Modelling of a train seat with subject exposed to lateral, vertical and roll vibration

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Abstract. Considering the characteristics of ride vibration of a train, a multi-body dynamic model of a double-unit passenger seat exposed to lateral, vertical and roll vibrations was developed and calibrated using measured seat transmissibilities from the acceleration at the floor to the accelerations at both the seat pan and backrest in lateral, vertical and roll directions. With use of an existing seated human model, two coupled models of the double-unit seat with one and two subjects were then developed and calibrated using the corresponding measured seat transmissibilities. The models showed good agreement with the experimental data. Using these models, modal analysis was conducted to find out the modal properties of the human-seat system and facilitate discussions in relation to the resonances in the seat transmissibilities. It was found the primary peak in the vertical transmissibility of the seat pan may arise from the mode of human body. Two vibration modes around 15 and 30 Hz of the seat contributed to the peaks with approximate frequencies in the seat transmissibilities.

1 Introduction
The transmissibility from the seat base to the human-seat interface is one of the most important characteristics of the seats for assessing ride comfort. Since the human body is a complex dynamic system, the transmissibility of a seat with a seated person shows great discrepancy from the counterpart with a rigid mass of the same weight [1]. Experiment is the main approach to studying the seat transmissibility, however, experiment with subjects is time-consuming and expensive, also limited by the vibration magnitude the subjects can endure. Sometimes, dummies can be used to take the place of subjects, but manufacturing such a dummy having the same biodynamic characteristics as human body is not an easy task. The experiment also requires the use of single-axis or multi-axis vibrator that is not always available. As an alternative, modelling approach has been widely used to assist in the experiment to understand the vibration transmission of the seat to occupants.

Various kinds of human-seat models have been developed to study the vibration transmission of human-seat systems under exposure to different excitations. The models can be generally categorized into lumped parameter models [2][3], multi-body dynamic models [4][5] and finite element models [6][7][8].

However, most of the models are so far restricted to single-axis translational vibration. They are usually developed in the symmetric plane of the seat, so the out-of-plane motion cannot be predicted by these models. The multi-body dynamic model is a compromise between the lumped parameter model and finite element model, sharing part of their advantages and avoiding part of their drawbacks. For example, multi-body model is usually anatomically representative, even if it is not as representative as finite element model; it has an advantage of being less computationally expensive than the finite element model. What is more, most
of the existing seat models were usually developed for single-unit car seats, which have quite a different structure from double-unit train seat. Therefore, taking into account the characteristics of ride vibration of a train, a kind of human-seat multi-body dynamic models that have the ability of truly reflecting double-unit train seat structure and vibration characteristics under exposure to combined in-plane vertical and out-of-plane lateral and roll vibrations is waiting to be developed.

2 Introduction of experiment

The experiment can be divided into two sections, the first is about measurement of the biodynamic response of seated human body (Figure 1(a)), and the second is about measurement of train seat transmissibility (Figure 1(b)). The experiment was carried out using a 6-axis motion simulator in the ISVR at the University of Southampton.

In the first section, the seat was rigid, having a vertical backrest. The excitation in every direction was random acceleration signal with approximately flat constant-bandwidth acceleration spectra defined in the range from 0.5 to 30 Hz. Every participant was exposed to 27 tri-axial excitations, which was the combination of the vertical excitation with three magnitudes (0.25, 0.5 and 1 m/s² r.m.s.), lateral excitation with three magnitudes (0.25, 0.5 and 1 m/s² r.m.s.), as well as roll excitation with three magnitudes (0.5, 0.75 and 1 rad/s² r.m.s.). The multi-axis excitations were almost incoherent between different directions. The rotational axis of roll excitation is defined as the intersection line between the seat pan upper surface and the symmetric (x-z) plane of the seat. During the experiment, the participants were asked to sit in an upright posture in contact with the backrest, with the feet resting on the footrest (average thigh contact). The accelerations and forces on the seat pan and backrest can be measured by SIT-pads and force transducers, respectively. The apparent mass that is the frequency response function from the acceleration to the force can be calculated.

In the second section, every participant was exposed to 27 tri-axial excitations with approximately flat constant-bandwidth acceleration spectra, which was also the combination of the vertical excitation with three magnitudes (0.25, 0.5 and 1 m/s² r.m.s.), lateral excitation with three magnitudes (0.25, 0.5 and 1 m/s² r.m.s.), as well as roll excitation with three magnitudes (0.5, 0.75 and 1 rad/s² r.m.s.). However, the rotational
axis was the intersection line between the symmetrical plane of the left seat and the platform (‘left’ is from the perspective of the seated subject). The excitation in every direction was random signal defined in the range from 0.5 to 50 Hz and almost incoherent between directions. The accelerations on the platform, seat pan and backrest of the left unit were measured with no subject, with one subject sitting on the left unit, and with two subjects. The participants were asked to sit in an upright posture in contact with the backrest, with the feet resting on a footrest (average thigh contact). The seat transmissibility that is the frequency response function from the acceleration on the platform to the one on the seat pan or backrest can be calculated.

3 Development of double-unit train seat model

3.1 Model description

The train seat has two units: the left one and the right one (‘left’ and ‘right’ are defined from the perspective of the seated subject). The train seat consists of seven parts, that is, the left and right backrests (\( B^l, B^r \)), the left and right seat pan frames (\( B^s, B_s \)), the left and right cushions (\( B^c, B^c \)), the seat base (\( B_b \)) (Figure 2). There are seven different coordinate systems defined in the model, one is the absolute coordinate system \((n_x, n_y, n_z)\), whose origin is at the centre of the intersection line of the cushion upper surface and backrest front surface of the left unit. \( n_x \) and \( n_y \) that are in the seat pan surface are directed along the fore-and-aft and lateral directions, respectively, while \( n_z \) is normal to the surface. The other six are relative coordinate systems \((n_x^i, n_y^i, n_z^i)\) (the left superscript \( j \) represents in the left or right unit, but there is no superscript \( j \) for \( i_B \)) with their origins defined at the centre of gravity of each part of the seat \( O_i \) (\( i = 5, 6, 9; j = l, r, no \)). The left and right backrests can respectively have inclinations with angles of \( \alpha \), \( \alpha \) relative to the vertical plane. The excitation is expressed as lateral excitation \((y_0)\) along \( n_y \), vertical excitation \((z_0)\) along \( n_z \) and roll excitation \((\theta_0)\) around the intersection line between the symmetric \((z-x)\) plane of the left unit of the seat and the platform surface.

The model has a total of 21 degrees-of-freedom: each part has translational motions in \( n_x \) and \( n_y \) directions, \( y_i \) and \( z_i \) (\( i = 5, 6, 9; j = l, r, no \)), and roll motions, \( \theta_i \) (\( i = 5, 6, 9; j = l, r, no \)), around the x-axis in their own relative coordinate systems \((n_{iB})\).
Figure 2 Double-unit train seat model

Pairs of contact points between the seat pan cushion ($i^B_5$) and the seat pan frame ($i^B_6$) are expressed as $i^C_6$ on the frame side and $i^C_6$ on the cushion side ($j = l, r$). The contact points between the left or right side of the seat base ($B_B$) and the platform are expressed as $i^C_7$ or $i^C_7$ on the seat base and $i^C_7$ or $i^C_7$ on the platform. Each pair of points is coincident at the equilibrium position but at different positions during vibration.

For the connection of the seat parts, backrest $iB_3$ and seat base $B_B$ are connected with seat pan frame $i^B_6$ at points $iD_4$ and $D_5$, respectively ($j = l, r$). The left seat pan frame $iB_5$ and the right $iB_9$ are connected at the point $D_6$. The connections and contacts are modelled by translational springs and dampers in y and z directions, and rotational springs and dampers around x direction as well.

The seat model is developed by calculating the relative motions and forces at the contact and connection points, and then establishing the motion equations for every part of the seat according to Newton’s second law. Since the model has geometrical nonlinearity, the motion equations are linearized assuming the roll angle for every part is small. Therefore, the inline transmissibilities from the inputs to the lateral, vertical and roll motions of the left seat pan and left backrest can be calculated by this model.

In the model, some parameters can be determined by measurement, such as the dimensions of the seat, the masses of the seat parts, etc. Other parameters including all the stiffness and damping at the contact and connection points, some parameters in relation to contact or connection positions are determined by model calibration using the experimental data.
The model was calibrated by minimizing the error between the lateral, vertical and \( r_x \)-axis inline transmissibilities at the left backrest and left seat pan measured in the experiment and those predicted by the model.

Reasonable constraints for the lower and upper bounds of all the parameters to be calibrated were given before model calibration. The model calibration was completed with a combined genetic algorithm and minimization function of constrained nonlinear multivariable problem (‘fmincon’ function) in Matlab. The frequency range considered for the calibration is 0.5-50 Hz. After that, all parameters of the model were determined.

3.2 Results
A good agreement between the experiment and model under different excitations was observed. Figure 3 showed an example of the comparison between the experimental data and model prediction, proving the effectiveness of the seat model. The model is also good at predicting the resonances of the transmissibilities at both the left seat pan and left backrest. The seat model may also be applicable for different backrest inclinations that have also been taken into account in the modelling.

![Figure 3](image_url)  
**Figure 3** The comparison between model and experiment for y-direction, z-direction and \( r_x \)-direction seat transmissibility at the left seat pan and left backrest under the excitation of 0.5 ms\(^{-2}\) r.m.s. lateral, 1.0 ms\(^{-2}\) r.m.s. vertical and 0.75 rad/m\(^2\) r.m.s. roll vibration.

The modal analysis with the model was further conducted to find out the relationship between the modal properties and the resonances in the transmissibilities according to the complex mode theory.

Because of the phase difference between different degrees of freedom, the modal shape for every modal frequency is better to be plotted by animation. Taking the excitation in Figure 3 as an example, two main modes were detected, 15.49 and 26.99 Hz, respectively, corresponding to the peaks in black circles and in blue circles. For the first modal shape in Figure 4(a), the modal shape is dominated by the roll and lateral motions of the seat pan and backrest, moving in phase for the left and right units. For the second modal shape in Figure 4(b), the modal shape is mainly the lateral and roll motions of the two backrests, however, their relative vibrations transform from in-phase vibration to out-of-phase one. The seat model was simplified as a linear model under one specific excitation, so there were different sets of seat parameters corresponding to different excitations. These two modal shapes of the seat under different excitations were almost the same, this was probably because the nonlinearity of the seat was not so significant.
Figure 4 The modal shapes of the seat under the excitation of 0.5 ms$^{-2}$ r.m.s. lateral, 1.0 ms$^{-2}$ r.m.s. vertical and 0.75 rad/m$^2$ r.m.s. roll vibration (red: deformed modal shape; yellow: undeformed modal shape).

4 Brief introduction of the seated human model

A 16-DOF multibody dynamic model of human body seated on rigid seat subjected to combined vertical, lateral and roll vibration was proposed in a parallel study by the authors [9] (Figure 5). The model is applicable for subjects of different heights and weights under different excitations.

The model was calibrated for 12 different subjects under 27 tri-axial excitations in the frequency range of 0.5-20 Hz. Similar to the seat model, every subject had one set of parameters under one specific excitation.

As an example, Figure 6 showed the experimental and modelling results of apparent masses from one subject (171 cm in height and 83.5 kg in weight) exposed to combined lateral (0.5 ms$^{-2}$ r.m.s.), vertical (1.0 ms$^{-2}$ r.m.s.) and roll (0.75 rad/m$^2$ r.m.s.) vibration. The experimental and the modelling results showed good agreement (Figure 6). Three main frequencies were registered by the modal analysis—1.01, 2.53, and 5.54 Hz, corresponding to the peaks in black, blue and green circles, respectively. For the first mode in Figure 7(a), the modal shape is mainly the roll and lateral motions of the upper torso ($B_7$), together with the motion of the head ($B_4$) in the reverse direction because of inertia effect of the head that is not in contact with the backrest. For the second mode in Figure 7(b), the modal shape is dominated by the lateral motions of the abdomen ($B_1$), pelvis ($B_2$) and thighs ($B_3, B_4$) as well as the roll and lateral motions of the upper torso ($B_7$) and head ($B_4$). For the third mode in Figure 7(c), the modal shape is dominated by the whole upper body vertical motion together with the roll and vertical motions of the pelvis ($B_2$) and vertical motions of the thighs ($B_3, B_4$).
Figure 5 Seated human body model.

Figure 6 The comparison between model and experiment for y-direction and z-direction apparent masses at the seat pan and backrest for a subject of 171 cm and 83.5 kg under the excitation of 0.5 ms$^{-2}$ r.m.s. lateral, 1.0 ms$^{-2}$ r.m.s. vertical and 0.75 rad/s$^2$ r.m.s. roll vibration.
Figure 7 The modal shapes of a subject of 171 cm and 83.5 kg under the excitation of 0.5 ms\(^{-2}\) r.m.s. lateral, 1.0 ms\(^{-2}\) r.m.s. vertical and 0.75 ms\(^{-2}\) r.m.s. roll vibration (red: deformed modal shape; yellow: undeformed modal shape).

5 Development of coupled human-seat models

5.1 Development of double-unit-seat-one-subject model

Assume the mass distribution of the subject seated on the train seat is the same as that seated on the rigid seat. For both the seated human model on the rigid seat and double-unit-seat-one-subject model, the head was not in contact with the seat. Therefore, the modelling of the human body seated on the train seat was the same as the seated human model in Section 4. The development of the double-unit-seat-one-subject model was conducted by integrating the seat model developed in Section 3 with the human body model described in Section 4.

However, for the seated human model on a train seat, the stiffness and damping at the contact points between human body and seat were different from the seated human model on a rigid seat. The former resulted from the interaction between the human tissue and the seat cushion, but the latter arose merely from human tissue. Therefore, these parameters in relation to the stiffness and damping at the contact points between human body and seat and the contact positions need to be redefined or readjusted.

With the above considerations in mind and using the similar modelling approach described above, a coupled model of the double-unit train seat with one subject sitting on the left unit was developed. After linearization, the inline transmissibilities from the inputs to the lateral, vertical and roll vibrations on the left seat pan and left backrest can be calculated by the model.

The parameters to be calibrated were all the contact stiffness and damping between the human body and seat, and some parameters concerning the contact positions. Similar to the seat and human models, this kind of model simplified the object as linear model under a specific excitation, so under different excitations, the model had different sets of parameters. Under a specific excitation, the other parameters except the calibrated parameters were kept the same as the double-unit seat model and seated human model that were determined under the same excitation.

In the same way as the development of the seat model, the double-unit-seat-one-subject model was calibrated in 0.5-50 Hz using the experimental data under the same excitation. Coupling the train seat model in Figure 3 with the human model in Figure 6, good agreement between the model prediction and
experimental results was shown in Figure 8, which proves the effectiveness of the double-unit-seat-one-subject model.

**Figure 8** The comparison between model and experiment for y-direction, z-direction and $r_x$-direction transmissibilities at the left seat pan and left backrest for a subject of 171 cm and 83.5 kg seated on the left of the double-unit train seat under the excitation of 0.5 ms$^{-2}$ r.m.s. lateral, 1.0 ms$^{-2}$ r.m.s. vertical and 0.75 rad/s$^2$ r.m.s. roll vibration.

By means of modal analysis, three main modal frequencies were found, that is, 4.67, 15.27, and 27.73 Hz respectively, corresponding to the black circle, blue circles and green circle in Figure 8. The first modal shape in Figure 9(a) is dominated by the motion of human body that is similar to Figure 7(c), accompanied with vertical and roll motions of the left seat pan. For the second modal shape in Figure 9(b), the motion is mainly the seat vibration that is analogue to Figure 4(a), together with a little lateral and roll motions of the human body. The dominating motion in the third modal shape depicted in Figure 9(c) is the seat vibration that resembles Figure 4(b), along with a little lateral and roll motions of the human body.

**Figure 9** The modal shapes of the double-unit-seat-one-subject model with a subject of 171 cm and 83.5 kg seated on the left under the excitation of 0.5 ms$^{-2}$ r.m.s. lateral, 1.0 ms$^{-2}$ r.m.s. vertical and 0.75 rad/m$^2$ r.m.s. roll vibration (red: deformed modal shape; yellow: undeformed modal shape).
However, without seated subject, the vertical seat transmissibility to the left seat pan had no such peak around 5 Hz (Figure 3). This means the primary peak in the vertical transmissibility of the seat pan may arise from the human body. The human body has attenuating effect on the two seat modes in Figure 4 because the peaks now become flatter and shorter.

5.2 Development of double-unit-seat-two-subject model
Similarly, a coupled model of the double-unit seat with two subjects was developed. After linearization, the inline transmissibilities from the inputs to the lateral, vertical and roll vibrations on the left seat pan and left backrest were also calculated by the model.

![Figure 10](image1.png)

*Figure 10* The comparison between model and experiment for y-direction, z-direction and \( r_z \)-direction transmissibilities at the left seat pan and left backrest for a subject of 171 cm and 83.5 kg seated on the left of the double-unit train seat and a subject of 183 cm and 85 kg on the right under the excitation of 0.5 ms\(^{-2}\) r.m.s. lateral, 1.0 ms\(^{-2}\) r.m.s. vertical and 0.75 rad/m\(^2\) r.m.s. roll vibration.

![Figure 11](image2.png)

*Figure 11* The modal shapes of the double-unit-seat-two-subject model with a subject of 171 cm and 83.5 kg seated on the left and a subject of 183 cm and 85 kg on the right under the excitation of 0.5 ms\(^{-2}\) r.m.s. lateral, 1.0 ms\(^{-2}\) r.m.s. vertical and 0.75 rad/m\(^2\) r.m.s. roll vibration (red: deformed modal shape; yellow: undeformed modal shape).
The parameters to be calibrated were all the contact stiffness and damping between the subject on the right and the seat, and some parameters concerning the contact positions. For the above-mentioned reason, under a specific excitation, the other parameters were kept the same as the double-unit-seat-one-subject model and the human model for the right subject that were determined under the same excitation.

In the same way, the model could be calibrated in 0.5-50 Hz using the experiment data under the same excitation. Coupling the double-unit-seat-one-subject model in Figure 8 with another human model for the right subject of 183 cm and 85 kg, Figure 10 illustrated good conformity between the model and experiment.

Three main modes were still observed, with frequencies of 4.67, 17.08 and 28.20 Hz, corresponding to peaks in the black, blue and green circles in Figure 10, respectively. The modal shapes for the three modes were illustrated in Figure 11(a), (b) and (c), respectively, which were analogue to those in Figure 9(a), (b) and (c) with a little motion of the right subject.

6 Model comparison and summary

For the four models—human model, double-unit seat model, double-unit-seat-one-subject model, and double-unit-seat-two-subject model, those modes with close modal frequencies and similar modal shapes for their shared parts were considered to arise from the same mode. The modal frequencies, damping ratios and modal shapes of the four models were compared and summarized in Table 1. It can be seen the primary peak in the vertical transmissibility to the seat pan for the two human-seat models arose from the mode of human body. However, the coupling of the human body with the seat reduced the modal frequency and the damping ratio somehow. For this human mode (mode 1), the axial and shear stiffness and damping of the tissue beneath pelvis seemed to play an important role in the modal frequency and damping ratio [10][11]. When the axial and shear stiffness and damping of the tissue beneath pelvis coupled with the stiffness and damping of the seat cushion in series, the overall stiffness and damping would both decrease, which may be the reason for the reduced modal frequency and damping ratio. What is more, the damping of the seat cushion was much larger than the human tissue, so the reduction of the damping ratio was not so obvious. On the other hand, the peaks around 15 Hz and 30 Hz in two human-seat models were induced by the seat modes—mode 2 and mode 3, respectively. The coupling with the human bodies increased the modal damping obviously, so the peak magnitudes in the seat transmissibilities reduced and the peaks changed from ‘sharp’ to ‘flat’ as the increase of subject number, which was more significant for the peaks around 15 Hz. The modal frequencies usually rose more or less as the increase of subject number except the modal frequency of mode 2 for the double-unit-seat-one-subject model.

### Table 1 Intercomparison among the models.

| Models                     | Human model for left subject | Seat model | Double-unit-seat-one-subject model | Double-unit-seat-two-subject model |
|----------------------------|------------------------------|------------|------------------------------------|------------------------------------|
| Mode 1                     | Frequency(Hz)               | 5.54       | 4.67                               | 4.67                               |
|                            | Damping ratio               | 0.28       | 0.24                               | 0.24                               |
|                            | Modal shape                 | Figure 7(c)| Figure 9(a)                        | Figure 11(a)                       |
| Mode 2                     | Frequency(Hz)               | NA         | 17.08                              | 15.27                              |
|                            | Damping ratio               | NA         | 0.038                              | 0.113                              |
|                            | Modal shape                 | NA         | Figure 4(a)                        | Figure 9(b)                        |
| Mode 3                     | Frequency(Hz)               | NA         | 28.20                              | 27.73                              |
|                            | Damping ratio               | NA         | 0.057                              |                                    |
|                            | Modal shape                 | NA         | Figure 4(b)                        | Figure 9(c)                        |

*NA means ‘not available’.
7 Conclusion
The transmissibilities predicted with the seat model, double-unit-seat-one-subject model, and double-unit-seat-two-subject model showed good consistency with the transmissibilities measured in lab experiment. These models were applicable for dynamic analysis of train seat with occupants under different magnitudes of combined lateral, vertical and roll vibrations. The models may also be suitable for different backrest inclinations that were also taken into account in the modelling of human body and seat. Modal analysis revealed that two modes around 15 and 30 Hz of the seat contributed to the peaks with approximate frequencies in the seat transmissibilities. The former vibration was dominated by the lateral and roll motions of two seat pans and backrests moving in phase. The latter one was mainly the lateral and roll motions of the two backrests moving in phase one minute and out of phase the next. The seated subjects exhibited a tendency of increasing the damping of these two modes, so the peaks of the transmissibilities corresponding to these two modes got reduced and flatter. The primary peak around 5 Hz in the vertical transmissibility to the seat pan arose from a vertical vibration mode of the human body with a little higher modal frequency. The human-seat models are going to be adopted for the prediction of the perceived vibration by the passengers on a train in the future study.

8 Reference
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