The Dynamic Interactions in High-Rise Vertical Transportation Systems

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Abstract. High speed and high capacity vertical transportation (VT) installations in the modern built environment service buildings of nearly 1000 m tall. Tall buildings are susceptible to large sway motions when subjected to wind loading or earthquake excitations. The low frequency sway motions cause resonance interactions in lift car/ counterweight suspension system, compensating ropes and overspeed governor ropes. This leads to poor ride quality and a high level of dynamic stresses which may result in damage to the installation. This paper presents the systems engineering approach to predict and quantify transient and steady-state resonant vibrations taking place in high-rise lift applications. The results and conclusions presented in the paper demonstrate that a good understanding of the dynamic behaviour of VT systems is essential for developing design strategies that minimize the effects of adverse dynamic responses so that the installation will operate without compromising the structural integrity and safety standards.

1 INTRODUCTION

The design and operation of high-performance systems for passenger transportation in the modern high-rise built environment present many technical challenges due to adverse dynamic responses and interactions that often arise due to various sources of excitation present in these systems [1]. In the modern high-rise built environment traction drive lift (elevator) systems are used.

The performance of lift installations in tall towers and buildings can be substantially affected by the behaviour of the host structure adverse environmental phenomena [2,3]. Strong wind conditions and earthquakes cause tall buildings to vibrate (sway) at low frequencies and large amplitudes creating a base motion excitation mechanism acting upon all building non-structural components. When the host (building) structure sways a broad range of resonance phenomena occur in the lift system with large whirling motions of ropes and cables being developed that are coupled with vertical motions of the car, counterweight and compensating sheave and often result in damage caused by the impact against the lift equipment located in the shaft and/or against the shaft walls. Aerodynamic phenomena affect the performance of high-speed lift systems. At high speeds the air flow around the car – frame assembly induces excessive vibrations and noise [4,5]. During the lift travel large air pressure differences between the front and rear of the car are being generated. Furthermore, the aerodynamic effects due to multiple cars running in the same shaft should also be considered [6,7].

This paper reviews and presents a systematic analytical approach to predict and quantify the transient and steady-state resonant vibration phenomena taking place in high-rise lift systems due to of the host structure motions. It is demonstrated that a good understanding of the dynamic behaviour of the main components of the lift systems is essential for developing design strategies that minimize the effects of adverse dynamic responses so that the installation will operate without compromising the structural integrity and safety standards.

2 DYNAMICS OF A HIGH-RISE LIFT INSTALLATION

The traditional traction-driven high-rise lift installation comprises the lift car/ counterweight system driven by tractive forces developed between the traction shave and the suspension means - long slender continua (LSC) such as steel wire ropes (SWR) or light-weight composite ropes [7]. In this arrangement an additional set of ropes, tensioned by the weight of the compensating sheave, are used for the compensation of tensile forces over the traction sheave.

2.1 Dynamic model with base excitation due to the structure motion

A schematic diagram of the dynamic model of the lift system is shown in Fig. 1. The modulus 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 time-varying $L_i = L_i(t)$, $i = 1,\ldots,4$. 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_{M1}, q_{M2} and q_{M3}, respectively. The speed and acceleration/ deceleration of the car are denoted by V and a, respectively.

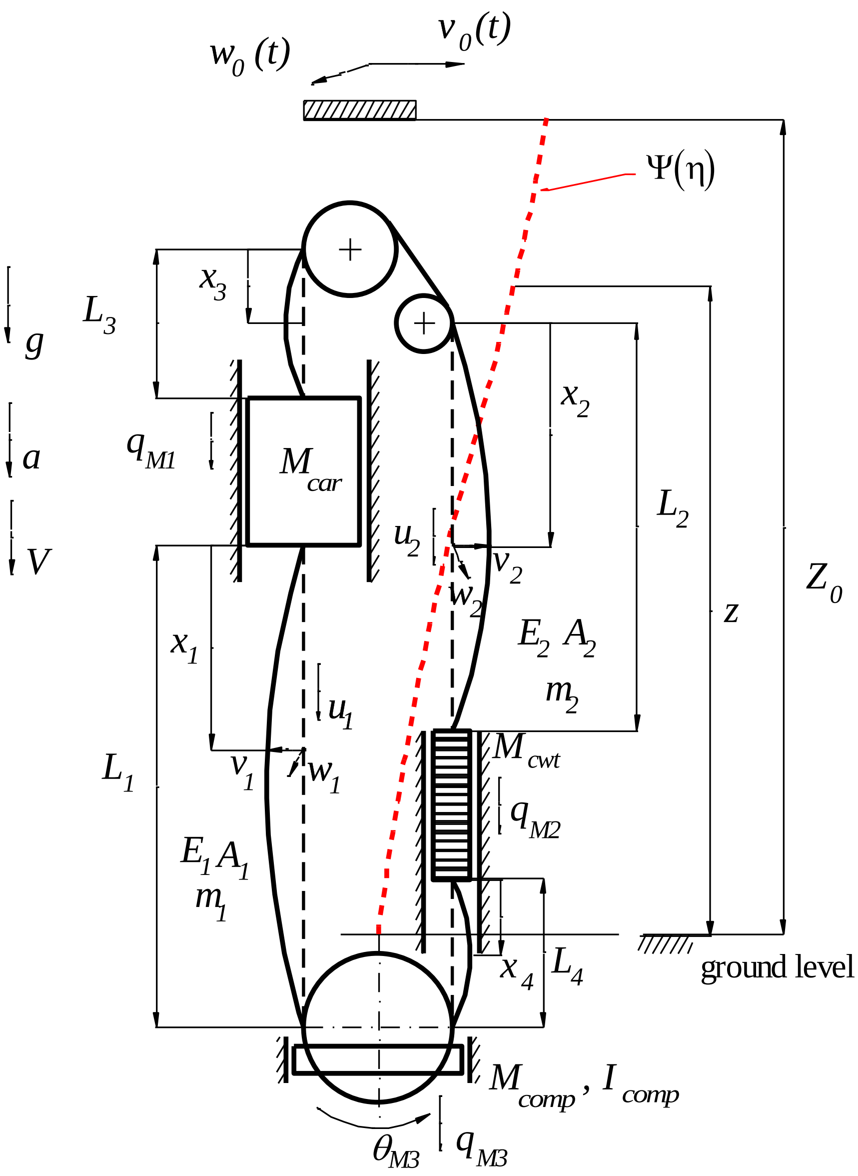

Figure 1 Simplified model of a high-rise lift system

The base motion excitation due to the building structure sway, which results in the in-plane motion $v_0(t)$ and out-of-plane motion $w_0(t)$ at the building top level, acts upon the suspension ropes and compensating cables that suffer from large dynamic displacements. Due to the variation of their lenghths the natural frequencies change during travel, rendering the system nonstationary. An adverse situation arises when the building is excited its natural frequency and vibrates periodically. This in turn may result in external, parametric and internal resonances in the lift system.

2.2 Mathematical model

Eqs (1) represent the mathematical model, based on the diagram in Fig. 1, with the excitation mechanism expressed by functions defined in terms of the deformations of the building represented by the shape function $\Psi\left(z/Z_0\right)$.

$$
\begin{aligned}
& m_i\bar{v}_{itt}-\left\{T_i-m_i\left[V^2+\left(g-a_i\right)x_i\right]+E_iA_ie_i\right\}\bar{v}_{ixx}+m_ig\bar{v}_{ix}+2m_iV\bar{v}_{ixt}=F_i^v\left[t,L_i\left(t\right)\right],\ i=1,\ldots 4,\\
& m_i\bar{w}_{itt}-\left\{T_i-m_i\left[V^2+\left(g-a_i\right)x_i\right]+E_iA_ie_i\right\}\bar{w}_{ixx}+m_ig\bar{w}_{ix}+2m_iV\bar{w}_{ixt}=F_i^w\left[t,L_i\left(t\right)\right],\ i=1,\ldots 4,\\
& M_{car}\ddot{q}_{M1}-E_1A_1e_1+E_2A_2e_3=0;\\
& M_{cwt}\ddot{q}_{M2}-E_1A_1e_4+E_2A_2e_2=0;\\
& M_{comp}\ddot{q}_{M3}+E_1A_1e_1+E_1A_1e_4=0,\\
& I_{comp}\ddot{\theta}_{M3}-RE_1A_1e_1+RE_1A_1e_4=0,\\
& e_1=\frac{1}{L_1\left(t\right)}\left[u_1\left(L_1,t\right)-q_{M1}\left(t\right)+\frac{1}{2}\int_0^{L_1}\left(\bar{v}_{1x}^2+\bar{w}_{1x}^2\right)dx_1+\frac{\Psi_1^2}{2L_1\left(t\right)}\left(v_0^2+w_0^2\right)\right],\\
& e_2=\frac{1}{L_2\left(t\right)}\left[q_{M2}\left(t\right)+\frac{1}{2}\int_0^{L_2}\left(\bar{v}_{2x}^2+\bar{w}_{2x}^2\right)dx_2+\frac{\left(\Psi_h-\Psi_2\right)^2}{2L_2\left(t\right)}\left(v_0^2+w_0^2\right)\right],\\
& e_3=\frac{1}{L_3\left(t\right)}\left[q_{M1}\left(t\right)+\frac{1}{2}\int_0^{L_3}\left(\bar{v}_{3x}^2+\bar{w}_{3x}^2\right)dx_3+\frac{\left(\Psi_{car}-\Psi_{mach}\right)^2}{2L_3\left(t\right)}\left(v_0^2+w_0^2\right)\right],\\
& e_4=\frac{1}{L_4\left(t\right)}\left[u_4\left(L_4,t\right)-q_{M2}\left(t\right)+\frac{1}{2}\int_0^{L4}\left(\bar{v}_{4x}^2+\bar{w}_{4x}^2\right)dx_4+\frac{\Psi_{cwt}^2}{2L_4\left(t\right)}\left(v_0^2+w_0^2\right)\right],\\
& 2q_{M3}-u_1\left(L_1,t\right)-u_4\left(L_4,t\right)=0
\end{aligned}
\tag{1}
$$

In this model e_i denote the quasi-static axial strains in the ropes, $\bar{v}_i\left(x_i,t\right),\bar{w}_i\left(x_i,t\right)$, represent the dynamic displacements of the ropes, T_i, denote the rope quasi-static tension terms, and a_i are the acceleration / deceleration rates of the car /counterweight, where i = 1,2,…, 4. The continous slowly varying nonlinear system (1) is discretized by using the Galerkin method [2] and the resulting ordinary differential equation (ODE) set of nonlinear equations is solved numerically.

3 CASE STUDY AND NUMERICAL RESULTS

Fig. 2 and Fig. 3 show the variation of the first two lateral natural frequencies (ω_1, ω_2) of the compensating ropes and the suspension ropes, respectively, in a lift installation servicing a 250 m tall building. The ropes are of standard SWR type and their natural frequencies are plotted against the corresponding length of each rope section, with the in-plane and out-of-plane excitation frequencies (Ω_v, Ω_w; Ω_v<Ω_w) represented by red horizontal lines, respectively.

On the other hand, the variations of the first four vertical mode natural frequencies ($\hat{\omega}_i,\ i=1,\ldots,4$) are illustrated in Fig. 4. It is evident that those frequencies are much higher than the resonance frequencies of the building structure ($\hat{\omega}_i/\Omega_{v,w} > 3.5,\ i=1,2,3$). It can be observed that there are length regions where the lateral fundamental and secondary resonance conditions arise in the system. For example, when the length of the car suspension rope is about 153 m the fundamental resonance of the car suspension system is taking place. The in-plane and out-of-plane displacements of the ropes are shown in Fig. 5. It can be observed that the displacements grow in time due to the resonance condition. The lateral responses of the ropes are coupled with the vertical motions of the car, counterweight and the compensating sheave assembly. These motions are shown in Fig. 6 vs time, with the frequency spectrum (obtained by the application of fast Fourier transform) illustrated in Fig. 7. It is evident that substantial motions of the vertical masses occur, with the dominant frequency being twice the frequency of the building sway. The dynamic interactions in the system result in tubular (whirling) motions of the ropes which is illustrated in Fig. 8 where the trajectory of the displacements of mid-span section (in-plane response vs. the out-of- plane response) are plotted.

4 CONCLUDING REMARKS

The dynamic behaviour of a high-rise lift system subjected to excitations arising from the building motions can be analysed by the application of suitable models. The models and simulation techniques can then be used to predict a range of dynamic interaction and resonance phenomena. The resonance frequencies of the ropes can be shifted / changed by the use of different masses of the compensating sheave assembly. The frequencies of the suspension ropes depend on the mass/ weight of the car (and the corresponding mass of the counterweight) as well as on the car loading conditions. Thus, more advance strategies, such as the active stiffness method [8], can be developed to minimize the effects of adverse dynamic responses of the system. 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.

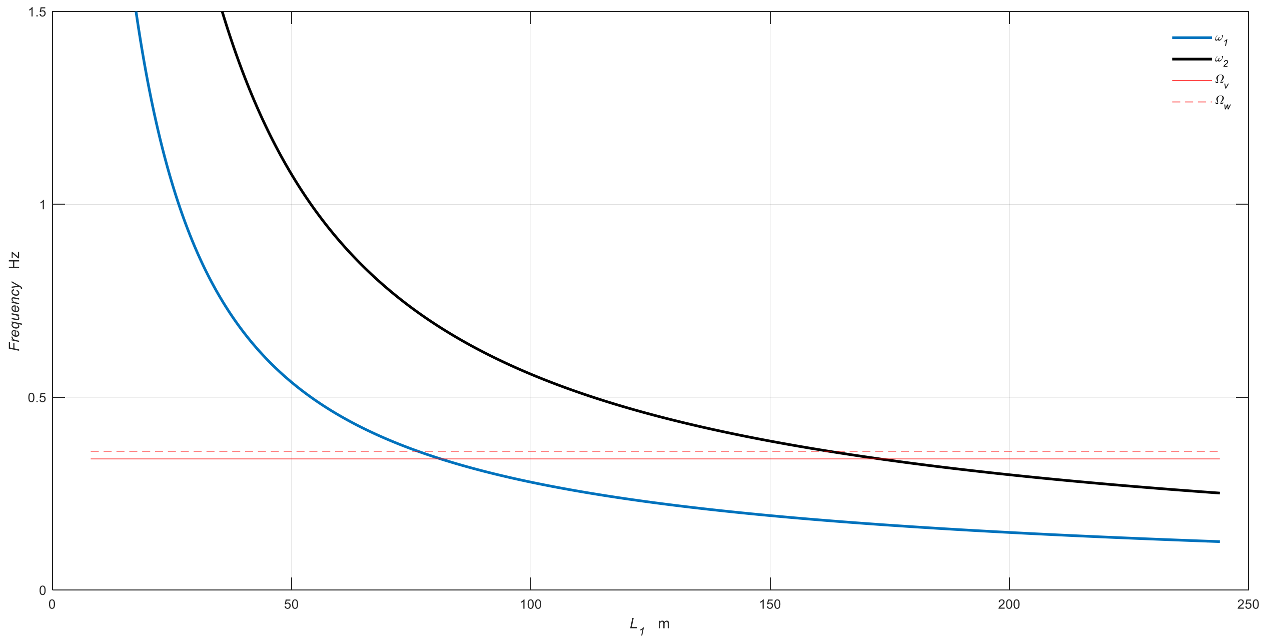

Figure 2 The natural frequencies: compensating ropes at the car side

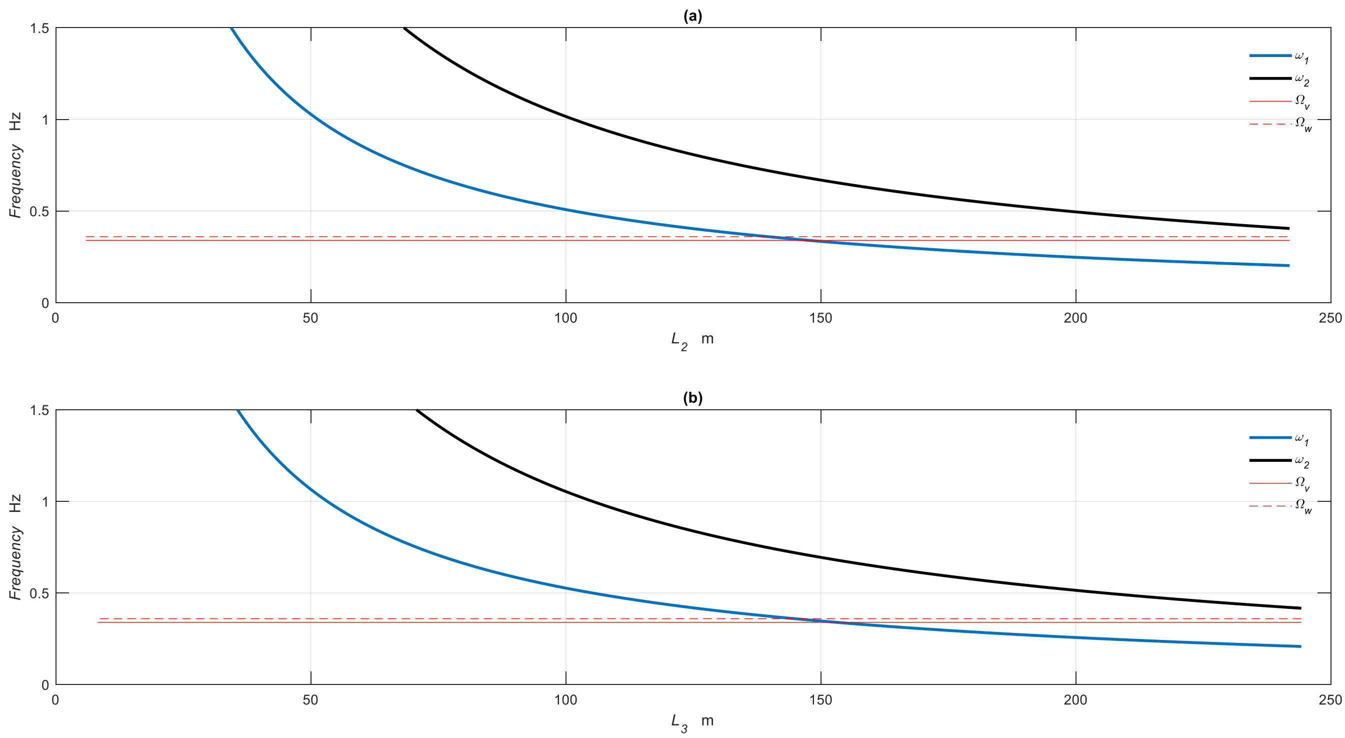

Figure 3 The natural frequencies: suspension ropes: (a) at the counterweight side and (b) at the car side

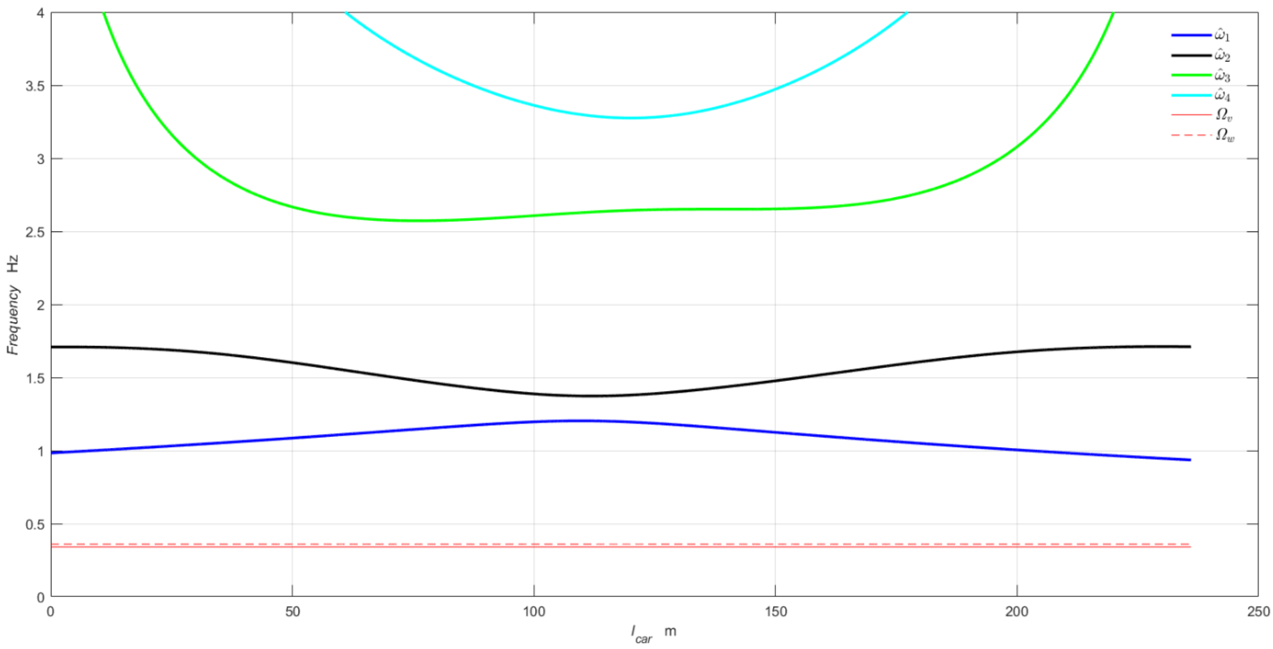

Figure 4 The natural frequencies: vertical modes

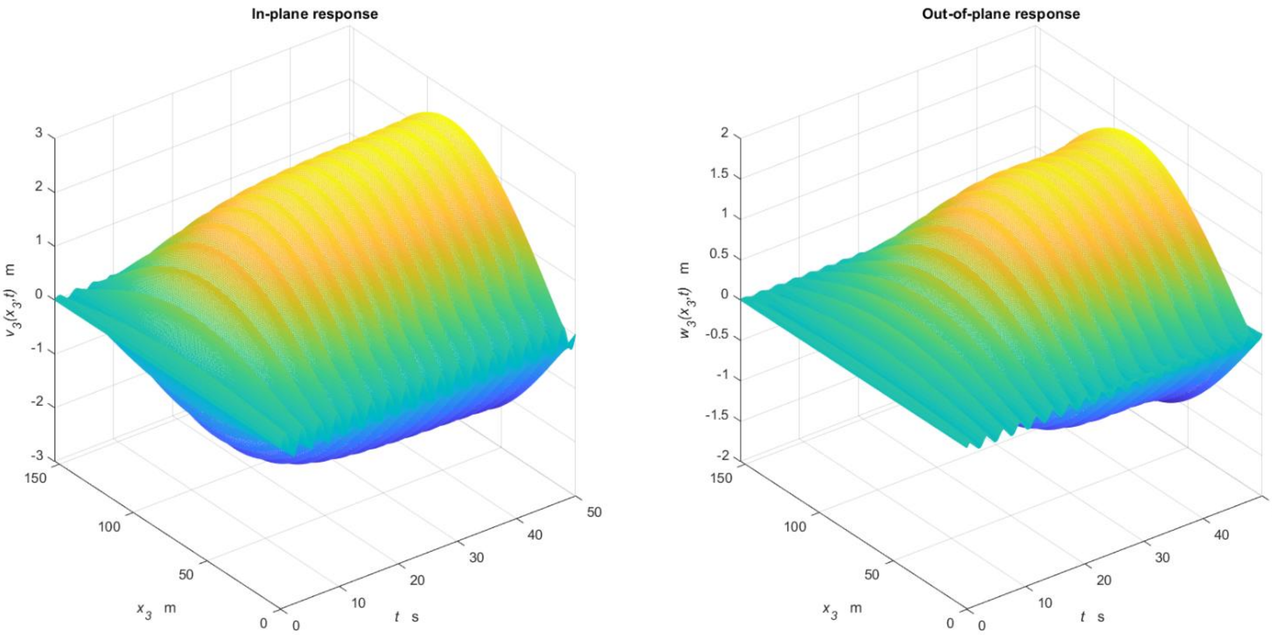

Figure 5 Displacements of the car suspension ropes

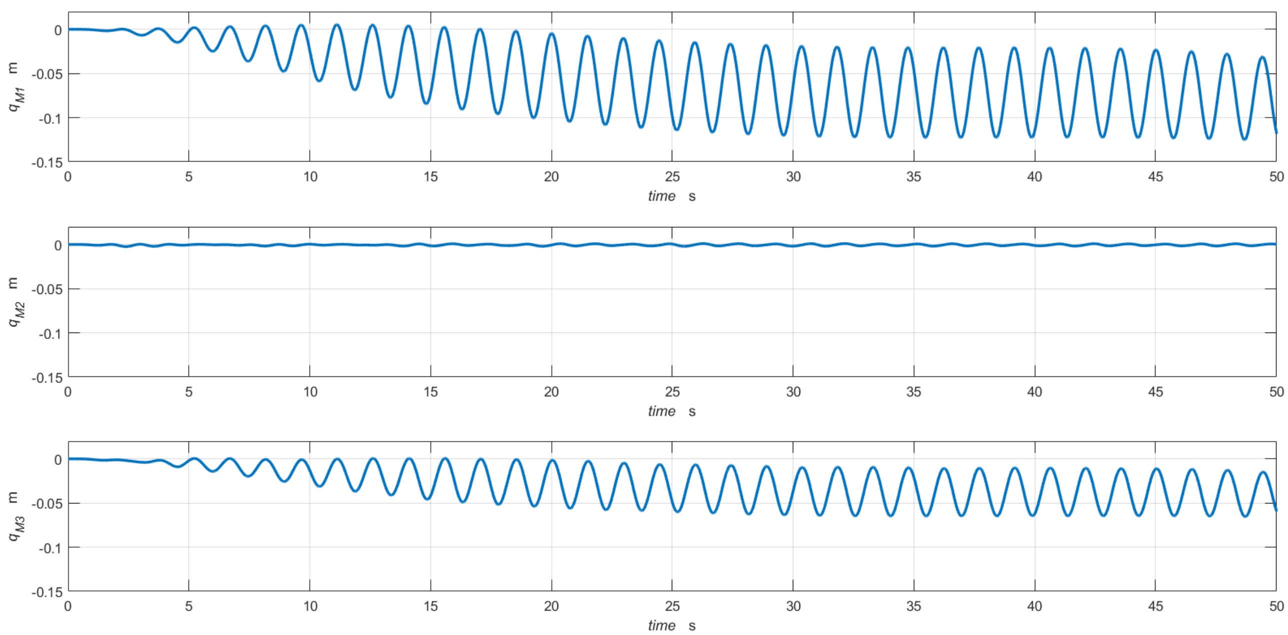

Figure 6 Vertical displacements of the car (q_{M1}), counterweight (q_{M2}) and compensating sheave (q_{M3})

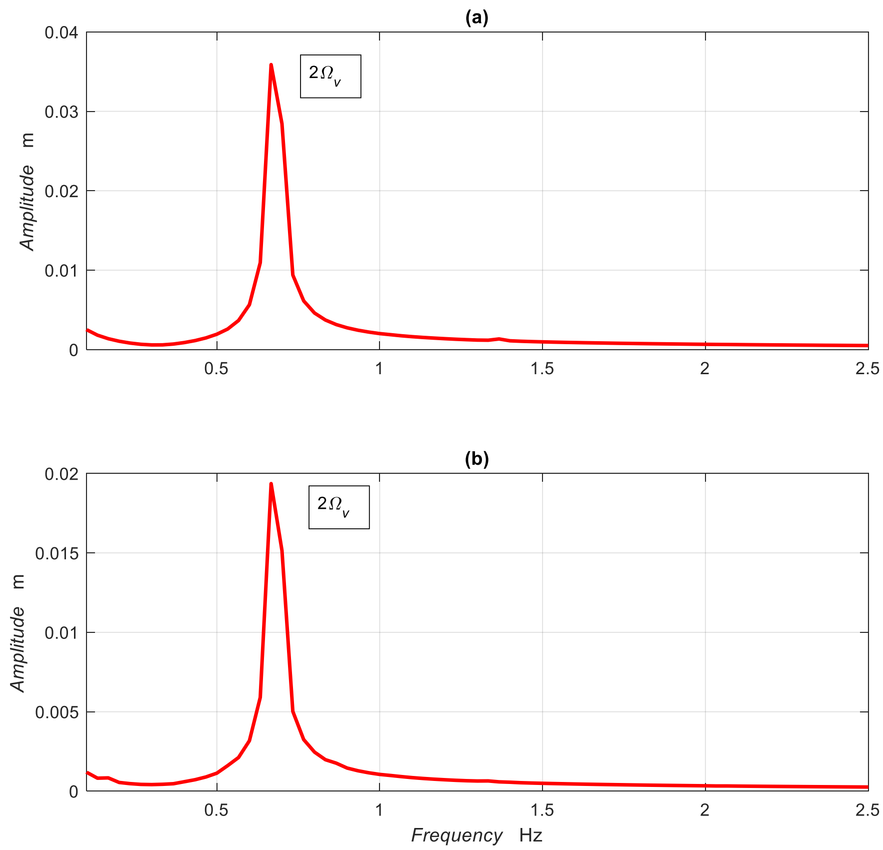

Figure 7 Frequency spectrum of vertical responses of the car (a) and compensating sheave (b)

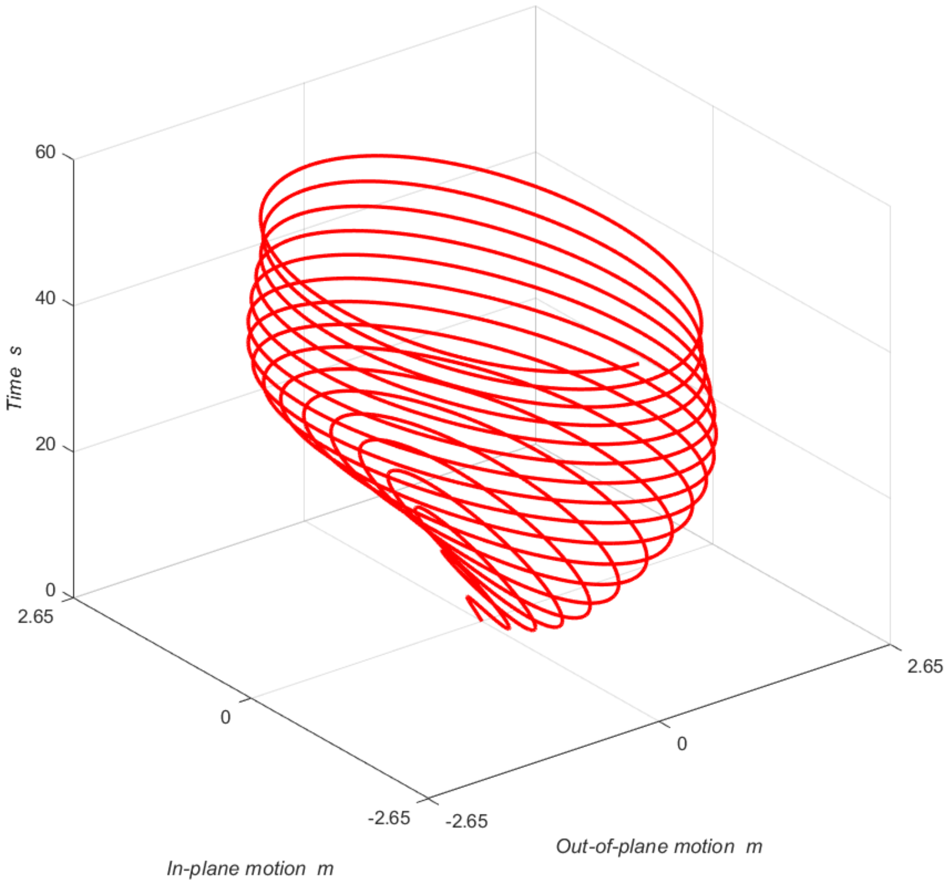

Figure 8 Whirling motions of the car suspension ropes (at x_3 = 78 m)

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BIOGRAPHICAL DETAILS

Stefan Kaczmarczyk has a master’s degree in Mechanical Engineering and he obtained his doctorate in Engineering Dynamics. He is Professor of Applied Mechanics and Postgraduate Programme Leader for Lift Engineering 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. Professor Kaczmarczyk has published over 100 journal and international conference papers in this field. He is a Chartered Engineer, elected Fellow of the Institution of Mechanical Engineers, and he has been serving on the Applied Mechanics Group Committee of the Institute of Physics.