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Vibration Problems in Lift and Escalator Systems: Analysis Techniques and Mitigation Strategies

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Abstract. The operation of lift and escalator installations is often affected by vibrations and vibro-acoustic noise. This leads to poor ride quality and a high level of dynamic stresses which may result in damage to the installation. Thus, a good understanding of vibration phenomena occurring in lifts and escalator systems is essential. Lift and escalator systems employ components rotating and translating at speed. Those include elastic tension members such as long ropes, cables, chains and belts. Due to their flexibility and loading conditions they are susceptible to vibration and their dynamic characteristics such as stiffness, mass and damping are time-varying. Thus, the analyst and the designer should be aware that the natural frequencies of a lift and escalator installation change with the time and speed of the transport motion. In lift systems the sources of excitation include the inertial load due to the system acceleration/deceleration profile; periodic excitation caused by the host building structure sway; excitation at the sheave from the drive machine; excitation at the car due to the car-guide rail interaction and aerodynamic effects. Vibration (and noise) in chain-driven escalator installations are often caused by the discrete nature of the chain links and their interactions with the sprocket. The dynamic loads produced by impact between the engaging roller and sprocket surface combined with polygonal action lead to excessive transverse vibrations of the chain. This in turn results in excessive friction wear which reduces the safe service life of the installation. The issues relevant to the vibration theory, modelling, testing and analysis of the dynamic response of lift and escalator systems are addressed in the paper. Then, passive, semi-active and active strategies to minimize the effects of adverse dynamic response of the system are discussed, so that the installation can operate under these conditions without alarm.

INTRODUCTION

The operation of lift and escalator installations is often affected by vibrations and associated vibro-acoustic noise. This leads to poor *ride quality* and a high level of dynamic stresses which may result in damage to the installation [1,2]. Thus, a good understanding of vibration phenomena occurring in lifts and escalator systems is essential in order to design a system which will satisfy ever demanding ride quality criteria. In this extended abstract the main issues concerning vibration problems arising in lift and escalator systems (vibrations in moving walks systems are also mentioned) are briefly discussed. Then, possible strategies to minimize the effects of adverse dynamic response of such systems are reviewed.

LIFT SYSTEMS

The underlying causes of vibration in an elevator system are varied, including poorly aligned guide rail joints, eccentric pulleys and sheaves, systematic resonance in the electronic control system, and gear and motor generated vibrations [1].

Vertical Car/ Counterweight Vibrations. In the vertical direction, the elevator car and counterweight are free to move and can oscillate on the 'spring' of the suspension ropes as shown in Fig. 1(a). The diagram presented in Fig. 1(b) illustrates a simplified vibration model of a lift car/ counterweight assembly. In this model *x(t)* is a vertical displacement of the car/ counterweight represented by an effective (equivalent) mass m_e suspended on a spring of an effective constant k_e. Damping in the system is represented by a *viscous damping* element of an effective damping constant c_e. Noting that there are n ropes, E is the modulus of elasticity and A denotes the effective cross-sectional area of each rope, with the roping configuration assumed as 1:1 we define the effective stiffness of the suspension system at the car side as

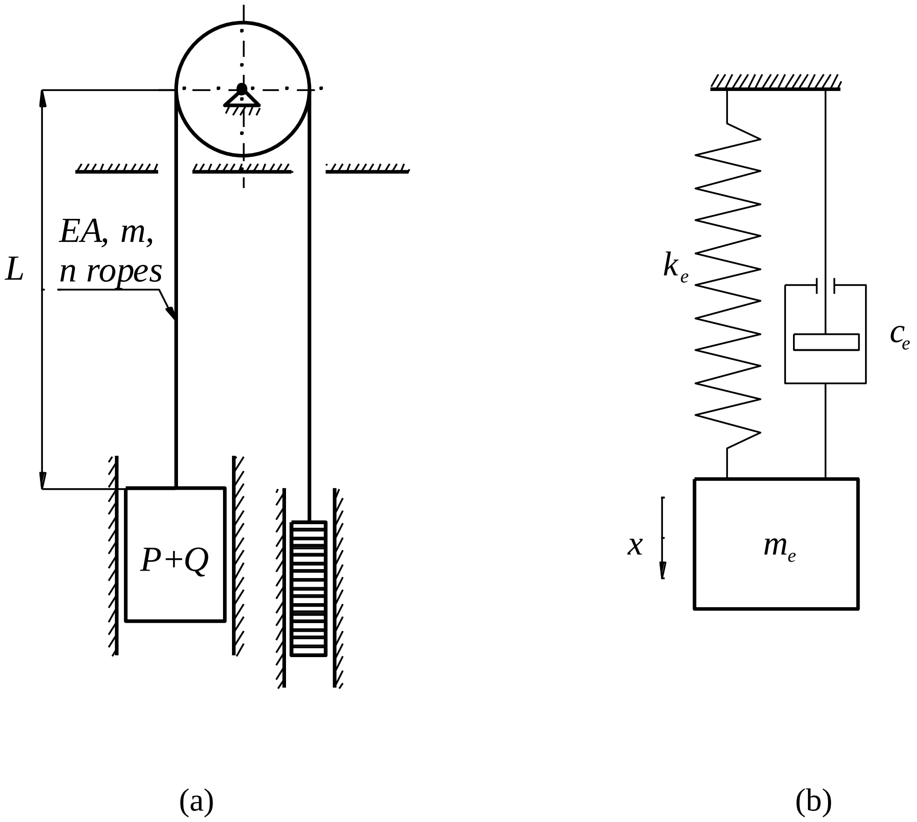

Figure 1.

$$k_e = n\frac{EA}{L} \tag{1}$$

where L is the length of the ropes at the car side. The equivalent mass of the car and suspension ropes at the car side when the lift is stationary is given by the following expression [3]

$$m_e = P + \alpha Q + \frac{1}{3}nmL \tag{2}$$

where P is the mass of an empty car, Q represents rated load and α is a 'loading factor' (when $\alpha = 1$ the car is carrying rated load). The quantity defined as

$$\omega_n = \sqrt{\frac{k_e}{m_e}} \tag{3}$$

is the natural frequency of the system. An important feature of the lift system is that the suspension ropes are of time varying length during the lift motion ($L = L(t)$). Furthermore, the number of passengers on board changes ($0 \le \alpha \le 1$). Consequently, the dynamic characteristics vary during travel, rendering the system non-stationary [4].

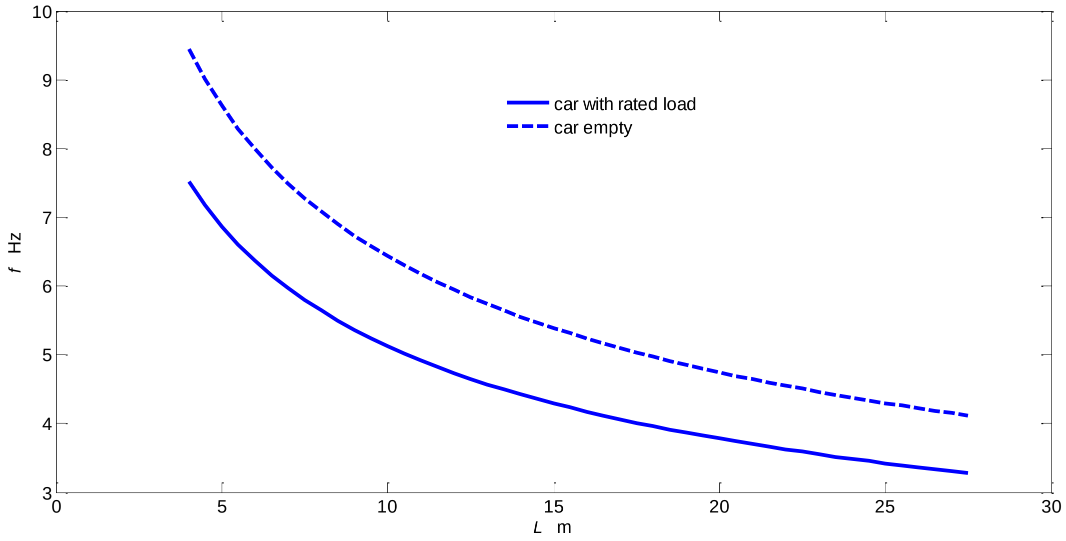

Figure 2. The variation of the natural frequencies with position of the car (determined in terms of *L*).

Fig. 2 shows the variation of the natural frequencies of a car of mass P = 15800 kg (for loaded and empty conditions; with Q = 9100 kg) with the position of the car in a hoistway for a lift installation of travel height H = 23.6 m. The car is suspended on n = 10 Drako 300T ropes of d = 16 mm and mass per meter m = 1.1 kg/m each. It is evident that the frequencies are increasing when the car is moved from its position at the bottom landing upwards and the length of the ropes L is getting shorter. The frequencies of the system with the car carrying no load are higher than the frequencies with the car with rated load. An adverse situation arises when one of the slowly varying rope frequencies approaches near the frequency of a periodic excitation existing in the system. This results in a *passage through resonance* [4]. In such case the lift car will not vibrate throughout its travel, but will pass through a resonant vibration at some particular stage in the travel. Very often, this vibration stage occurs at or near the highest floor, as the suspension ropes become short. Fig. 3 illustrates transient vibrations which might be experienced.

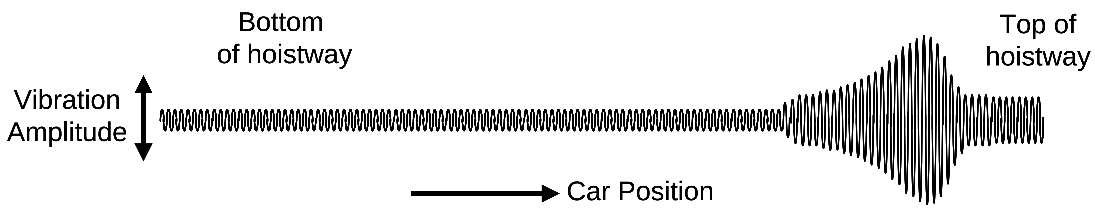

Figure 3. Transient vibration of a lift car.

Horizontal Car/ Counterweight Vibrations. In the horizontal direction a lift car is constrained by the guiding system. The guide rail irregularities introduce lateral excitation to the car during its travel. A simple model of a car of mass M moving at a constant speed V and guided by rails R_1 and R_2 is shown in Fig. 4(a). The guide shoes are represented by spring – viscous damper elements of coefficient of stiffness k and viscous damping coefficient c, respectively (in this model the influence of the hoist rope stiffness, damping and inertial characteristics is not accounted for).

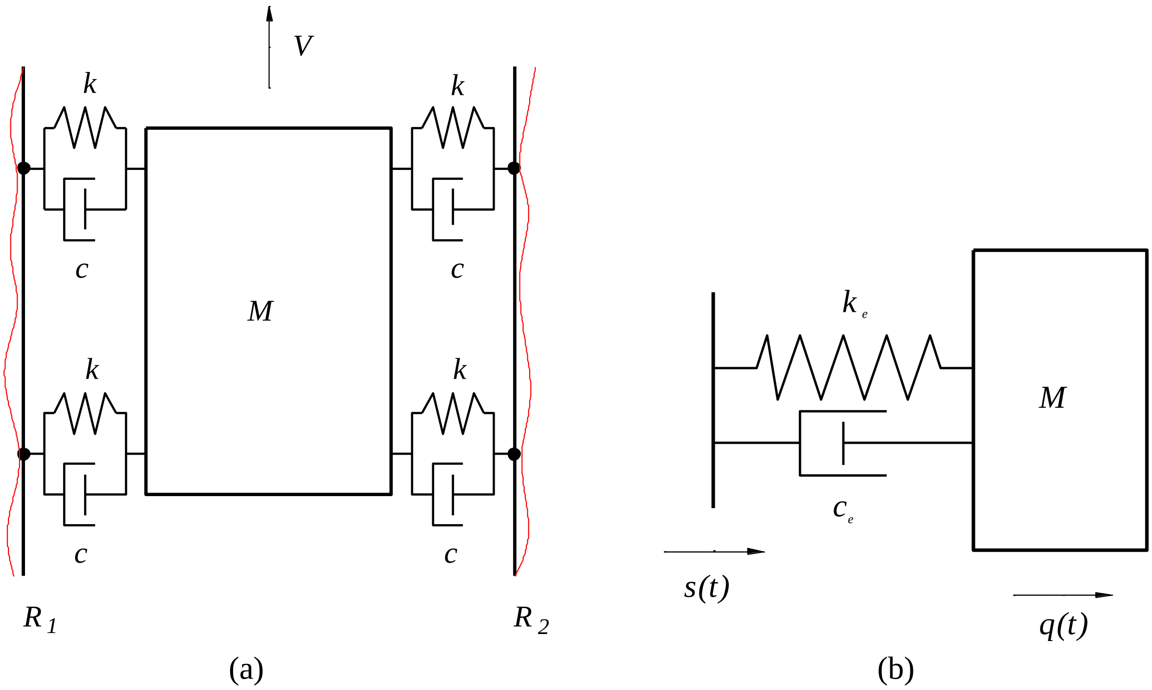

Figure 4. Simple model of a lift car guided by rails.

In vibration analysis this model can be simplified further and an equivalent single-degree-of-freedom (SDOF) system as shown in Fig. 4(b), where the lateral (horizontal) displacement of the car is denoted as *q(t)*. In this representation the combined stiffness and damping of the car – guide rail interface is given by the equivalent stiffness coefficient k_e and the equivalent damping coefficient c_e. The unevenness and/or bending of the guide rails results in a kinematic excitation represented by base motion *s(t)*. The excitation imparted by the rail joints can be approximated by a harmonic function $s(t) = s_m \cos \Omega t$ of the fundamental frequency $\Omega = \frac{2\pi}{\lambda} V$, where λ represents a wavelength equal to the distance between the joints. Subsequently, if $r = \frac{\Omega}{\omega}$, where $\omega = \sqrt{\frac{k_e}{M}}$ is the natural frequency of the system, the ratio of the maximum steady-state response amplitude q_m to the maximum input displacement s_m is given by

$$\frac{q_m}{s_m} = \sqrt{\frac{1+(2\varsigma r)^2}{\left(1-r^2\right)^2+(2\varsigma r)^2}} \tag{4}$$

where $\varsigma = \frac{c_e}{2M\omega}$ denotes the damping factor. The quantity defined by Eq. 4 is referred to as *displacement transmissibility*. This ratio is plotted in Fig. 5 for various values of ς. It is evident from this plot that if r is greater than $\sqrt{2} \approx 1.41$, the vibration amplitude of the car is smaller than the amplitude of rail displacement and isolation occurs. Near the resonance (*r = 1)* the transmissibility is determined by the amount of damping, namely by the value of ς and the larger the damping ratio, the better the resonance suppression. However, in the isolation region the smaller the value of ς, the better the isolation. For a damping ratio of *50%* ($\varsigma = 0.5$) the amplification at resonant frequency is in the range *1.5* to *2*. Simultaneously, the car – rail guide interface provides satisfactory isolation for the frequency range of $r > \sqrt{2}$. The analysis of the SDOF model of the car – rail interactions provides some fundamental understanding of the dynamic behaviour of the lift car. However, the suspension system should be included into the model in order to investigate the influence of the guide rail excitation on the overall performance of the lift system [4].

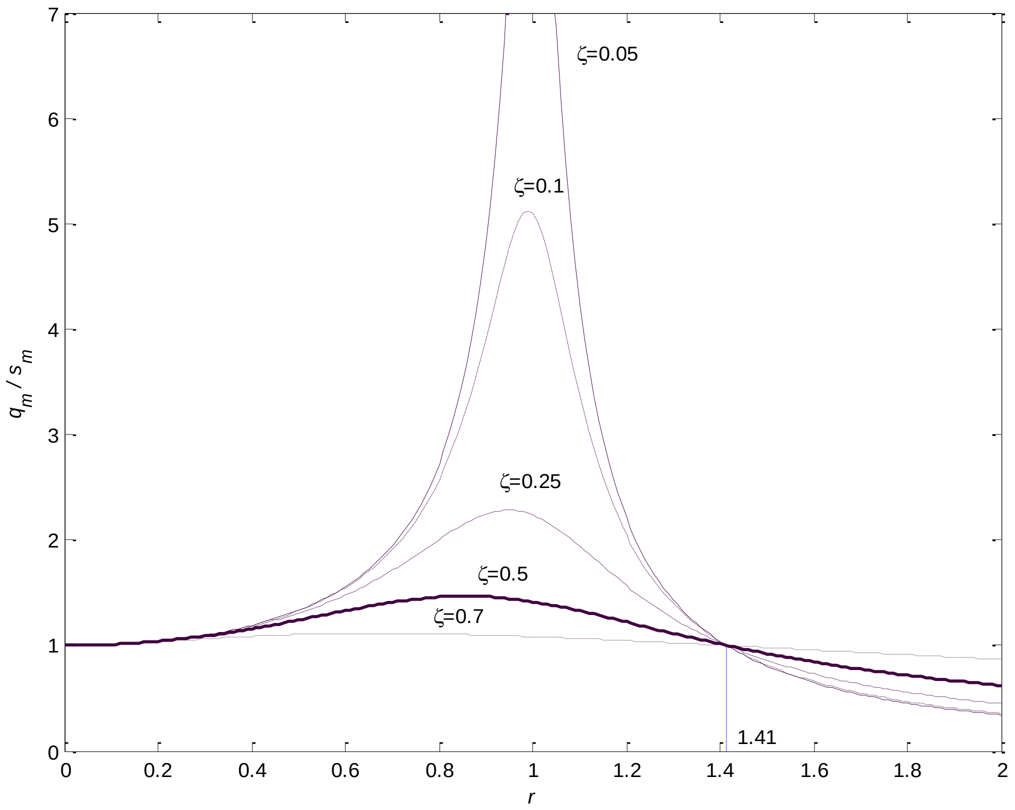

Figure 5. Displacement transmissibility.

ESCALATOR AND MOVING WALK SYSTEMS

Escalators and moving walks are generally similar in basic construction. They are chain-driven; with the chain being a major source of vibration. Vibrations are also induced by steps, rotating imbalance, misalignment, motor drive system dynamics and other typical causes.

Chain Dynamics. Chain dynamic behaviour is affected by the discrete nature of the chain links and sprocket teeth. Compared to traction driven elevators with the car and counterweight suspended on steel wire ropes (or on other means such as synthetic fibre ropes or coated steel belts) this discrete nature makes the chain drive unique with both advantages as disadvantages. The advantage is that the drive is a positive drive and no slip between the chain and the wheel (sprocket). However, the dynamic behaviour of chain drives is complex and they suffer from high level of noise and vibration [6]. Transverse and longitudinal vibrations of the chain are caused by the combined effect of so called polygonal action and impacts between the rollers and sprockets [7].

Impact Loads. The impact between the engaging roller and the sprocket is due to the velocity of the roller relative to the sprocket surface as the roller seats.

Polygonal Action. The polygonal action (effect) takes place due to the fact that the chain lying on the sprocket forms a polygon rather than a circle (see Fig. 6). This leads to motion of the chain with fluctuating speed, with the maximum and minimum speed determined as

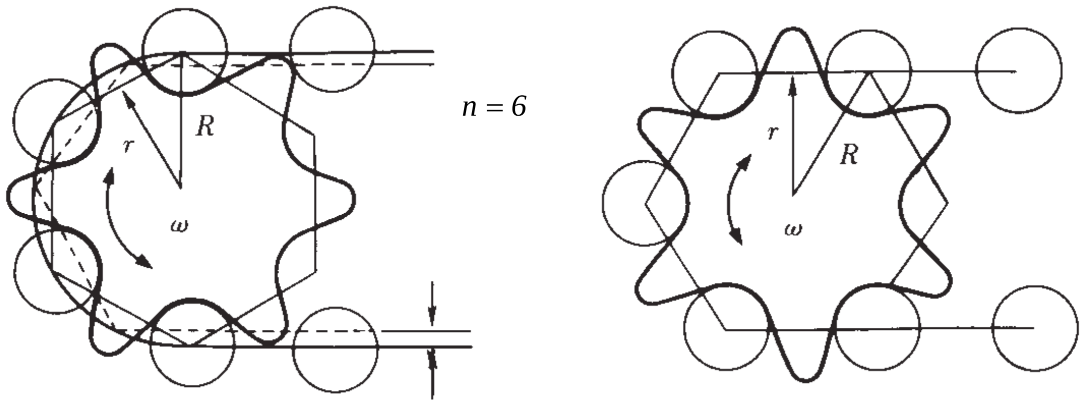

Figure 6. Polygonal effect.

$$\begin{aligned} v_{max} &= \omega R \\ v_{min} &= \omega r \end{aligned} \tag{5}$$

where ω is the rotational speed of the sprocket (in rad/s) and the relationship between the radius minimum r and the maximum radius R is given as

$$r = R\cos\left(\frac{\pi}{n}\right) \tag{6}$$

where n represents the number of teeth in the sprocket. The ratio of speed change (fluctuation) can be quantified by the following equation

$$\eta = \frac{v_{max} - v_{min}}{v_{max}} = 1 - \cos\left(\frac{\pi}{n}\right) \tag{7}$$

It is evident that the transport speed fluctuation is reduced if the number of teeth is increased (see Fig. 7). The chain vibrates according to the speed fluctuation and vibrations will be reduced if the number of teeth is increased. The polygonal effects lead to external and parametric excitations. The equation of transverse vibrations of the chain, in the case of negligible sag and moving at a constant low speed, can be formulated as follows [8]

$$m\left[w_{tt}(x,t) + 2vw_{xt}(x,t) + v^2 w_{xx}(x,t)\right] - \left\{\left[T_0 + P(t)\right]w_x(x,t)\right\}_x = f(x,t), \tag{8}$$

where m is mass per unit length of the chain, $w(x,t)$ is the chain displacement with x denoting the spatial coordinate measured along the span, T_0 represents static tension of the chain and P is a force combining the loads due to the polygonal effects and impacts, and $f(x,t)$ is an external periodic load originating from the polygon effects. It is evedent from Eq. 8 that both the polygonal action load and impact load represent parametric excitations.

The polygonal action load is periodic and depends on the angular speed of the sprocket. Thus, external and parametric resonances may arise in the escalator chain drive. A small excitation due to the polygonal action can produce a large tranverse response of the chain when the frequency of the external load becomes close to one of the natural frequencies of the chain. When the frequency of the parametric excitation is near twice one of the natural frequencies of the chain the *principal parametric resonance* results [9,10].

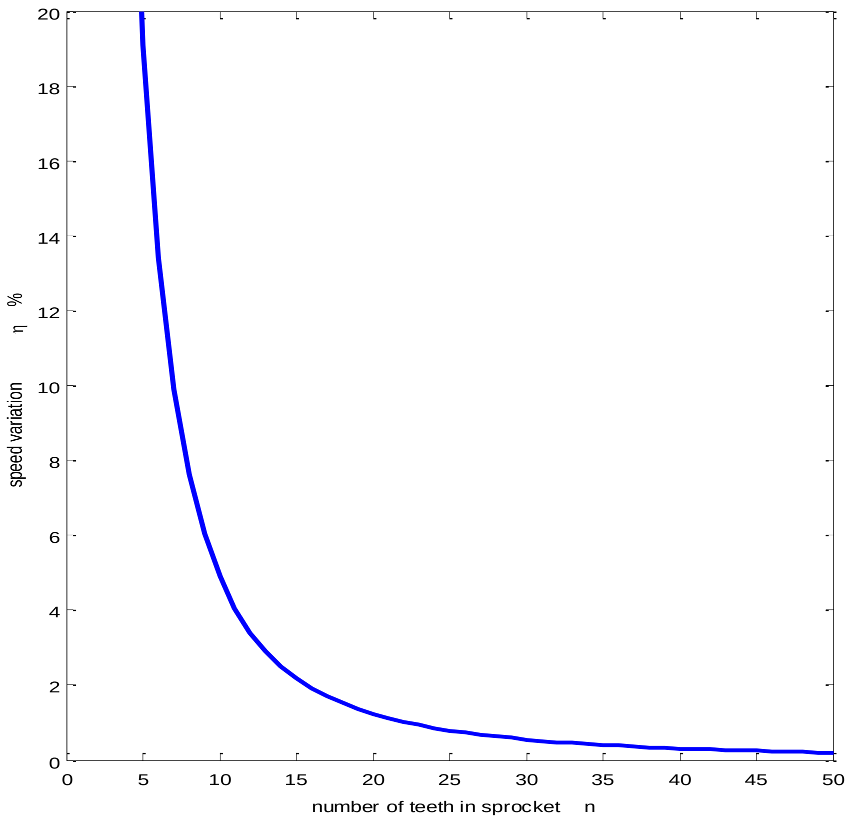

Figure 7. Ratio of speed fluctuation.

VIBRATION SUPPRESSION

Passive and active vibration control and suppression techniques can be employed in lift and escalator systems. The main passive methods used to suppress and mitigate the effects of excessive vibrations include the following:

- the control of the natural frequencies to avoid resonance under external excitations;
- the use of viscoelastic materials (viscoelastic damping treatment) to dissipate vibrational energy and to prevent excessive response of the system;
- the use of vibration isolators to reduce the transmission of excitation from one part of the machine to another;
- the application of an auxiliary mass neutralizer or vibration absorber to reduce the response of system.

In lift systems vibration isolation is often applied.The lift car is mounted within the sling structure on elastomeric isolation pads to reduce vibration transmission to passengers. In the machine system mounted on steel beams/frame and supported by a concrete floor vibration isolation pads are inserted between the machine/frame and frame/floor to reduce the transmission of excitation forces and vibration to the suspension/car system. The car roller guides are equipped with spring-damper elements to suppress vibration due to the rail excitation sources.

Various passive methods of suppressing chain vibrations in escalator systems have been proposed. The fundamental approach is to reduce the speed fluctuations by the application of chain guide rails of various designs [11,12].

Alternatively, active vibration control (AVC) techniques, that involve actuators to generate forces and to apply them to the structure/machine in order to reduce its dynamic response, can be employed. The fundamental principle of AVC is illustrated in Fig. 8. The vibration (response) *x* of the machine of mass *m* is measured using a motion sensor. The response is then used to determine the force to apply to the machine via the actuator (hydraulic or piezoelectric device or an electric motor). The mathematical algorithm to calculate the force is called the *control law*. The system comprising the sensor, actuator and the electronic circuit to read the sensor's output and determine and supply the required signal to the actuator is referred to as *feedback control* system.

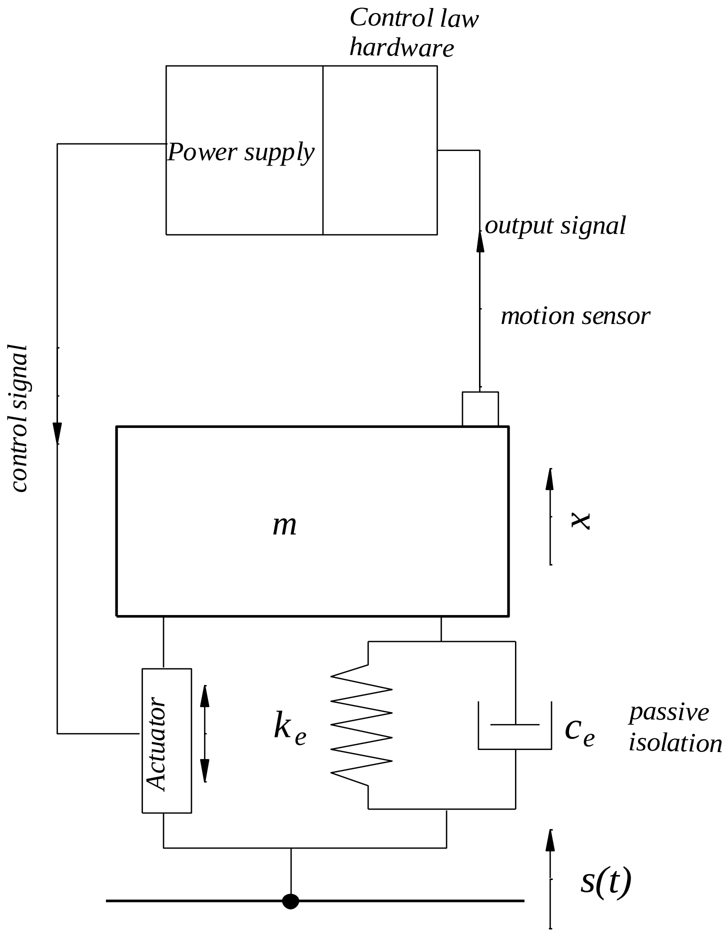

Figure 8. Active vibration control.

In lift systems this method has been used to develop *active roller guides* (ARG) to reduce the horizontal vibrations of lift car caused be guide rail excitation in high-speed, high-rise applications. Also, *active vibration dampers* can be applied under the car, between its floor and sling, to suppress vertical vibrations [13].

CONCLUSIONS

Vibration phenomena in lift and escalator systems lead to poor ride quality. Eccentric pulleys and sheaves, systematic resonance in the electronic control system, and gear and motor generated vibrations are typical causes of vertical vibrations of a lift car. Uneven, bent rails, incorrect installation and rough surface cause horizontal vibration of the car and of its suspension members. Escalators and moving walks are chain-driven and the chain is a major source of vibration. Passive and active vibration suppression techniques can be used to control vibration phenomena. The latest techniques in the AVC technology can be adopted to mitigate the effects of vibrations and deployed to control adverse dynamic behaviour of lift and escalator systems.

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