The Assessment of Vibration Absorption Capacity of Elevator’s Passengers

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Abstract. The transfer function expressions for damped two degrees of freedom systems with damped vibration absorbers were used to model and simulate the dynamic response of an elevator’s car. The numerical simulation is implemented in MATLAB and experimental tests are conducted. The elevator’s car under various load conditions with passengers wearing different type of shoes was excited over a range of frequencies and amplitudes. The agreement of the results from experimental tests with the model predictions is analysed. The simulation model is validated and the passenger’s role to act as a dynamic absorber is assessed.

1. Introduction

The analysis of the dynamic behaviour of the elevator car system plays an important role in elevator engineering. Nowadays, passengers demand high ride quality of elevators. In particular low vibration and noise levels must be ensured in the modern design of elevators. International standards encourage the development of uniform, reliable and precise measurement and processing techniques to be applied within the elevator industry [1].

Possibly, the main difficulty in developing a tool, such as software for elevator dynamic analysis, is to model the dynamic characteristics and behaviour of passengers who are the most complex and important elevator mechanical “component”, in sufficient detail. The main goal of this paper is to propose a simplified model to predict the dynamics of an elevator car with passengers load.

An elevator is a complex mechanical system [2]. Great errors results occur when trying to isolate the problem to the one directly involved component without prior justification and modeling. So we must refer to the Elevator System ‘ES’ for mechanical behavior of the elevator purposes [3]. On the opposite extreme, detailed and rigorous description on the dynamics of the human body parts requires complex and sophisticated models like those applied to dummies for crash tests on elevators [4] or on vehicle cars. This may be too much difficult in our first attempt in understanding the ES, that is, the elevator car mechanics.

The ISO standard for lift ride quality measurements [1] states the aspects that influence the dynamic measurements results for comfort. First, coupling of the instrumentation to the car floor must be similar to that for a person by a minimum feet pressure of 60kPa. Secondly, on personnel traveled within the instrumentation, is stated that not more than 2 persons should be present. Additionally, the instrumentation shall be place on “any car floor coverings which are normally present”. Consequently, the ISO states the components that can affect the acceleration measurements in the car: the instrument, the persons and the floor coverings.
Experimental measurements on real elevators with standard equipment for lift ride quality and subsidiary analysis based on more than 15 geared elevators under several running conditions (travel direction, speed, number of persons, starting floor, …) have revealed that the vertical car vibrations are mainly based on a single frequency that slips slightly with the traveled direction, independently of the number of passenger’s traveled.

First we tried with the simplest undamped one degree of freedom expression, considering the suspension ropes and terminal springs as the elastic constant, and the suspended mass (car plus frame plus passenger traveled) as the equivalent mass; but is not capable to explain the constancy of the vibration frequency when varying the number of passengers. So, it is clear at this point, that we need a damped model as simple as possible and some laboratory experiments to understand the results on real elevators.

2. Simplified mechanical model

In a typical elevator system the car frame is suspended by sling and the car is mounted within the car frame on isolation blocks (see Figure 1). By considering vertical displacements only the elevator system comprising the car and the passengers/weights is represented by a two degrees of freedom model with \( y_{PA} \) and \( y_{CA} \) defining the passengers and car displacements, respectively (see Figure 2).

![Figure 1. The car-passengers subsystem.](image1)

![Figure 2. Discrete model.](image2)

The model consists of mass \( m_{PA} \) representing the passengers and of mass \( m_{CA} \) representing the car. The passenger mass is supported at the car floor via a spring-damper element of stiffness coefficient \( k_{PA} \) and viscous damping coefficient \( c_{PA} \), respectively. The car is isolated from the frame by isolation blocks represented by a spring-damper element of stiffness coefficient \( k_{CA} \) and viscous damping coefficient \( c_{CA} \), respectively.

The stiffness coefficient \( k_{PA} \) and viscous damping coefficient \( c_{PA} \) are defined as

\[
\frac{1}{k_{PA}} = \frac{1}{k_{pa}} + \frac{1}{k_{sh}} + \frac{1}{k_{fl}} \quad \frac{1}{c_{PA}} = \frac{1}{c_{pa}} + \frac{1}{c_{sh}} + \frac{1}{c_{fl}}
\]

(1)

where \( k_{pa} \) denotes the average stiffness of the human body in the vertical direction, \( k_{sh} \) denotes the shoes vertical stiffness, \( k_{fl} \) denotes the vertical stiffness of the covering floor of the car, \( c_{pa} \) denotes the average viscous damping coefficient of the human body in the vertical direction, \( c_{sh} \) denotes the shoes...
viscous damping coefficient in the vertical direction and \(c_b\) denotes the covering floor viscous damping coefficient of the car in the vertical direction.

The elevator car is attached to the car frame by a number of isolation blocks and their combined stiffness and damping coefficients are determined as

\[
k_{PA} = nk_{sb}; c_{PA} = nc_{sb}\tag{2}
\]

where \(k_{sb}\) and \(c_{sb}\) are the spring/damper constants for a single isolation block, respectively and \(n\) the number of isolation blocks.

As shown in Figure 2, the car frame is subjected to a harmonic base motion excitation \(y_{FR}\). A frequency sweep is applied to investigate the forced response of the system and to identify the resonance frequencies.

Then, the differential equations of motion of the forced-movable base system in matrix form are given as:

\[
\begin{bmatrix}
m_{CA} & 0 \\
0 & m_{PA}
\end{bmatrix}
\begin{bmatrix}
\ddot{y}_{CA} \\
\ddot{y}_{PA}
\end{bmatrix}
+
\begin{bmatrix}
c_{CA} + c_{PA} & -c_{PA} \\
-c_{PA} & c_{PA}
\end{bmatrix}
\begin{bmatrix}
\dot{y}_{CA} \\
\dot{y}_{PA}
\end{bmatrix}
+
\begin{bmatrix}
k_{CA} + k_{PA} & -k_{PA} \\
-k_{PA} & k_{PA}
\end{bmatrix}
\begin{bmatrix}
y_{CA} \\
y_{PA}
\end{bmatrix}
=
\begin{bmatrix}
c_{CA}y_{FR} + k_{CA}y_{FR} \\
0
\end{bmatrix}
\tag{3}
\]

Now we introduce the dimensionless constants

\[
\omega = \frac{\omega_{PA}}{\omega_{bCA}} = \frac{1}{\sqrt{m_t}} \sqrt{\frac{k_{PA}}{k_{CA}}}; \omega = \frac{\omega}{\omega_{bCA}}; \omega^2 = \frac{k_{CA}}{m_{CA}}; \omega^2 = \frac{k_{PA}}{m_{PA}}
\]

and the additional constants for the dimensionless time, and damping factors, respectively,

\[
\tau = \omega_{bCA} t; 2\zeta_{CA} = \frac{c_{CA}}{m_{CA} \omega_{bCA}}; 2\zeta_{PA} = \frac{c_{PA}}{m_{PA} \omega_{bPA}}
\]

Then the expression by (3) transforms into:

\[
\frac{d^2y_{CA}}{d\tau^2} + \left(2\zeta_{PA} + 2\zeta_{PA} m_t \omega_t\right) \frac{dy_{CA}}{d\tau} + \left(1 + m_t \omega_t\right) y_{CA} - 2\zeta_{PA} m_t \omega_t \frac{dy_{PA}}{d\tau} = m_t \omega_t \frac{dy_{CA}}{d\tau} - m_t \omega_t \frac{dy_{PA}}{d\tau} - 2\zeta_{CA} \frac{dy_{FR}}{d\tau} - y_{FR} = 0
\]

\[
\frac{d^2y_{PA}}{d\tau^2} + 2\zeta_{PA} \omega_t \frac{dy_{PA}}{d\tau} + \omega^2_{PA} y_{PA} - 2\zeta_{PA} \omega_t \frac{dy_{CA}}{d\tau} - \omega^2_{CA} = 0
\]

After computing the Laplace transform of equations (4) and solve for \(Y_{CA}(s)\) and \(Y_{PA}(s)\), which are the transformed functions of \(y_{CA}(\tau)\) and \(y_{PA}(\tau)\) respectively, we get after following similar steps as those from Balachandran[8]:

\[
Y_{CA}(s) = \frac{K_{FR}(s) E_{PA}(s)}{D_{PA}(s)}
\]

\[
Y_{PA}(s) = \frac{K_{FR}(s) C(s)}{D_{PA}(s)}
\]

where
where $Y_{FR}(s)$ is the Laplace transform of $y_{FR}(\tau)$, the frame displacement. In our case, for the sinusoidal frequency sweep, we state:

$$y_{FR}(\tau) = y_0 \sin(\Omega_0 \tau) \quad (6)$$

where $\Omega_0 = \omega_r / \omega_{CA}$, $\omega_r$ is the angular frequency of the shaker, and $y_0$ is the displacement amplitude of the shaker.

The time domain solution of the system can be obtained through the inverse Laplace transforms of $Y_{CA}(s)$ and $Y_{PA}(s)$ by using MATLAB.

3. Test Setup

The laboratory car-frame was excited by a shaker actuator with closed loop displacement control and varying amplitude, phase and frequency.

The test setup is shown in Figure 3. A closed loop controlled actuator was fixed to a rigid frame and to the car by a rigid base. The car consists of a floor and a ceiling joined by four rigid angular bars. A special fixture has been developed to distribute the load due to weights on four pairs of shoes standing on the car floor. Two arrangements were done: one for traveling persons and the other for rigid masses. In the arrangement for rigid masses the shoes are fitted with wood-foot fixtures which are supporting a rigid base where the rigid masses or weights are held. In this way, the shoes are subjected to similar boundary conditions as the human feet.

Every test must specify the span or displacement amplitude, the frequency bounds of the sweep, the car and passenger/weight masses, the type of shoes and covering floor and the spring/damper constants of the isolation blocks. The basic tests outputs consist of a plot of the shaker/frame displacement transmissibility in $\mu m/\mu m$ versus frequency in Hz, and other plot of the car to shaker/frame displacement phase versus frequency in Hz.
For transmissibility measurements a CSI 2130 portable analyzer with two IMI 603C1 piezoelectric accelerometers was fix to the actuator and to the rigid base of the car.

4. Results, Theoretical Comparison and Discussion

A MATLAB routine was developed to determine the theoretical car displacement transmissibility, which is defined as the ratio of the amplitude of the car response to that of the frame/excitation motion:

\[ |G(\Omega)| = \frac{|Y_{CA}(\Omega j)|}{|Y_{FR}(\Omega j)|} \]  \hspace{1cm} (7)

and derived from (5) taking \( y_{FR}(0)=0 \) and substituting \( s \) by \( \Omega j \):

\[ |G(\Omega)| = \frac{(1 + 2j\zeta_{CA}\Omega)(\omega_j^2 - \Omega^2 + 2j\zeta_{PA}\omega_j\Omega)}{\Omega^4 - (\omega_j^2 + 2j\zeta_{PA}\omega_j\Omega)(1 + m_r) + (1 + 2j\zeta_{CA}\Omega)(\omega_j^2 + 2j\zeta_{PA}\omega_j\Omega)} \]  \hspace{1cm} (8)

If the passengers/weights mass is very small compared to the car mass then \( m_r \to 0; \omega_r \to \infty \), the expression (8) becomes:

\[ G'(\Omega) = \frac{1 + 2j\zeta_{CA}\Omega}{\Omega^2 + 1 + 2j\zeta_{CA}\Omega} \]  \hspace{1cm} (9)

which is the amplification factor for a single degree of freedom system with the car itself.

The frequencies where the magnitude of \( G(\Omega) \) will peak is of much interest in lift ride quality measurements. These correspond to the resonant frequencies that, generally, are not the same as the undamped or damped natural frequencies [7].

The transmissibility given by expression (8) is analyzed using a typical 312 kg car with 6 persons capacity mounted on 4 rubber isolation blocks (\( m_{CA}=312kg; k_{sb}=0.267MN/m; \zeta_{sb}=0.032; k_{CA}=4k_{sb} =1.07MN/m; \zeta_{CA}=0.032=5(4c_{sb}/(k_{CA}m_{CA}))^{1/2} \)) in the frequency range of interest between 0 and 25Hz.

For one passenger of mass \( m_{1p}=82kg \) and stiffness \( k_{1p}=76kN/m \) [6]: \( m_{PA}=m_{1p}, k_{PA}=k_{1p} \) and the dimensionless constants is found to be \( m_r=82/312=0.26 \) and \( \omega_r=0.52 \). The corresponding transmissibility plots are shown in Figure 4 for different \( \zeta_{PA} \) ranging from 0.001 to 0.6.

![Figure 4: Transmissibility for 1 passenger](image1)

![Figure 5: Transmissibility for 2 peaks.](image2)
Only if \( \zeta_{PA} \) is less than 0.2 the first expected peak for two degrees of freedom system appears (see the zone for \( \omega/2\pi \approx 5\text{Hz} \)). Similar intensity of the two frequency peaks is only possible if \( \zeta_{PA} \) is very small (\( 0.02 \leq \zeta_{PA} \leq 0.02 \)) or if the dimensionless frequency ratio \( \omega_r \) be larger than 0.52 (see Figure 5 for \( \omega_r = 0.72, 1.4 \) times the actual value of 0.52, using more robust isolation blocks: \( k_{IB} = 0.7\text{MN/m}; \) \( \xi_{IB} = 0.075; \) \( k_{CA} = 2.8\text{MN/m}; \) \( \xi_{CA} = 0.075 \) and 1 passenger of 85kg mass). Thus, it can be concluded that the first resonance is damped out for the high damping ratio. The two masses tend to become locked together and appear to behave like a single mass. Consequently, only one resonant frequency is produced and its value coincides with that of the original primary system that the car is.

The effect of increasing the number of the passengers \( n \), can be demonstrated with the assumption that all of them move in phase with each other and at the same amplitude for equal mass \( m_{1p} \), stiffness \( k_{1p} \) and damping factor \( \zeta_{1p} \).

\[
\frac{1}{k_{PA}} = \frac{1}{nk_{1p}} \quad \Rightarrow \quad \zeta_{PA} = \frac{c_{PA}}{2\sqrt{m_{PA}k_{PA}}} = \frac{nc_{1p}}{2\sqrt{n\cdot m_{1p} \cdot k_{1p}}} = \zeta_{1p}
\]

(10)

The above expression (10) means that the damping ratio \( \zeta_{PA} \) is independent of the number of ideal passengers. The plots presented in Figure 6 shows the effect of increasing number of passengers together with the experimental results and the type of shoes worn. It is found that maximum amplitudes in the empty car resonance frequency are increasingly attenuated with the number of passengers and that sport shoes are preferred to moccasins type. For all \( \Omega \), \( \zeta_{PA} \) and number of ideal passengers:

\[
\left| G^0(\Omega) \right| \geq \left| G(\Omega, \zeta_{PA}) \right|
\]

(11)

that is: the new curves, basically, are “covered” by that for the empty car \( G^0 \) and the amplification factor is lowered for all \( r \) by increasing passengers.

Experimental results in Figure 6 are denoted by ‘x’ dot in the same colour as the corresponding theoretical curve. Good agreement is found in the predicted resonance frequencies and amplitudes for the empty car and 1 passenger cases.

**Figure 6.** Influence of the passengers and type of shoes and experimental data.

**Figure 7.** Varying the car mass with a single passenger.
The results for different car masses, $m_{CA}$, with a single passenger are shown in Figure 7. It can be observed the coincidence of the peaks frequency with those of the empty car (denoted by vertical lines), and a small decrease of the peak amplitudes with the decreasing car mass. This confirms the experimental data on real lifts [5] that the attenuation of the car vibration acceleration rate when traveling passengers is effective with different car masses and even more efficient with increasing ratio of passengers to car mass $m_r$.

The effect of rigid masses instead of persons is predicted by (8) if $\omega_r \to \infty; \zeta_{PA} \to 0$:

$$|G'(\Omega)| = \frac{1 + 2j\zeta_{CA} \Omega}{(1 + m_t)\Omega^2 + 1 + 2j\zeta_{CA} \Omega}$$

which is the amplification factor for the car with its mass increased accordingly. In this case, the special fixture described in the test setup was used with a total weight mass of 200kg ($m_{PA} = 200$kg) standing on four pairs of moccasins type shoes ($k_{PA} = 2.5$MN/m; $\omega_r = 3.79; \zeta_{PA} \to 0$). Applying (12) we get the resonance frequency of the system $f$, given as:

$$f(m_{PA} = 200kg; \zeta_{PA} = 0) = \frac{1}{2\pi} \sqrt{\frac{nk_{PA}}{m_{CA} + m_{PA}}} \approx 11.7Hz$$

Clearly different from that for the car itself:

$$f(m_{PA} = 0) = \frac{1}{2\pi} \sqrt{\frac{nk_{PA}}{m_{CA}}} = 15Hz$$

Using the MATLAB routine for equation (8) and the experimental configuration ($m_t = 200/312; \omega_r = 1.94$) in a wide range of $\zeta_{PA}$, was obtained a resonance frequency of 11.56Hz. The effect of traveling weights ‘over’ shoes instead of passengers is illustrated in Figure 8. Both, the resonance peaks are slightly amplified and the resonance frequency is lowered with increasing weight mass.
Finally, the effect of the increasing car damping is illustrated in Figure 9 for two idealized passengers wearing moccasins. The expected benefit for better attenuation is shown.

5. Conclusions

The amplification factor for a damped two degrees of freedom model of an elevator car was analyzed to predict the effect of passenger vibrational behaviour and characteristics on the dynamic response of an elevator’s car. It is evident that rigid weights, even when supported on moccasins or sports shoes, don’t diminish the resonance response of the car. The effect of lowering the resonance frequency by coupling the weight mass to the car mass is predicted. However, in the case of the car with the passengers load it is observed that the resonance response is reduced. This effect becomes more prominent when sports shoes are used instead of moccasins-type shoes as predicted by the amplification factors.

It is concluded that the dynamic response of the elevator car can be substantially influenced by the passenger’s characteristics and behaviour during the elevator travel. Thus, more consideration should be given in BS ISO 18738 standard to the requirements and methodology for the measurement of elevator ride quality. In particular, a more in-depth analysis is needed to specify the type of shoes the person can wear when measuring elevator ride quality during elevator travel. In addition, the issues of the number of persons present in the car and the type of the car floor covering should be assessed.

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