate the effects of resonance conditions affecting the rope dynamics.

Various ways to limit large motions of lift ropes in high-rise applications due to low frequency building sways can be considered. Gibson [5] and Caporale [6] as well as Sun [7] discussed the use of a car follower to restrain the movements of ropes. This approach was used to control excessive rope and travelling cable sway in the World Trade Centre Towers in New York as well as in the Sears Tower in Olympia and Center Building in Chicago.

The resonance frequencies of the ropes can be shifted / changed by the use of different masses of the compensating sheave assembly. The masses of the compensating sheave assemblies can be increased or decreased in order to shift the resonance conditions. The number of ropes and their characteristics would then need to be considered. This would in turn trigger checking the system design parameters. Relevant calculations need then to be carried out to ensure that the minimum values of factors of safety and the traction conditions/ requirements would satisfy the safety regulations [8].

It should also be considered that the nature of the dynamic conditions present in the building structure is such that a small change in the natural frequencies of the structure might result in large changes of the resonance conditions. The overall stiffness of the structure depends on a number of factors and there might be some uncertainties about the final values of the structure eigenfrequencies. Thus, it is important to be aware that the natural frequencies of the structure might change with time.

Passive methods might involve the application of viscous dampers placed near the rope terminations at the car/ counterweight and acting in a lateral direction [9]. Semi-active control strategies include the application of magnetorheological dampers that achieve significant vibration reduction compared to viscous dampers [10]. The application of transverse tuned mass damper (TMD) technologies can reduce the dynamic responses in the system [11]. More recently passive negative stiffness control technique has been considered [12].

Active vibration control methods using boundary lateral motion [13,14] or longitudinal motion [15-16] have also been considered. The latter strategy utilizes the fact that the longitudinal elastic stretching of the slender element is coupled with its lateral motion. An actuator is used to produce a longitudinal oscillatory motion of the support in order to cause the time variation of transverse (lateral) stiffness which in turn results in extracting energy from the system. Such an active control method is termed *active stiffness control* [17].

The active stiffness method can be applied to minimize the effects of adverse dynamic responses of suspension and compensating in lift systems [18-20]. The means to induce a variation of the rope tension of the compensation rope comprises at least one servo actuator. For example, the system shown in Fig. 10 is equipped with a servo actuator to produce the control vertical motion $\underline{u_{comp}}$ to adjust the position of the CSA. The motion of the CSA is dictated by a suitable feedback control law. Fig. 11 show the maximum displacements of the compensating ropes and demonstrate the effectiveness of this approach when a multimode feedback law is applied. This law is implemented to reduce the resonance response of the compensating ropes when the car position corresponds to the distance l_{car} = 57 m. At this position the compensating ropes are subjected to the fundamental resonance condition (see Fig. 2). A multimode feedback law applied is given as

$$u_{comp}(t) = a_u \frac{\sum_{n=1}^{N} q_n \dot{q}_n}{\sum_{n=1}^{N} \alpha_n^2 q_n^2} \tag{2}$$

where a_u is the control factor q_n represent the modes of the compensation rope system and α_n are the mode weighting coefficients. The results presented in Fig. 11 show that the application of the tie down passive system results in smaller displacements (the line in blue). The active control results in a more substantial reduction of the rope displacements (shown in red). The control motion of the CSA (shown in green) is generated by using a_u = 0.5 in (2). The control law accommodates all in-plane and out-of-plane modes of the ropes so that the modal spillover phenomenon [17] is avoided.

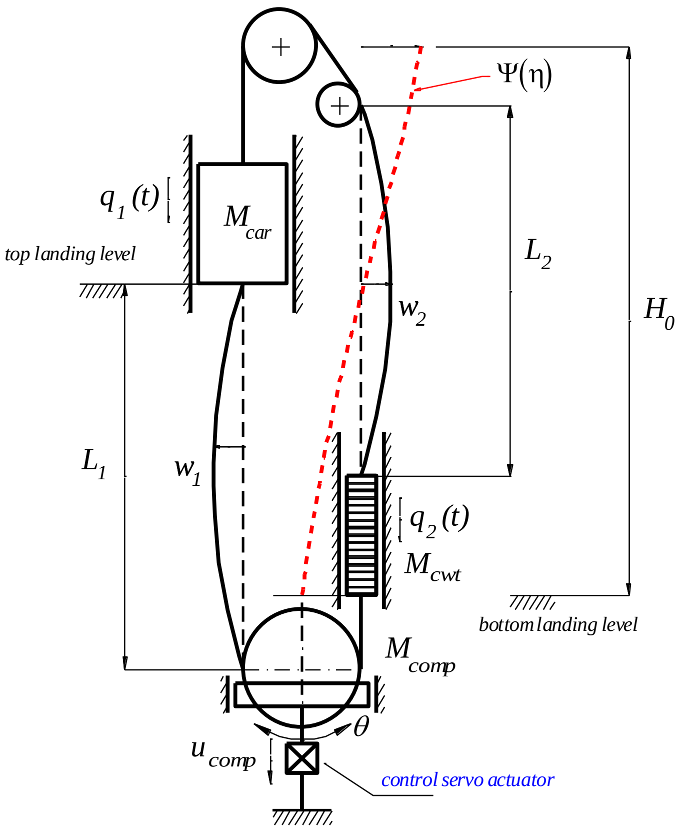

Figure 10: Active control strategy

5 CONCLUDING REMARKS

Numerical simulation results presented in this paper show the effect of resonance conditions on the dynamic responses of high-rise lift systems arising due to the sway of the host building structure. The system suffers from large lateral displacements of the suspension and compensating ropes that often exceed allowable limits. These responses are coupled with the vertical motions of the car. Counterweight and CSA. The simulation results inform the development of measures to be taken to mitigate the effects of adverse dynamic interactions that arise. Various passive methods can be deployed to mitigate the effects of resonance conditions that arise in the system. However, it should be noted that the nature of the dynamic conditions present in high-rise building systems is such that small changes of the natural frequencies of the structure might result in large changes of the resonance conditions that arise in the lift installation. Thus, more advanced strategies, such as the active stiffness method can be developed to minimize the effects of adverse dynamic responses of the system.

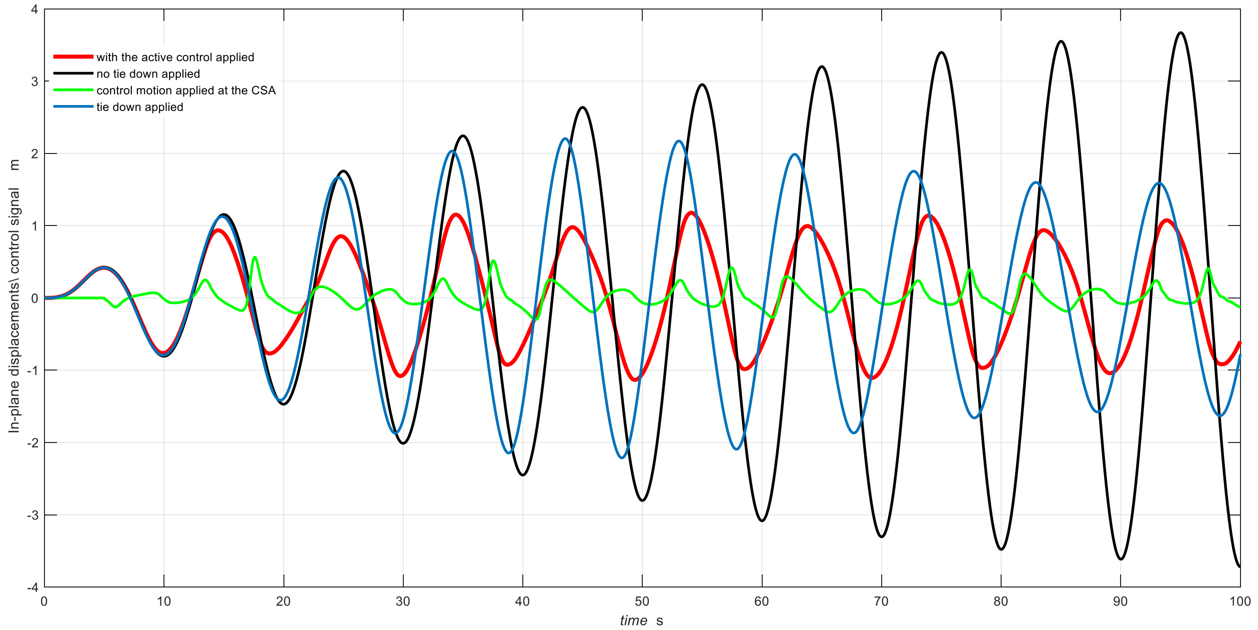

Figure 11: Effectiveness of active control strategy applied to reduce the response of compensating ropes

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BIOGRAPHICAL DETAILS

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