Study on the vertical vibration of an occupant - seat cushion system

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ABSTRACT

Dynamic interaction between a human body and the seat cushion was addressed, aiming at seat design for better NVH performance of a passenger vehicle. Referring to DIN EN ISO 3386-1 and previously researches, static and dynamic tests were conducted on polyurethane foams typically used as vehicle seat cushions. A one-dimensional stress strain relation was derived based on the measured data, in which the nonlinear elastic behavior of the foam material is regarded as piecewise linear in terms of the static strain while the viscoelastic property is described using a fractional derivative. A model was then established with some simplification to reflect the interaction in the vertical direction between a foam cushion and a seated occupant, where the seated human body was depicted by a lumped parameter model. Apparent mass of the system was calculated using the model, which were validated by experiments with or without seat cushion. The influence of the foam cushion is thus demonstrated on the vertical vibration of the occupant, which will be helpful to vehicle seat design.

Keywords: Occupant vibration, Seat cushion I-INCE Classification of Subjects Number(s): 46, 47

1. INTRODUCTION

Ride comfort has become one of the key performances in vehicle design evaluation. The up to date evaluation has mainly been based on subjective experiences. Evaluation based on objective measurement or simulation on the other hand, is gaining momentum, and researches on related topics are attracting more and more attention. To this end, besides many other investigations, it is necessary to establish reasonable biodynamic models for a seated human body, to accurately describe the static and dynamic behavior of seat cushion materials, and to understand the interaction between a vehicle occupant and the seat cushion.

Researches on vertical vibration characteristics of a seated human body might be dated back to Coermann's work in 1962 (1). Owing to long time efforts of many scholars in the west, quite a few national or international standards have been promulgated, including ISO5982-1981 and ISO5982-2001. Maeda and Mansfield' work published in 2005 shows that there exists clear difference between model parameters for the apparent masses of the Japanese people and those presented in ISO 5982-1981 based on the western peoples (2). Since early 1980s, Chinese researchers from Changchun Automobile Research Institute, Tsinghua University and so on have conducted investigation on vertical vibration of seated Chinese human beings (3-6), which also reveal the difference between Chinese and western people in terms of vibrational parameters.

The seats used in many vehicles rely on seat cushion to isolate vibration or impact. As a widely used cushion material, soft polyurethane foam with open cells presents nonlinear stress-strain relation and hysteresis even when subjected to uniaxial compressive loading and unloading. To describe the nonlinear elasticity and linear or nonlinear viscoelasticity, many constitutive laws have been put forward. Compared with analytical constitutive laws, phenomenological relations, including those with the convolution integral or the fractional derivatives (7-9), are more applicable to engineering problems.

Nishiyama (10) built an occupant-seat model of 6 degrees of freedom to describe the vibrations of a human body seated on a vehicle seat. In his model, a multi-body dynamic model was used for the seated human body, while linear springs and dashpots were used to reflect the interaction between the

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human body and seat at different parts. Taking into consideration the nonlinear stiffness and damping property of the seat cushion materials, Patten and Pang (11), and Wu and Rakheja et al (12) studied the vertical vibration of a cushion-human body system using a biodynamic model of a seated human body with two or three degrees of freedom, respectively. Bajaj and his colleagues (13) calculated the vibrational responses of a seated human body by a model which is a combination of the Nishiyama's human body model and a convolution integral model for the seat cushion.

This paper tries to present in the following four sections an approach for vertical vibration analysis on a human body-seat cushion system aiming at Chinese people and using fractional derivation model for the cushion foam. Section 2 briefly introduce researches on seated human bodies, namely vibrational tests, biodynamic modeling and parameter identification. In Section 3, the stress and strain relations of one polyurethane foam widely used in China mainland as vehicle seat cushions are discussed with typical data from the pseudo-static and dynamic tests, and a constitutive relation is then constructed by combining a piecewise linear strain function and a fractional derivative. Based on the work in the aforementioned two sections, a vertical vibration model is firstly established for a seated human being and cushion configuration in the fourth section. In section 5, calculation and experiments are carried out respectively to figure out apparent masses of the system. Comparisons validate the applicability of the presented approach and exemplify contribution of the cushions to isolate potential vibrations. The last section is a summary about the work.

2. BIODYNAMIC MODEL OF SEATED HUMAN BEINGS

Extensive experiments have been performed on a generation of Chinese people by referring to ISO 5982-2001, aiming at a better understanding with the vertical vibration characteristics of Chinese human body. Two biodynamic models were chosen to describe these characteristics, and model parameters were figured out respectively (5, 6). The homemade seat with load cells used in the experiments is shown in Figure 1, and one of the biodynamic model is depicted in Figure 2.

Figure 1 – Home-made seat sensor

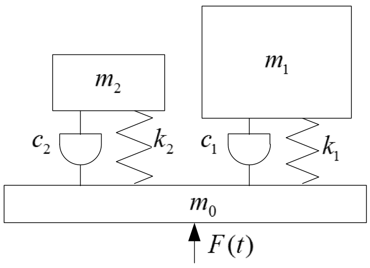


Figure 2 – A model for seated human bodies

In Figure 2, $m_p\,(p = 0,1,2)$ are masses, $c_j, k_j\,(j = 1,2)$ are damping and stiffness coefficients and $F(t)$ is the external excitation.

Previous researches of the current authors (14) and other scholars (15) have revealed that there is in a range of 0-20.0Hz no more than 2 resonance peaks in the amplitude-frequency curve of the vertical apparent mass or driving point impedance of a seated human body, and that the model shown in Figure 2 can be accurate enough when model parameters are identified properly. As a result, the model shown in Figure 2 will be chosen for construction of an occupant-seat model in section 4.

Assume $F(\omega)$ and $Z_0(\omega)$ as the Fourier transforms of the external force and displacement measured at the seat surface, respectively. According to the definition of the apparent mass, one can get after derivation the expression for the apparent mass describing the vertical vibration property of a seated human body as follows

$$H_2(\omega) = \frac{F(\omega)}{-\omega^2 Z_0(\omega)} = \sum_{p=1}^{2} \frac{m_p(k_p + \mathrm{i}\omega c_p)}{k_p - m_p\omega^2 + \mathrm{i}\omega c_p} + m_0, \quad \mathrm{i} = \sqrt{-1} \tag{1}$$

The problem of model parameter identification can be solved by a specifically defined optimization procedure (6). For the optimization, an objective function is defined in terms of the amplitude and phase angle of the apparent mass simultaneously, in which the measured data and derived expressions from Equation (1) are combined. The resulted nonlinear least square problem can be changed to a series of linear algebraic problems by the Gauss-Newton algorithm. The approach can be used to identify model parameters of any single volunteer using directly measured data, or to figure out model parameters for a group of volunteers using the average of the measured data.

3. FOAM MODEL AND PARAMETER IDENTIFICATION

In order to understand the mechanical behavior of a polyurethane foam material widely used in China mainland as vehicle seat cushions, pseudo-static tests and dynamic tests have been performed on material specimens, referring to the standard DIN EN ISO 3386-1 and the work of White et al (16). The dimension and density of the specimens are 75 mm×75 mm×75 mm and 40 kg/m^3, respectively. During the tests, temperature and humidity within the lab were kept separately at around 23℃and 50%. As almost no lateral deformation was observed from the specimens during the tests, the cross section area of the specimens can be assumed to be constant. Furthermore, influence of the specimen dimension variation was neglected according to the conclusion from Chou and his colleagues' research (17). Dynamics tests were implemented at several pre-compression levels, which was defined in the light of possible static deflections of a vehicle seat cushion (18).

Typical measured data will be presented in this section, on which a brief analysis was made to highlight the dynamic stress-strain relation of the cushion material at various pre-compression values. Based on the analysis and for predicting occupant-seat interaction, a stress-strain model is then proposed using a fractional derivative, and the procedure to identify model parameters is also sketched.

3.1 Mechanical Properties of the Cushion Material

Figure 3 shows the dynamic stress and strain of the foam material subjected to different pre-compressions and dynamic harmonic excitation with an identical frequency of 6.0Hz but different strain amplitudes. To have a complete scenario of mechanical property of the foam material, a stress-strain curve is also presented in the figure which is obtained at pseudo-static compression tests.

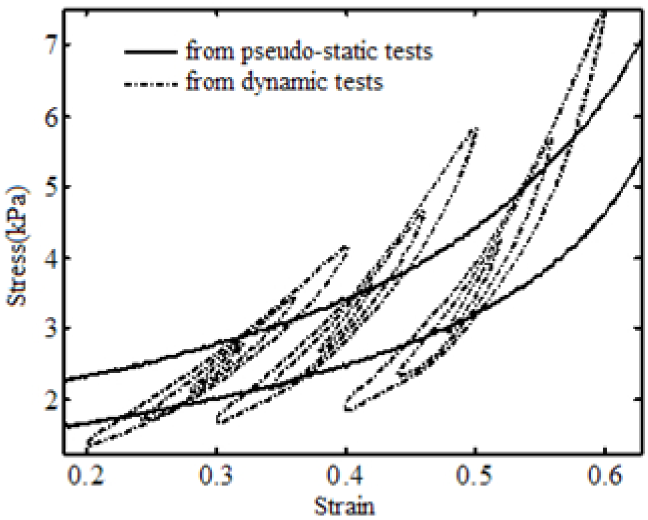


Figure 3 – Typical stress-strain curves of the form obtained from tests

It is noted from the figure that hysteresis loops exist with the dynamic strain and stress of the foam material when subjected to a simple harmonic excitation at various pre-compressions. For a specific pre-compression and an exciting frequency, the slope of the long axis of the dynamic hysteresis loop decreases with an increasing excitation amplitude. For a dynamic excitation with fixed frequency and amplitude on the other hand, the slope varies with the pre-compression as expected.

Appling Fourier transform to the measured stress and strain data, respectively, and calculating the ratio of the transformed stress and strain in the frequency domain, yield the dynamic stiffness $G(\omega)$ of the material if the amplitude of the dynamic excitation is small enough. For a fixed dynamic strain amplitude of 0.02, the moduli of the dynamic stiffness of the cushion foam under 3 pre-compression levels are depicted in Figure 4 with respect to the exciting frequency.

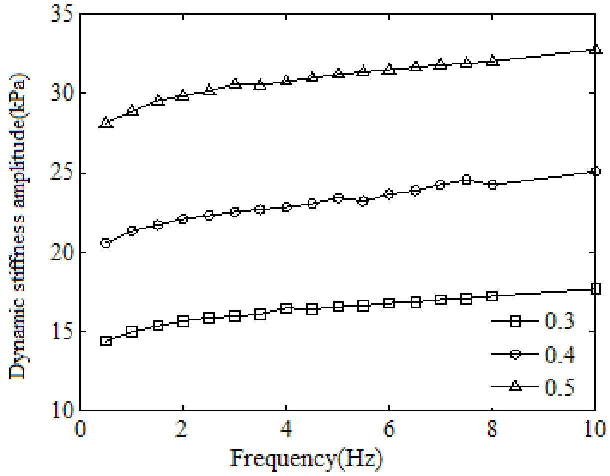


Figure 4 – Dynamic stiffness of the form at a dynamic strain amplitude of 0.02

It can be observed from Figure 4 that the amplitude of the dynamic stiffness increases with the increased pre-compression level for a fixed exciting frequency, and increases slowly with the exciting frequency for a specific pre-compression value. The observations are a clear reflection of the nonlinear elasticity and frequency-dependent property of the cushion foam.

3.2 A Constitutive Relation for the Cushion Foam

The observations with those measured data clearly demonstrate that an accurate stress strain relation of the cushion foam should be able to describe not only the nonlinear elasticity, but also the frequency dependence. To this end, the constitutive relation can be constructed by combining a strain polynomial and a convolution integral or fractional derivative with respect to strain and strain rate, where the strain polynomial is responsible for the nonlinear elasticity while the rest terms cover frequency dependence.

The focus of this paper is the vertical vibration characteristics of the occupant-seat system under static equilibrium. As the dynamic excitation to a seat is usually small during most time of vehicle operations, it should be applicable to linearize the elastic stress of the seat cushion at any static equilibrium. As a result, the dynamic stress – strain relation of the cushion foam can be approximated as

$$\sigma = k\varepsilon + \sigma_{\mathrm{V}} \tag{2}$$

Here k is an elastic parameter which varies with static equilibrium or the pre-compression levels, and σ_{V} represents the viscoelastic stress. Applying Fourier transform to Equation (2), one get the dynamic stiffness in the frequency domain as

$$G(\omega) = k + \frac{\sigma_{\mathrm{V}}(\omega)}{\varepsilon(\omega)} \tag{3}$$

According to the observations from Figure 4, it can be figured out that the second term in the right hand side of Equation (3), namely the viscoelastic component, should capture the frequency dependence of the dynamic stiffness. Obviously, the frequency dependence can't be achieved by simply using a viscous damping coefficient or a product of the coefficient and the strain rate. Instead, more complex expressions should be adopted. After analysis and comparison, we use the following model with fractional derivatives to describe the material viscoelasticity

$$\mathrm{D}^{\alpha}\left[\sigma_{\mathrm{V}}(t)\right] + b\sigma_{\mathrm{V}}(t) = C_1\varepsilon(t) + C_2\,\mathrm{D}^{\beta}\left[\varepsilon(t)\right] \tag{4}$$

where the Riemann – Liouville definition is used for the fractional derivative as

$$\mathrm{D}^{\alpha}\left[x(t)\right] = \frac{d^{\alpha}x(t)}{dt^{\alpha}} = \frac{d^{n}}{dt^{n}}\left[\frac{1}{\Gamma(n-\alpha)}\int_{-\infty}^{t}\frac{x(\tau)}{(t-\tau)^{\alpha-n+1}}d\tau\right] \tag{5}$$

In the definition, n is a positive integer satisfying $n-1<\alpha<n$, while $\Gamma(a)=\int_0^{\infty} y^{a-1}e^{-y}dy$.

To establish the model presented by Equations (2) and (4), an optimization problem is defined as follows to figure out the model parameter vector $\boldsymbol{x} = \begin{bmatrix} k & a & \beta & b & c_1 & c_2 \end{bmatrix}^{\mathrm{T}}$, where the objective function is

$$Q = \min_{\boldsymbol{x}} \sum_{n=1}^{N} \left\{ \left[G\left(\omega_n, \boldsymbol{x}\right) - G_{\mathrm{m}}\left(\omega_n\right) \right] \cdot \left[\overline{G}\left(\omega_n, \boldsymbol{x}\right) - \overline{G}_{\mathrm{m}}\left(\omega_n\right) \right] \right\} \tag{6}$$

Here $G\left(\omega_n, \boldsymbol{x}\right)$ and $G_{\mathrm{m}}\left(\omega_n\right)$ are the calculated and measured dynamic stiffness at a frequency of ω_n, respectively, while $\overline{G}\left(\omega_n, x\right)$ and $\overline{G}_{\mathrm{m}}\left(\omega_n\right)$ are their conjugates in sequence. N is the number of the sampling frequencies in the dynamic tests.

As the model should conform to thermodynamics, a set of constraint conditions can be derived as below

$$0 \le \beta - \alpha \le 1,\ C_2 \ge 0,\ k \ge 0,\ bC_2 \ge C_1,\ k \ge -\frac{C_1}{b} \tag{7}$$

The optimization problem can be solved by colligating the two functions of *MultiStart* and *fmincon* in Matlab.

4. MODELING ON A HUMAN BODY-SEAT SYSTEM

A simple configuration consisting of a seated human body, a foam cushion and a seat, as shown in Figure 5, is used to exemplify the influence of a seat cushion on the vertical vibration of a seated human being. In the configuration, one piece of the aforementioned polyurethane foam with a dimension of 380 mm×380 mm×100 mm is directly put on the seat shown in Figure 1, and a volunteer is seated on the foam.

By applying the models of the seated human body and the foam material presented in previous sections, a mechanical model of the target system can be constructed as shown in Figure 6. In the figure, m_{s} is the mass of the faceplate of the seat, $F(t)$ is the external force applied to the seat faceplate, σ_{E} is the elastic stress, and z_{s} and z_0 are the displacements of the seat faceplate and the top surface of the cushion, respectively.

Figure 5 – A human body-seat system

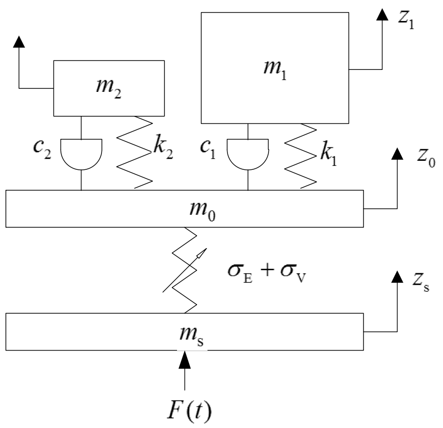


Figure 6 – Dynamic model of the system

Equations of motion and vibration properties of the system will be derived in the following subsections according to the model depicted in Figure 6.

4.1 Motion Equation of the System

Using the Newton's second law, the vertical vibration equation of the human body–seat system shown in Figure 6 can be written as

$$m_{\mathrm{s}} \ddot{z}_{\mathrm{s}} + V\left(t\right) = F(t) \tag{8}$$

In the equation, $V\left(t\right)$ is the supporting force supplied by the foam cushion during vibration.

Assuming the initial thickness of the foam cushion and the equivalent contact area between the human body and the cushion are h and S, respectively, the supporting force can be expressed as

$$V(t) = S\sigma(t) = S\left[\sigma_{\mathrm{E}}(t) + \sigma_{\mathrm{V}}(t)\right] \tag{9}$$

Recalling Equation (2) for the foam material model, and assuming the pre-compressive strain of the cushion at equilibrium to be $\varepsilon(t)$, Equation (9) can be rewritten as

$$V(t) = S\left[k\varepsilon(t) + \sigma_{\mathrm{V}}(t)\right], \text{ with } \varepsilon(t) = \frac{z_{\mathrm{s}} - z_0}{h} \tag{10}$$

4.2 Apparent Mass of the System

Substituting Equation (4) into Equation (10), and applying Fourier transform to the resulted equation, yield

$$V(\omega) = S\left[k + \frac{C_1 + C_2(\mathrm{i}\omega)^{\beta}}{b + (\mathrm{i}\omega)^{\alpha}}\right]\frac{Z_{\mathrm{s}}(\omega) - Z_0(\omega)}{h} \tag{11}$$

The supporting force can also be determined according to the interaction between the cushion and the human body. Referring to Figure 6, one can readily have with some derivation

$$V(\omega) = -\omega^2 Z_0(\omega) H_2(\omega) \tag{12}$$

Applying Fourier transform to Equation (8), and using Equations (11) and (12), the vertical apparent mass of the human body-seat system shown in Figure 6 can be derived according to the definition as

$$M(\omega) = m_{\mathrm{s}} + \frac{H_2(\omega)}{1 - \dfrac{\omega^2 h H_2(\omega)}{S\left[k + \dfrac{C_1 + C_2(\mathrm{i}\omega)^{\beta}}{b + (\mathrm{i}\omega)^{\alpha}}\right]}} \tag{13}$$

In Equations (12) and (13), if the vertical vibration property of a seated human body is described by one of other biodynamic model, one can find that Equation (13) is also applicable by substituting the corresponding apparent mass $H_{\mathrm{m}}(\omega)$ for $H_2(\omega)$. At this situation, the system model shown in Figure 6 should be reconstructed by using the corresponding model of the seated human body to replace the one shown in Figure 2.

5. EXAMPLES AND DISCUSSIONS

In this section, the apparent mass of the human body-seat system will be computed according to Equation (13) using the identified model parameters of one volunteer and the polyurethane foam. Vibration tests were carried out on the physical configuration to directly measure the apparent mass of the system. Comparisons are made to validate the calculation and to understand the value of the foam cushion in vibration isolation.

5.1 Basic Data

Using the method and procedure presented in section 2, one can figure out the model parameters for the volunteer, as shown in Table 1. The equivalent mass of the seat faceplate was also obtained by experiments as m_{s}=8.1 kg.

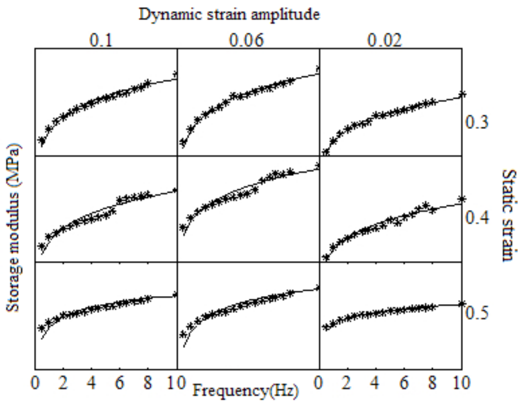
Table 1 – Model parameters of a seated human body

Mass, kg			Damping, N·s/m		Stiffness, N/m	
m_0	m_1	m_2	c_1	c_2	k_1	k_2
6.0	29.7	9.8	725.0	555.0	2.591×10^4	6.211×10^4

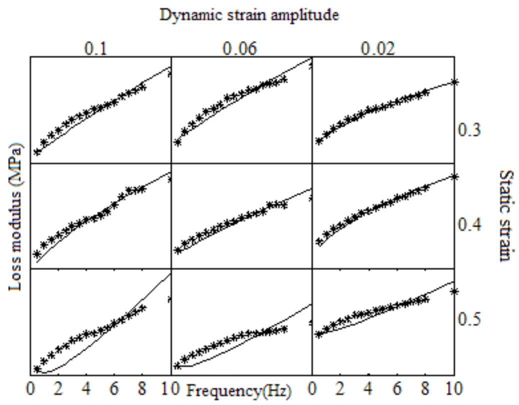
For the cushion foam material, by solving the optimization problem as introduced in section 3, the parameters in the constitutive relations under three pre-compression levels are determined as given in Table 2. The accuracy of the model is checked by comparing the calculated storage modulus and loss modulus of the foam with their counterparts from experiments, Figure 7.

Table 2 – Constitutive constants of the cushion foam

Static strain	Dynamic strain amplitude	b, $s^{-\alpha}$	α	β	C_1, $kPa \cdot s^{-\alpha}$	C_2, $Pa \cdot s^{\beta-\alpha}$	k, kPa
0.3	0.1	0.299	0.016	0.879	-136	45	87.9
	0.06	0.302	0.018	0.856	-138	48	89.4
	0.02	0.299	0.018	0.622	-131	180	87.7
0.4	0.1	0.299	0.020	0.707	-137	172	92.9
	0.06	0.351	0.102	0.795	-347	168	34.6
	0.02	0.299	0.025	0.585	-123	371	88.3
0.5	0.1	0.409	0.184	1.184	-30.5	62	36.2
	0.06	0.397	0.158	1.158	-36.9	60	40.3
	0.02	0.366	0.107	1.107	-55.7	64	54.2



(a) Storage modulus



(b) Loss modulus

Figure 7 – Dynamic moduli from calculation and tests (* measured; — calculated)

Figure 7 shows that the predicted storage moduli of the material under all the three pre-compression levels and 3 dynamic strain amplitudes, agrees very well with those obtained from experiments.

As to the loss modulus, the accuracy of the calculated results are more complicated. On one hand, the predicted values are in good agreement with their experimental counterparts for all the three dynamic strain amplitudes when the pre-compression strain is 0.3 or 0.4. When the pre-compression strain is 0.5, clear difference exists between the computed loss moduli and those from experiments with bigger dynamic strain amplitudes, namely 0.06 or 0.1.

Static contact status between the foam cushion and a standard fake stern was measured using a CONFORMat sensor from Tekscan with the setup shown in Figure 8. From the measurement, the equivalent area of the contact surface can be figured out as $0.1305m^2$. It should be mentioned that the fade stern weighs 52.0 kg.

For a specific volunteer and a predefined dynamic excitation, we determine the pre-compression of the cushion and the dynamic strain amplitude by analogy to corresponding tests on the standard fade stern and the same cushion. To this end, one needs to know the weight of the volunteer and the ratio of the weight carried by the seat. The latter parameter is seat-dependent, and is usually a constant

Figure 8 – Test bed for contact status

for a fixed seat. In our tests, the ratio is 83%, and the dynamic excitation is controlled by the acceleration of the platform of the vibrator.

For the following derivation, an acceleration of 2.0m/s^2 was chosen, and after analogy to the chosen volunteer whose model parameters are given in Table 1, the pre-compression strain is determined as 0.5, and the dynamic strain amplitude is 0.1. With these data in mind, one can choose model parameters for the cushion foam by referring to Table 2.

5.2 Apparent Mass Calculation and Validation

Using the model parameters for both the seated human body and the cushion, we can calculate the apparent mass of the system in each frequency within an interesting range according to Equation (13). The amplitudes of the calculated apparent mass are shown in Figure 9 against with their experimental counterparts in a range from 0.0Hz to 20.0Hz. In addition, the amplitude – frequency curve of the volunteer seated on the same seat but without any foam cushion is also presented in the figure, so as to clearly demonstrate the contribution of the cushion.

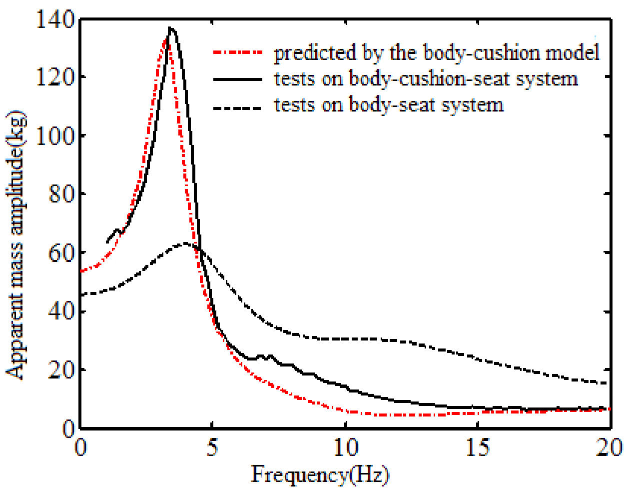


Figure 9 – Comparison of the apparent masses

It can be observed from Figure 9 that the calculated amplitude – frequency curve agrees pretty well with its experimental counterpart up to 5.0Hz. For frequencies higher than 5.0Hz and lower than 15.0Hz, there exists clear difference between the predicted value and measured one. The error should be attributed to evaluation about the equivalent contact area and pre-compression level, and to the error of the foam model in predicting the loss modulus.

On the other hand, Figure 9 shows a great reduction with the peak vibration bored by the volunteer, as the peak apparent mass of the volunteer with the foam cushion is about 2 times more than the peak value without the foam cushion. This is due to reduction with vibration transmission. In addition, due to the cushion introduction, the frequency corresponding to the peak value decreases slightly.

When the exciting acceleration increases, further investigations reveals that the peak-value frequency in the amplitude – frequency curve of the apparent mass decreases, but the variation of the peak value is not monotonous. In our case, when the exciting acceleration changes from 0.5 m/s^2 to 1.0 m/s^2 , and then to 2.0 m/s^2, the peak value of the apparent mass decreases at first and then increase.

6. SUMMARY

This paper presents a systematic approach to understand the interaction between a human body and the seat cushion. After a brief introduction to the vertical vibration characteristics of a seated human body and the mechanical properties of a vehicle seat cushion foam, modeling of a seated human body and the foam are presented in sequence. By combining both models, the vibration equation was derived for a constructed system with a human body, a foam cushion and a rigid seat. An explicit expression was then obtained in frequency domain for the apparent mass of the human body in the system. Calculations and tests are conducted on the model or the system, respectively. The results clearly show that the seat cushion dramatically reduces the vibration transmitted to the human body, and that the established model can reliably capture the first peak value in the amplitude – frequency curve of the apparent mass. The approach and observation are helpful to the analysis of vehicle ride comfort and the design of advanced vehicle seats.

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