11th International Conference on Vibration Problems
Z. Dimitrovová *et al.* (eds.)
Lisbon, Portugal, 9-12 September 2013

INFLUENCE OF THE NUMBER OF PASSENGERS ON THE DYNAMIC RESPONSE OF AN ELEVATOR CAR SYSTEM

I. Herrera*[1], H. Su[2], S. Kaczmarczyk[2]

[1]Escuela de Ingenierías Industriales, Universidad de Extremadura, Avda. de Elvas s/n, 06006 Badajoz, Spain
iherrera@unex.es

[2]School of Science and Technology, The University of Northampton, St. George's Avenue, Northampton, NN2 6JD, UK
{huijuan.su,stefan.kaczmarczyk}@nortahmpton.ac.uk

Keywords: Elevator car system, Stiffness, Damping, Vibration, Modal Analysis.

Abstract. *In prediction of the dynamic behaviour of an elevator car system, it is important to take into account the influence of passengers' behaviour in the car. In this paper a simulation model is extended to analyse the influence of various loading car conditions on the dynamic response of the elevator system. This involves the investigation into the dynamic response of the car with different loads. An experimental rig with a rectangular elevator platform fixed on the top of four silent blocks attached to a shaker is designed to conduct experimental tests. The car is excited over a range of frequencies and amplitudes. A number of passengers wearing different type of shoes in the car are investigated. The transmissibility measurements are carried out with a harmonic excitation applied first to an empty car and then to the car with a number of passengers. The excellent agreement from experimental tests with the model predictions is achieved. The passenger's role to act as a dynamic absorber is assessed and recommendations to achieve the best ride quality under load conditions are provided and summarised.*

1 INTRODUCTION

The analysis of the dynamic behaviour of the elevator car system plays an important role in elevator engineering and superior ride quality of elevators is demanded nowadays. In particular vibration in the low frequency range must be ensured in the modern design of elevator systems. International standards encourage the development of uniform, reliable and precise measurement and processing techniques to be applied within the elevator industry [1].

It is a challenging task to develop a precise and reliable tool/software in order to model the dynamic characteristics of the elevator system. As it is well known that behaviour of passengers is the most complicated factor and in turn the most important elevator mechanical "components", it obviously becomes even difficult to include these components to the model. Recently, elevator model and measurement procedures have been researched [2, 3] to determine the damping and stiffness characteristics of passengers.

Several works have studied the vertical vibration of the human body in different postures: seated [4-6], supine [7] and standing [8-13]. In vertical transport we are mainly interested in the standing posture of the human body. Both vertical and lateral vibrations are of interest. In this paper the vertical vibration is considered to be the most important. The first input for getting the best model of the passengers is to study in more detail the response of the human body to vertical vibration in the standing position.

Griffin studies the human body vibrations under certain environments (transport systems or transport medias) [4]. The effects of posture and vibration magnitude on the vertical apparent mass and the fore-and-aft cross-axis apparent mass of standing human body during exposure to vertical vibration were investigated by Subashi *et al* [8-9]. The nonlinear change in resonance frequency decreased from 6.39 Hz to 5.63 Hz with increasing vibration magnitude from 0.125 to 0.5 ms^{-2} r.m.s in the upright standing posture [9]. Huang summarizes the state of the art for standing subjects in his research [7]. It is important to emphasize that the biodynamic responses of the standing human body exposed to whole-body vibration have been found to be dependent on a broad range of variables. These may include: posture (e.g. kyphotic and erect), muscle activity (e.g. tensed and relaxed), body characteristics (e.g. age, mass, height and gender), vibration waveform (e.g. sinusoids, narrowband random stimuli, and broadband random stimuli), and vibration magnitude etc. A linear system will have the same dynamic behaviour (i.e. resonance frequency, and magnitude of response) with different vibration inputs. In the past two decades, the biodynamic responses of the human body have been found to be nonlinear: the resonance frequencies in frequency response functions (e.g. apparent mass and transmissibility) decrease with increasing vibration magnitude. These meritorious works refer to standing on bare feet human bodies.

Modelling of the interface of human body in sportive activities has been reported [10, 14-17]. Nikoyan reviewed the mass-spring-damper modelling of the human body to study running and hopping, thus taken into account the influence of shoes to determine the Ground Force Reaction [10]. Even-Tzur focused on shock absorption of EVA material of sport shoes [14]. These researches investigated the significant phenomena of the collision with the ground during hopping, trotting or running which is far from the mechanical response of the human body standing quietly when travels in an elevator car. A setup using a 4-meter-long reinforced concrete plank and a shaker to determine human modal parameters has been investigated using an indirect method [11]. The human-floor system was simplified by one two-degrees of freedom system. An overall frequency 5.24±0.40Hz and damping ratio 0.39±0.05 were obtained.

The current investigation focuses on the influence of human body on elevator car systems. The model developed in previous work [2, 3] is extended to assess the influence of the number of passengers. The recent research [4, 7-9] refers to the response of the human body vibrations under certain transport media standing on bare feet in the elevator system. The current investigation will then take into account the inverse cause-effect connection: the influence of the human body on such environments in particular passengers wearing different type of shoes when travelling in an elevator car to represent real operational conditions.

2 MECHANICAL MODEL

In the elevator system, vibrations in both lateral (x and z axes) and vertical (y-axis) directions are of interest. From the standing posture of the human body point of view, vibration in the y-axis is mainly considered to be the most important aspect with respect to the assessment of the influence of passengers in the car.

The simplified two degrees of freedom mechanical model of the elevator car system consisting of a passenger, car and car frame (sling) is represented in Figure 1. The vertical displacements of the passenger, car and car frame are denoted y_{PA}, y_{CA} and $y_{FR,}$ respectively. Masses of the passenger, car and car frame are defined as m_{PA}, m_{CA} and $m_{FR,}$ respectively. Passenger is supported on the car floor via a spring-damper element with the coefficients of stiffness k_{PA} and viscous damping c_{PA}. Car is mounted within the car frame on isolation blocks represented by a spring-damper element with the coefficients of stiffness k_{CA} and the coefficients of viscous damping c_{CA}, respectively.

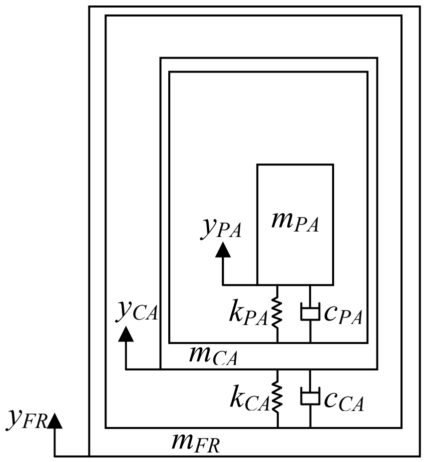

Figure 1. The simplified mechanical model of the elevator car system

The coefficients of stiffness k_{1p} and the coefficients of viscous damping c_{1p} for one passenger can be obtained via the result of a number of springs and dampers in series as follows:

$$\frac{1}{k_{1p}} = \frac{1}{k_{pa}} + \frac{1}{k_{sh}} + \frac{1}{k_{fl}}; \frac{1}{c_{1p}} = \frac{1}{c_{pa}} + \frac{1}{c_{sh}} + \frac{1}{c_{fl}} \tag{1}$$

where k_{pa} denotes the equivalent stiffness coefficient of the passenger in the vertical direction, k_{sh} denotes shoes-stockings vertical stiffness coefficient, k_{fl} denotes the vertical stiffness coefficient of the car floor. Similarly, c_{pa}, c_{sh} and c_{fl} denote the equivalent vertical viscous damping coefficients for passenger, shoes-stockings and the car floor.

Assuming that all passengers travelling in the car move in phase relative to each other and have the same mass with the average passenger's mass being m_{1p}, coefficients of stiffness and viscous damping are k_{1p} and c_{1p} as defined in Eq. (1), the resulting coefficients k_{PA} and c_{PA} for all passengers in the car are:

$$k_{PA} = nk_{1p}; c_{PA} = nc_{1p} \tag{2}$$

where n is the number of passengers travelling in the car.

As the car is attached to the car frame by a number of isolation blocks located at the each corner of the car floor, the resulting stiffness and damping coefficients for the car are:

$$k_{CA} = bk_{ib}; c_{CA} = bc_{ib} \tag{3}$$

where b is the number of isolation blocks used in the car floor, k_{ib} and c_{ib} are the stiffness and viscous damping coefficients of a single isolation block.

The car displacement transmissibility, which is the amplitude ratio of the car response to that of the car frame excitation, is coincident with the car acceleration transmissibility and can be obtained as follows [2]:

$$\left|G(\Omega)\right| = \left| \frac{\left(1+2j\zeta_{CA}\Omega\right)\left(\omega_r^2 - \Omega^2 + 2j\zeta_{PA}\omega_r\Omega\right)}{\Omega^4 - \left[\left(\omega_r^2 + 2j\zeta_{PA}\omega_r\Omega\right)\left(1+m_r\right) + \left(1+2j\zeta_{CA}\Omega\right)\right]\Omega^2 + \left(1+2j\zeta_{CA}\Omega\right)\left(\omega_r^2 + 2j\zeta_{PA}\omega_r\Omega\right)} \right| \tag{4}$$

where

$$\omega_r = \frac{\omega_{nPA}}{\omega_{nCA}}; m_r = \frac{nm_{1p}}{m_{CA}}; \Omega = \frac{\omega}{\omega_{nCA}}; \omega_{nCA}^2 = \frac{k_{CA}}{m_{CA}}; \omega_{nPA}^2 = \frac{nk_{1p}}{nm_{1p}}; \zeta_{CA} = \frac{c_{CA}}{2m_{CA}\omega_{nCA}}; \zeta_{PA} = \frac{nc_{1p}}{2nm_{1p}\omega_{nPA}} \tag{5}$$

where ω is the angular frequency of the excitation source, ζ_{PA} and ζ_{CA} are the damping parameters for passengers and car respectively, ω_r is the angular velocity ratio, and m_r is the mass ratio of all passengers' mass to the car mass. It can be clearly observed that ω_{nPA}, ζ_{PA} and ω_r are independent on the number of passengers travelling in the car. Therefore, the car displacement transmissibility $\left|G(\Omega)\right|$ is directly dependent on n due to the fact that m_r is dependent on n, see Eqs. (4, 5).

3 TRANSMISSIBILITY SPECTRUM

The fundamental criterion for the design of isolation block of an elevator car is to diminish the magnitude of the transmissibility $\left|G(\Omega)\right|$ for any load of the elevator. In other words, for each m_r ranging from zero to 100% load, excitation frequency $\omega/2\pi$ should be within the required range.

The magnitude of the transmissibility $\left|G(\Omega)\right|$ in Eq. (4) is studied for an elevator used in office buildings, a typical 17 passenger elevator, with the following parameters: m_{CA}=500kg, k_{ib}=213898N/m, b=10, ζ_{CA}=0.032, ζ_{PA}=0.3, m_{1p}/m_{CA}=0.15 and k_{1p}/k_{CA}=0.05.

The natural frequency of the car is determined as $\omega_{nCA}/2\pi$=10.41Hz which generates a frequency spectrum in the range of interest 0 to 20Hz. Figure 2 shows the transmissibility curve of the above elevator with different load occupancy, i.e. different number of passengers, with respect to a series of the mass ratio m_r. It can be seen that the results are influenced significantly by the stiffness coefficient k_{1p} for one passenger and the damping coefficient ζ_{PA} due to the factor that these parameters depend mainly on the response of human body.

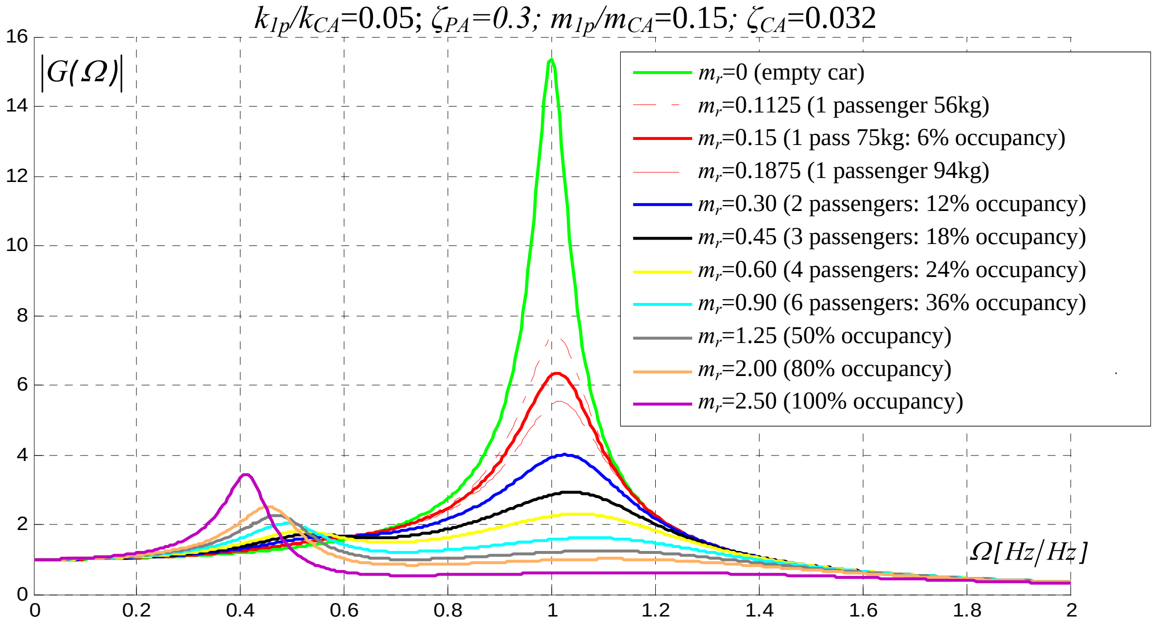

Figure 2: Transmissibility for the 17 passengers' elevator with different load occupancy.

4 RESONANT FREQUENCIES

The frequencies to which the car acceleration transmissibility $|G(\Omega)|$ reaches maximum magnitude is of much interest in measuring the lift ride quality. These correspond to the resonant frequencies that, generally, are not the same as the un-damped or damped natural frequencies [19]. The level of $|G(\Omega)|$ in the frequency range 1-20Hz is used to design the isolation blocks of the elevator. The resonant frequencies can be computed for all ω_r, m_r, and ζ_{PA} at very low value of ζ_{CA} [18, 19]. Parameters to be optimized for the optimal high ride quality purposes are k_{CA} and ζ_{CA} for the car and, to some extend, ζ_{PA} for passengers, which can be accomplished by changing the car floor covering characteristics or relevant parameter in Eq. (1). The resonant frequencies can be obtained by making ζ_{CA} close to zero in Eq. (4) as:

$$|G(\Omega)| = \sqrt{\frac{\left(\omega_r^2 - \Omega^2\right)^2 - 4\zeta_{PA}^2\omega_r^2\Omega^2}{\left[\Omega^4 - \left[\omega_r^2\left(1+m_r\right)+1\right]\Omega^2 + \omega_r^2\right]^2 - 4\zeta_{PA}^2\omega_r^2\Omega^2\left[1-\left(1+m_r\right)\Omega^2\right]^2}} \tag{6}$$

The resonant frequencies coincide with the condition of the denominator in Eq. (6) equals to zero. Obviously, it is not an easy task to get the roots of such an expression. Following the graphical analysis by Hartog [20], a series of curves of the transmissibility for different values of ζ_{PA} are plotted and it is observed that all of them intersect each other in two points P and Q, P for the lowest frequency and Q for the highest frequency. As reported in the previous research [2], there is some insignificant shift for the case of low damping of the car ζ_{CA}. Therefore, estimating the resonant frequencies for P and Q becomes much easier by making Eq. (6) in the following form:

$$|G(\Omega)| = \sqrt{\frac{A\zeta_{PA}^2 + B}{C\zeta_{PA}^2 + D}} \Leftrightarrow A/C = B/D \Rightarrow \Omega^4 - 2\Omega^2\frac{1+\omega_r^2\left(1+m_r\right)}{2+m_r} + \frac{2\omega_r^2}{2+m_r} = 0 \tag{7}$$

The roots of Eq. (7) can then be obtained as:

$$\Omega_{P,Q}^2 = \frac{1+\omega_r^2(1+m_r)}{2+m_r} \pm \sqrt{\left[\frac{1+\omega_r^2(1+m_r)}{2+m_r}\right]^2 - \frac{2\omega_r^2}{2+m_r}} \tag{8}$$

Figure 3 shows both roots plotted in solid lines with blues for $10\Omega_P$ and reds for $10\Omega_Q$ for a number of angular velocity ratios ω_r with respect to the mass ratio m_r from zero to 2.4 for $k_{1p}/k_{CA}=0.1$. The transmissibility for the non-dimensional frequencies Ω_P and Ω_Q are also plotted in dashed lines with blues $G(\Omega_P)$ and reds for $G(\Omega_Q)$.

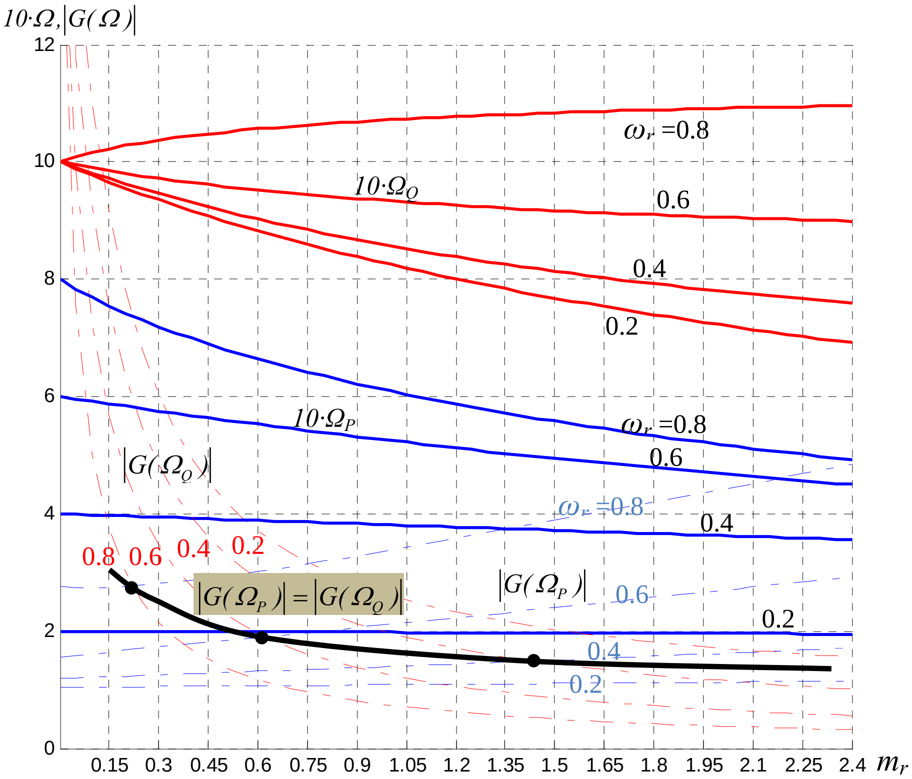

Figure 3: Non-dimensional frequency and Transmissibility of P and Q

Assuming that the angular frequency ratio ω_r is below 0.6, which is a common practice in the elevator's design, Figure 3 shows that increase in the mass ratio m_r from zero reduces the P and Q frequencies from $10\cdot\Omega_P=10\cdot\omega_r$ (that is: $\Omega=\omega_r$ or $\omega=\omega_{nPA}$ and then the excitation frequency is coincident with the natural frequency of the human body) and $10\cdot\Omega_Q=10$ (that is: $\Omega=1$ or $\omega=\omega_{nCA}$ and then the excitation frequency is coincident with that for the empty car), respectively. Also, the maximum transmissibility of P is below 3 and the maximum transmissibility of Q is over 4 (if m_r is less than 0.15). Then, it is more critical the transmissibility for higher frequencies over the point Q than that of lower frequencies below the point P. Figure 3 also shows that increase in the mass ratio m_r increases the transmissibility for $|G(\Omega)_P|$ and reduces the transmissibility for $|G(\Omega)_Q|$. Hence, the angular frequency ratio for optimum transmissibility ω_{ropt} can be obtained as follows [20]:

$$\forall m_r : |G(\Omega_P)| = |G(\Omega_Q)| \Leftrightarrow \omega_{ropt} = \frac{1}{1+m_r} = \alpha \Leftrightarrow \sqrt{\frac{k_{1p}}{k_{CA}}}\sqrt{\frac{m_{CA}}{m_{1p}}} = \alpha \Rightarrow \frac{k_{CA}}{k_{1p}} = \frac{1}{\alpha^2}\frac{m_{CA}}{m_{1p}} \tag{9}$$

In order to achieve peak magnitudes of the transmissibility at the points P or Q, passengers damping coefficient for an optimal absorber ζ_{PAopt} can be established [18-20]:

$$\zeta_{PAopt} = \sqrt{\frac{3m_r}{8(1+m_r)^3}} \tag{10}$$

This leads to a number of proposed car stiffness coefficients and optimal passengers damping coefficients and the results are shown in Table 1.

There are two typical elevator systems investigated, residential buildings elevators (4-6-8 passengers) and office buildings elevators (13-17 passengers). In residential buildings elevators, m_r varies from 0 to 1.65 with the initial step as 0.265; in office buildings elevators, m_r varies from 0 to 2.564 with the smallest step as 0.15 [21]. The special case m_r=0, i.e. the empty car, is not important when considering the ride quality but for maintenance or sound effects influencing the off pit structure. A case of one passenger occupancy is considered to be the most important one in testing protocols for assessing lift ride quality [1] and also it is the most frequent usage in practice in residential buildings with low traffic.

Elevator Type		m_r	m_{CA}/m_{1p}	ω_{ropt}	k_{CA}/k_{1p} (ω_{ropt}-)	k_{CA}/k_{1p} (ω_{ropt}+)	$\overline{k_{CA}/k_{1p}}$	$\overline{\omega_r}$	ζ_{PAopt}	$\lvert G(\Omega_P)\rvert$
No of Passengers	Q [kg]									
4	300	0.265-1.06	3.77	0.79-0.49	6.04	16.01	7.2 6.04[b] 6.04[b]	0.724 0.79 0.79	0.22-0.21	4.36 3.31[c] 5.57[d]
6	450	0.225-1.35	4.44	0.82-0.42	6.67	24.55	8.53	0.722	0.21-0.20	5.29
8	600	0.207-1.65	4.84	0.83-0.37	7.05	34.06	9.55	0.712	0.21-0.20	5.97
13	975	0.154-1.60[a]	6.51	0.87-0.38	8.66	43.93	12.42	0.724	0.19-0.18	8.00
17	1275	0.15-2.05[a]	6.64	0.87-0.33	8.79	61.70	13.60	0.699	0.19-0.16	10.1
Freight 1 passenger	1275	0.15-0.05	6.64-22.6	0.87-0.96	8.79	24.68	18.42	0.60-1.11	0.19 0.12	20.46

Table 1: Optimal car stiffness ratio k_{CA}/k_{1p} for maximum transmissibility and passengers damping ζ_{PAopt} to different rated loads Q (a: Restricted to 80% of maximum occupancy ratio; b: Minimize transmissibility for 1 passenger; c: 1 passenger travel; d: full occupancy travel)

5 EXPERIMENTAL RESULTS

A laboratory model of the elevator car system was tested with closed loop displacement control and varying amplitude, phase and frequency for excitation acceleration amplitude of 0.250 ms^{-2} r.m.s [2]. The laboratory rig was designed to match the configuration of a typical 312kg car with six passengers capacity mounting four rubber isolation blocks. The following parameters are used: m_{CA}=312 kg; k_{ib}=0.267MN/m; ζ_{CA} =0.032. The rig was tested to confirm the isolation blocks specifications and to determine the stiffness k_{1p} and damping ratio ζ_{PA} for one passenger wearing two types of shoes [3].

In the current study, the rig was tested for three passengers wearing moccasin shoes first and then sport shoes. Transmissibility of acceleration amplitudes was obtained when the frequency sweep of the excitation is applied from 1 to 20Hz. The average of the three passengers was calculated and the corresponding mass ratio m_{1p}/m_{CA} was then determined. The transmissibility for the three passengers (n=3) were obtained using Eq. (4). Figure 4 shows the experimental results of the transmissibility with the theoretical values for different passengers involved.

Despite the simplifications made and the complex dynamic characteristics of the actual passenger system, the experimental results correspond very well to the theoretical model predictions.

It is evident that an increasing number of passengers loaded on the car attenuates considerably its acceleration amplitudes in the dominant range of frequencies which is that close to the natural frequency of the empty car (Ω=1). It is shown how full occupancy of passengers cars resonance is shifted to a range of frequencies which is close to the natural frequency of the human body ($\Omega=\omega_r$). Anyway, the amplitudes of this acceleration are lower than those of the empty car and one passengers car. It is shown how the type of shoes worn by the passengers influences the acceleration amplitudes of the car.

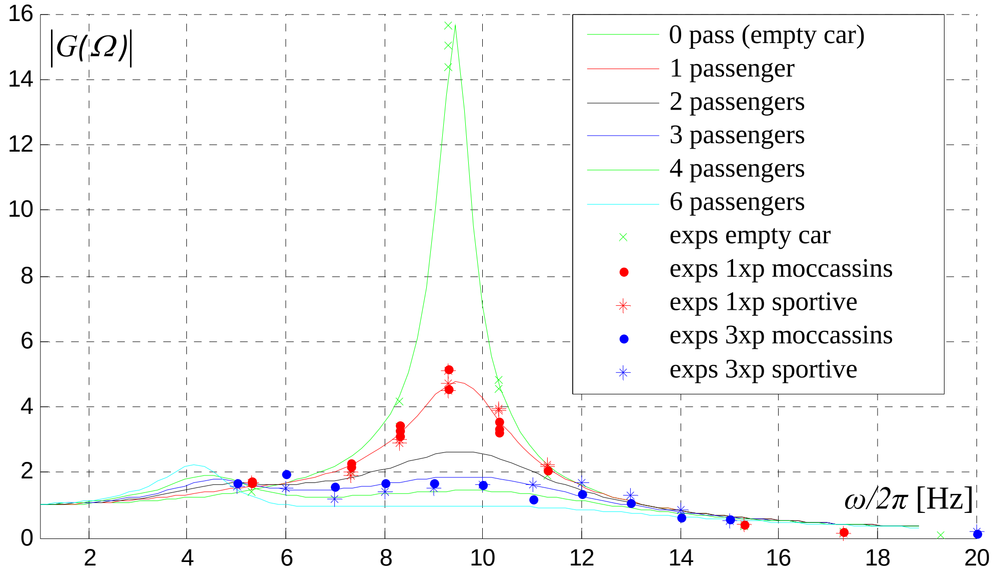

Figure 4. Theoretical and experimental transmissibility comparison

6 CONCLUSSIONS AND FUTURE WORK

The mechanical response of an elevator car under a range of different passenger load conditions, from empty to full, has been investigated in the paper. Increasing the number of passengers reduces considerably the transmissibility of the car acceleration in the dominant range of frequencies which is close to the empty car resonance frequency condition. It is clear that low occupancy makes the transmissibility excessive in the proximities of the natural frequency of the empty car. On the other hand, under higher passenger load the passengers act as as a high efficiency dynamic absorber system except in the frequency range lower than the natural frequency of the human body (5-7 Hz). The most unfavourable configuration is that of the empty car. Of course, no comfort or ride quality is compromised with the empty car but for maintenance purposes or off pit structure problems. The next unfavourable configuration is clearly for low occupancy, i.e. 1 or 2 passengers. The optimum design for comfort and ride quality should be based in this transmissibility. In the case of any additional economic reasons a compromise between this configuration and that for full occupancy should prevail. The machine should be conveniently isolated to avoid any excitation below 7 Hz and the suspension means should be checked for not inducing any excitation below 7 Hz.

Experimental tests with a car loaded with three passengers wearing two types of shoes have been carried out. The elevator's car was excited over a range of frequencies and amplitudes. An excellent agreement of the results from experimental tests with the model predictions has been achieved. The technique presented in this paper seems to be sensitive enough as to measure the mechanical response of passengers wearing different type of shoes in order to provide criteria for optimal covering floor properties of elevator cars.

7 ACKNOWLEDGEMENTS

Authors would like to thank Mr. José Luis Yus from MP Ascensores for his help with the compilation of weight distribution data of commercial elevators.

REFERENCES

[1] ISO 2003, Lifts_Measurement of lift ride quality. *ISO 18738:2003(E)*

[2] I. Herrera and S. Kaczmarczyk, The Assessment of Vibration Absorption Capacity of Elevator's Passengers. *7th International conference on Modern Practice in stress and Vibration Analysis*, Journal of Physics: Conference Series 181, 2009.

[3] I. Herrera, H. Su and S. Kaczmarczyk, Investigation into the damping and stiffness characteristics of an elevator car system. *Applied Mechanics and Materials,* Trans Tech Publications, Vols. 24-25,77-82, 2010.

[4] M.J. Griffin: Handbook of Human Vibration. *Academic Press Limited*, London, 1990.

[5] J. Rosen and M. Arcan, Modeling the Human Body/Seat System in a Vibration Environment. *Journal of Biomechanical Engineering* Vol 125, 223-231, 2003.

[6] X. Linan, Z. E, L. Mingli, S. Xiaochun and Z. Fan, Human Vibration Characteristic and Experiment Research on Man-Machine System in Dynamic *Environment. 9th Inernational Conference on Computer-Aided Industrial Design and Conceptual Design.* 169-174*,* 2008.

[7] Ya Huang, Mechanism of nonlinear biodynamic response of the human body exposed to whole-body vibration. *Thesis. Faculty of Engineering, Science and Mathematics, Institute of Sound and Vibration, University of Southampton.*

[8] G.H.M.J. Subashi, Y. Matsumoto and M.J. Griffin, Apparent mass and cross-axis apparent mass of standing subjects during exposure to vertical whole-body vibration. *Journal of Sound and Vibration,* 293, 78-95, 2006.

[9] G.H.M.J. Subashi, Y. Matsumoto and M.J. Griffin, Modelling resonances of the standing body exposed to vertical whole-body vibration: Effects of posture. *Journal of Sound and Vibration,* 317, 400-418, 2008.

[10] A.A. Nikoyan, A.A. Zadpoor, Mass-spring-damper modelling of the human body during running and hopping - an overview. *Proceedings of the Institution of Mechanical Engineers, Part H: Journal of Engineering in Medicine*. Review 1121-1135. 2011.

[11] X. Zheng and J.M.W. Brownjohn, Modeling and simulation of human-floor system under vertical vibration. Smart *Structures and Materials 2001*. Proceedings of SPIE 4327, 2001.

[12] T.C. Gupta, Identification and Experimental Validation of Damping Ratios of Different Human Body Segments Through Anthropometric Vibratory Model in Standing Posture. *Transactions of the ASME*, Vol 129, 566-574, 2007.

[13] F.M.L. Amirouche, M. Xie, A. Patwardhan, Optimization of the Contact Damping and Stiffness Coefficients to Minimize Human Body Vibration. *Journal of Biomechanical Engineering*, Vol 116, 413-420,1994.

[14] N. Even-Tzur, E. Weisz, Y. Hirsch-Falk and A. Gefen, Role of EVA viscoelastic properties in the protective performance of a sport shoe: Computational studies. *Bio-Medical Materials and Engineering 16*, 289-299, 2006.

[15] Q.H. Ly, A. Alaoui, S. Erlicher, and L. Baly, Towards a footwear design tool: Influence of shoe midsole properties and ground stiffness on the impact force during running. *Journal of Biomechanics* Vol 43, 310-317, 2010.

[16] J.F. Wilson and R. D. Rochelle, Football dynamics for racewalkers and runners barefoot on compliant surfaces. *Journal of Biomechanics* Vol 42, 2472-2478, 2009.

[17] W. Qassem, Impulse response due to jumping on shoes of various stiffness and damping. *Bio Medical Materials and Engineering* 13, 167-180, 2003. IOS Press.

[18] B. Balachandran, Vibraciones. *Thomson*. 2006.

[19] Da Silva, Vibration, Damping, Control, and Design. *CRC Press.* 2007.

[20] H.P. Den Hartog, Mechanical Vibrations. *Mc Graw-Hill*. 1956

[21] J.L. Yus, Weight distribution of elevator car components for different sizes. *MP Ascensores*. Zaragoza. Spain, 2013. Internal Report.