Journal of Sound and Vibration (1998) **214**(1), 121–137

THE PREDICTION OF SEAT TRANSMISSIBILITY FROM MEASURES OF SEAT IMPEDANCE

L. WEI AND J. GRIFFIN

Human Factors Research Unit, Institute of Sound and Vibration Research, University of Southampton, Southampton S017 1BJ, England

(*Received* 6 *February* 1997, *and in final form* 5 *February* 1998)

A method of predicting seat transmissibility from mathematical models of the seat and the human body is described. The complex dynamic stiffness of a seat is determined by measurement using an indenter rig, and its stiffness and damping subsequently determined by curve-fitting. By using the fitted stiffness and damping of the seat model, and a previously determined dynamic model of the human body, the seat transmissibility is predicted mathematically. The method is illustrated with data obtained with a car seat and also a rectangular sample of foam. The seat and foam transmissibilities were predicted over the frequency range 1·25–25 Hz using two alternative models of the human body (a one-degree-of-freedom model and a two-degree-of-freedom model). The predicted seat transmissibilities were close to those measured in a group of eight subjects over the entire frequency range. The two-degree-of-freedom model of the human body provided better predictions where the seat and foam showed a second resonance around 8 Hz. The need for a non-linear mathematical model of the human body and a non-linear seat model is discussed.

1. INTRODUCTION

In a wide variety of transport systems, vibration is transmitted to drivers, pilots, crew or passengers through seats. Most seats are compliant and modify the vibration by amplifying low frequencies and attenuating high frequencies. One objective of the seat designer is to achieve an overall reduction in vibration discomfort of the seat occupant compared with the discomfort that would be experienced with a rigid seat [1].

Various empirical methods for quantifying the dynamic performance of a seat and its overall isolation efficiency are available. However, these mostly involve the measurement of seat transmissibility with subjects sitting in the seat. This is because the transmission of vibration through a seat is dependent on the mechanical impedance of the body supported on the seat: the seat and the body act as a coupled dynamic system. Tests with subjects are time-consuming when conducted in field studies. Laboratory motion simulators can be used, but suitable simulators are expensive to operate and there are inherent risks to exposed persons giving rise to the need for a range of medical and ethical precautions [2]. It would be desirable to predict the dynamic performance of a seat without exposing subjects to vibration.

One method of predicting seat dynamic performance without using human subjects is to replace the human body with a dummy having the appropriate mechanical impedance (see, e.g., references [3–5]). Although there are some very useful applications for such anthropodynamic dummies there are also difficulties. It is currently difficult to achieve the correct impedance and it may be expected to be difficult to check that the dummy is within

0022–460X/98/260121 + 17 $25.00/0/sv981540

calibration before use. Passive dummies will tend to exhibit friction and have a non-linearity unlike that of the human body. The mechanical impedance of the body is non-linear and it will be difficult to match this non-linearity with a passive mechanical system.

An alternative method of predicting seat dynamic performance without using human subjects is to use a mathematical prediction based on separate measurements of the impedance of the seat and the impedance of the human body. Fairley and Griffin [6] showed that the method can give useful predictions but the method has not been developed subsequently.

A further advantage of the mathematical method of prediction is that it encourages the development of a better understanding of the dynamic performance of seat components (e.g., suspensions, foams, covers). Eventually, with a full understanding of the role and the dynamic performance of each seat component it may be possible to predict seat dynamic performance from the physical and chemical construction of the various seat parts. By these means a mathematical model could be used to identify the desired dynamic properties of a seat and the method of achieving this performance could also be predicted. For example the required mix of foam ingredients might be predicted.

The prediction of seat transmissibility requires knowledge of the mechanical impedance of the seated human body. A separate study has recently evolved two optimum models of the impedance of the seated body: a one degree-of-freedom model and a two-degree-of-freedom model [7]. The present study compares the prediction of seat transmissibility using these two models.

The purpose of the present study is to assess the accuracy with which seat transmissibility can be predicted from the proposed models of human mechanical impedance and alternative measurements of seat mechanical impedance. The method has been applied to a complete car seat and to a rectangular sample of foam.

2. EXPERIMENTAL MEASUREMENTS

The experiments were conducted separately with a car seat and with a rectangular sample of foam. The car seat was the driving seat from a modern small family car. It was constructed from a steel frame with moulded foam supported from beneath by a contoured steel seat pan and fully encased within a cover. The TDI foam in the seat had a density of 50 kg m^3. The rectangular sample of foam was 500 mm wide by 420 mm deep and 120 mm thick. It is described as a "soft feeling type" polyurethane foam used for car seating. It had a density of $\approx$40 kg m^3 and a hardness of about 7·0 kPa.

Three types of measurement were undertaken. The mechanical impedance of both the seat and the foam were measured and their transmissibilities determined by using both inert objects on the seat and a group of human subjects.

2.1. MEASUREMENT OF SEAT MECHANICAL IMPEDANCE WITH AN INDENTER

Consider a seat which is supported on a vibrator with its upper surface deflected by an indenter attached to a Kistler 9321A force transducer (see Figure 1). The indenter, having the shape of a SIT-BAR [8], was screwed down until the required force on the seat was reached and then locked in position. An Entran piezoresistive accelerometer (type EGCSY-240*-10) was mounted on the vibrator platform beneath the seat. The force on the indenter and the acceleration at the base of the seat were measured during a 100-s period of random vibration (0·5 ms^{-2} r.m.s.) produced by the electrodynamic vibrator. The vibration had a flat acceleration power spectral density over the range 1·0–30 Hz ($\pm$10%).

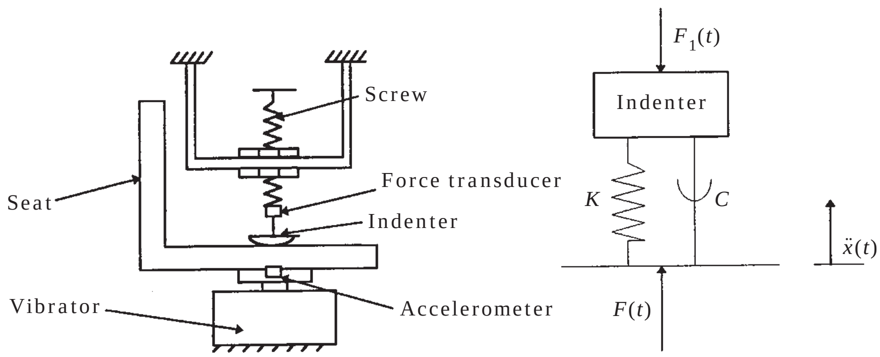

Figure 1. Using the indenter to load the seat and the foam.

The measurements were obtained with each of six pre-loads (300–800 N) applied to the surface of the seat and the foam sample. Signals from the force transducer and the accelerometer were signal conditioned and acquired at 100 samples per second into an *HVLab* system.

2.2. MEASUREMENT OF SEAT TRANSMISSIBILITY USING A SANDBAG

The transmissibilities of the seat and the foam were measured while they supported a sandbag assumed to be a rigid mass (see Figure 2). Five different masses of sandbag (30–70 kg) were used while the base of the seat was excited by a 100-s period of random vibration at $0{\cdot}5\ \mathrm{ms}^{-2}$ r.m.s. The transmissibility was calculated from the acceleration measured beneath the seat (or foam) and the acceleration measured between the sandbag and the surface of the seat (or foam).

2.3. MEASUREMENTS OF SEAT TRANSMISSIBILITY WITH A RIGID MASS

The transmissibilities of the seat and the foam were also measured while they supported each of two rigid masses 22 mm wide by 14 mm deep by 15 mm (or 30 mm) thick (as for the sandbag in Figure 2). The weights of the rigid masses were ≈ 30 and 50 kg. The seat was again excited by a 100-s period of random vibration at $0{\cdot}5\ \mathrm{ms}^{-2}$ r.m.s. Two accelerometers were used for these measurements, mounted at the same place on the seat (or foam) as described above.

2.4. MEASUREMENT OF SEAT TRANSMISSIBILITY USING HUMAN SUBJECTS

The transmissibilities of the seat and the foam were measured while they supported eight male subjects (mean age 35 years; mean mass 64 kg). Again, the base of the seat was excited by a 100-s period of random vibration at $0{\cdot}5\ \mathrm{ms}^{-2}$ r.m.s. (see Figure 3). The vibration at

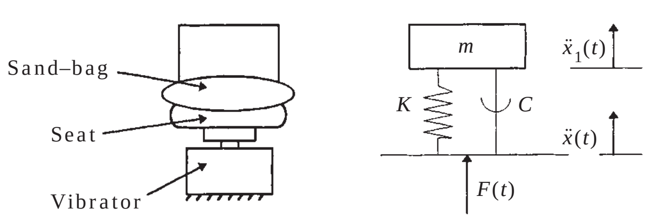

Figure 2. Sandbag as the load on the seat.

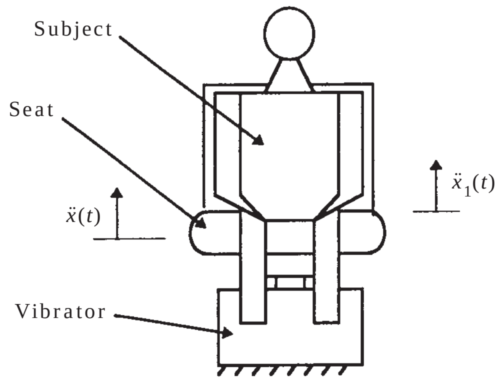

Figure 3. Seat loaded with a person.

the subject–seat interface was measured by using an Entran piezoresistive accelerometer (type EGCSY-240*-10) in an SAE pad (see reference [9]).

3. THEORY AND RESULTS

3.1. INDENTER

When using the indenter to load the seat, the response of the seat and foam system is given by

$$F_1(t) = C\dot{x} + Kx, \tag{1}$$

where x is the displacement (in this experiment the displacement was less than ± 4 mm), $\dot{x}$ is the velocity and $F_1(t)$ is the force measured by the indenter. From this equation the complex ratio of force to displacement is given by:

$$Z(\omega) = F(\omega \mathrm{i})/x(\omega \mathrm{i}) = K + C\omega \mathrm{i}. \tag{2}$$

The ratio of the force to the displacement, $Z(\omega)$, is called the dynamic stiffness, a complex quantity. Dynamic stiffness was used in preference to the mechanical impedance, the ratio of the force to the velocity, because by using the dynamic stiffness the equivalent stiffness K, and the equivalent damping C, are more easily seen.

A curve fitting method was used to obtain seat parameters K and C (i.e., the effective stiffness and damping) from the real and imaginary components of $Z(\omega)$. The least square error method with an optimization algorithm was utilized [10]. The parameters in the above equation were refined to minimize the function

$$error = \frac{1}{N}\sum_{i=1}^{N} (Z_f(i) - Z(i))^2, \tag{3}$$

where $Z_f(i)$ is the corresponding dynamic stiffness from the curve fit at the ith frequency point and $Z(i)$ is the dynamic stiffness in the measured data. With values for the parameters chosen at random used as starting values, the parameters were varied systematically by using the optimization algorithm. The measured and calculated values of the modulus of the dynamic stiffnesses ($\sqrt{K^2 + (C\omega)^2}$) of the foam and the seat over the range of pre-load conditions are presented in Figures 4 and 5. The calculated values of stiffness and damping are tabulated in Table 1.

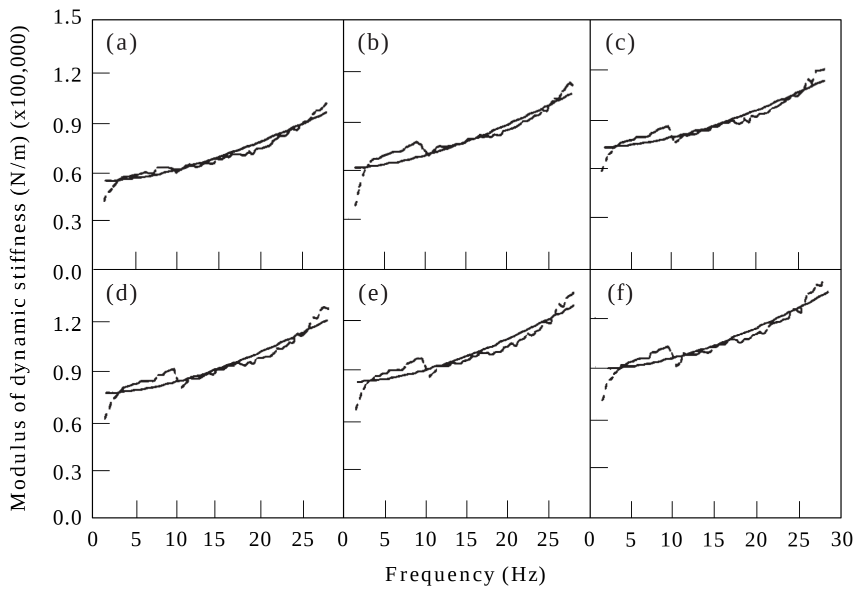

Figure 4. Dynamic stiffness of foam block and fitted model for pre-load forces (a) 300 N, (b) 400 N, (c) 500 N, (d) 600 N, (e) 700 N, (f) 800 N. - - - , Measured values; ——, fitted values.

The stiffness and the damping of both the seat and the foam changed with variations in the pre-load (see Figures 6 and 7). The measurements with the indenter show that when the pre-load increased, the stiffness and the damping of both the seat and the foam increased.

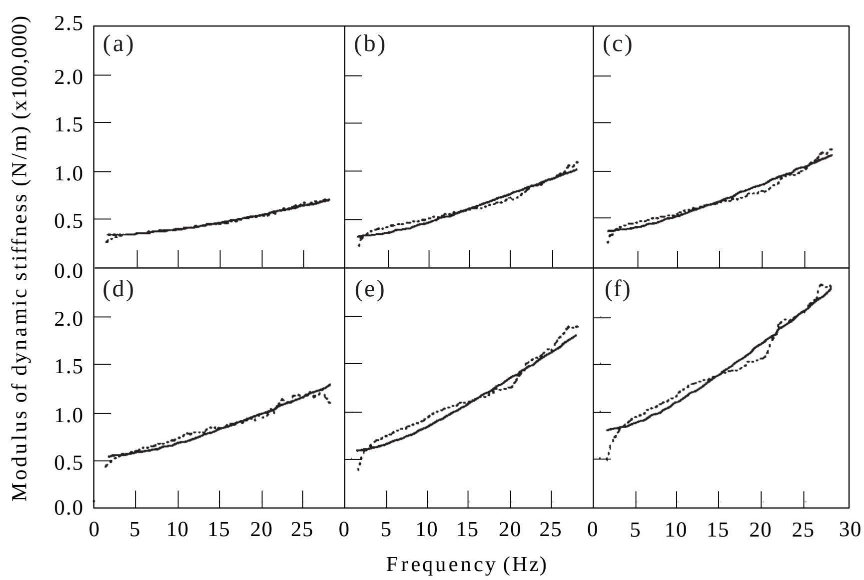

Figure 5. Dynamic stiffness of seat and fitted model for pre-load forces (a) 300 N, (b) 400 N, (c) 500 N, (d) 600 N, (e) 700 N, (f) 800 N. - - - , Measured values; ——, fitted values.

TABLE 1

The stiffness and damping coefficient of the seat and the foam with different pre-loads

	Seat					
	Indenter		Sandbag		Mass	
	K (N/m)	C (Ns/m)	K (N/m)	C (Ns/m)	K (N/m)	C (Ns/m)
300 N	42 300	260	38 471	1323	35 786	204
400 N	44 121	270	57 426	1345		
500 N	50 210	276	54 327	1364	47 481	301
600 N	59 300	280	67 838	1475		
700 N	68 000	285	64 782	1357		
800 N	73 000	293				
	Foam					
	Indenter		Sandbag		Mass	
	K (N/m)	C (Ns/m)	K (N/m)	C (Ns/m)	K (N/m)	C (Ns/m)
300 N	21 167	354	30 381	870	18 576	235
400 N	23 904	457	37 643	868		
500 N	25 082	515	35 787	777	23 187	492
600 N	34 903	570	41 062	681		
700 N	42 340	740	39 186	570		
800 N	54 363	831				

3.2. SANDBAG

With the sandbag used as a load on the seat, the response of seat and foam system is given by

$$m\ddot{x}_1 = F(t), \qquad m\ddot{x}_1 + C(\dot{x}_1 - \dot{x}) + K(x_1 - x) = 0. \tag{4, 5}$$

The seat transmissibility is then

$$T(\omega) = \ddot{x}_1(\omega)/\ddot{x}(\omega) = K + C\omega\mathrm{i}/(K - m\omega^2 + C\omega\mathrm{i}). \tag{6}$$

Again, the seat and foam parameters K and C were obtained by using curve fitting. Figure 8 compares the measured transmissibilities and those predicted from the fitted values of K and C. Table 1 lists the values of K and C obtained with the five different masses of sandbag. Figure 8 shows that using the sandbag instead of the human-body gave a transmissibility curve unlike that with a subject; especially the transmissibility at resonance is much greater. For the measurements reported here, the resonance frequency was appreciably higher with the sandbag than with the human subjects.

Again, the stiffness and damping of the seat and foam changed with variations in the pre-load (see Figures 6 and 7). The stiffness values were similar to those obtained with the indenter, but the indenter seemed to provide the more consistent values. The damping coefficients were very different for the two methods, especially at low pre-load forces where a much higher damping was indicated from measurements obtained with the sandbag. The difference may possibly have arisen because the sandbag had a larger contact area than the indenter. The inconsistent effects of increased load may have arisen because increases in mass of the sandbag resulted in increased size of the sandbag. Figure 8 shows only the

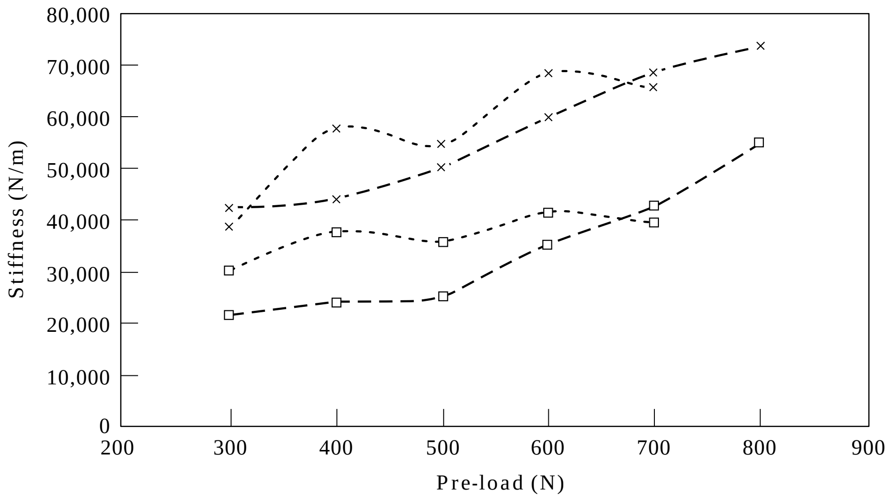

Figure 6. The stiffness of the foam and seat with different pre-loads. – – □ – –, Foam using indenter as load; - - - □ - - - , foam using sandbag as load; – – × – –, seat using indenter as load; - - - × - - - , seat using sandbag as load.

transmissibilities for the seat–sandbag system; the transmissibilities for the foam-sandbag system were similar, but the resonance frequency was lower and the transmissibility at resonance was higher. When the load on the seat (or the foam) increased, the resonance frequency decreased and the transmissibility at resonance increased.

3.3. MASS

The procedure used with the sandbag was also followed with use of the data obtained with the two rigid masses. This provided the stiffness and damping of the seat for loads of 300 and 500 N (see Table 1). The transmissibilities obtained with rigid masses were the same as those with the sandbag, except that the transmissibilities at resonance were higher.

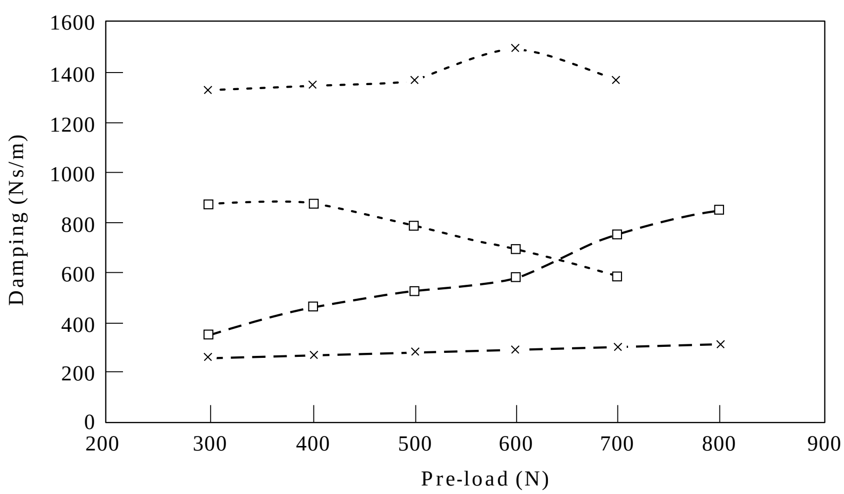

Figure 7. The damping coefficient of the foam and seat with different pre-loads. – –□ – –, Foam using indenter as load; - - - □ - - - , foam using sandbag as load; – – × – –, seat using indenter as load; - - - × - - , seat using sandbag as load.

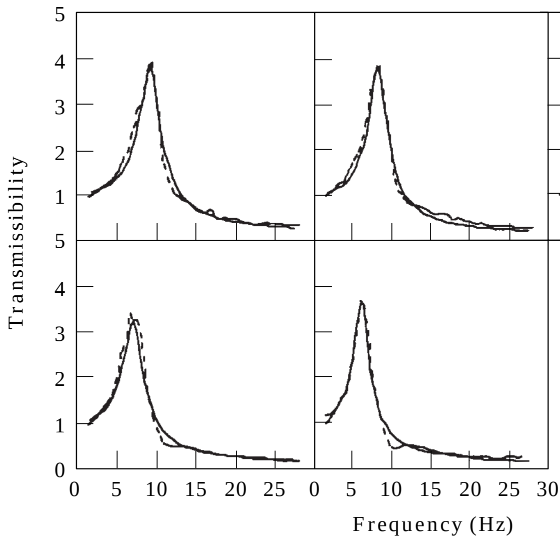

Figure 8. The seat transmissibility using sandbag as the load from 300–700 N and values from fitted model. - - - -, Measured values; ——, fitted values.

Comparing the seat and foam experiments, showed that the foam gave a slightly lower resonance frequency and a slightly lower transmissibility at resonance.

3.4. HUMAN SUBJECTS

The prediction of seat transmissibility with human subjects was based on the one-degree-of-freedom and two-degree-of-freedom models developed by Wei and Griffin [7] (from measurements of the apparent masses of 60 subjects obtained by Fairley and Griffin [11]). Four alternative mathematical models were investigated to represent the human body response in the vertical vibration. From the results, two models (a one-degree-of-freedom model and a two-degree-of-freedom model) were chosen. The two-degree-of-freedom model has the same form as the model in ISO 5982 [12] but different masses, stiffnesses and damping. The model parameters were determined from the measured apparent masses (the ratios of the force to the acceleration) by curve fitting (see Table 2). The two seat–person mathematical models are shown in Figures 9 and 10, where K and C represent the seat and foam dynamic characteristics selected from the indenter results in Table 1 appropriate to the subject's weight. The parameters of the two models of the human body are listed in Table 2.

TABLE 2

Parameters of the one-degree-of-freedom model (model 1) and the two-degree-of-freedom model (model 2) of the apparent mass of the body

	K_1 (N/m)	C_1 (Ns/m)	K_2 (N/m)	C_2 (Ns/m)	m (kg)	m_1 (kg)	m_2 (kg)
*Model 1	44 130	1485	–	–	7·8	43·4	–
*Model 2	35 776	761	38 374	458	6·7	33·4	10·7

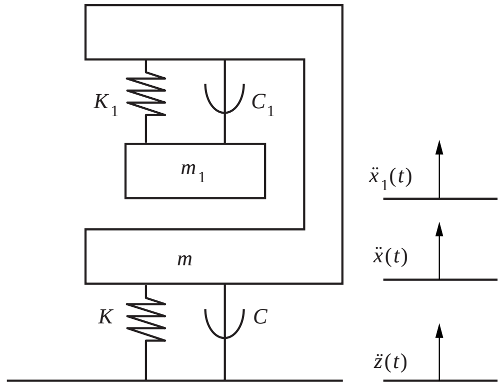

Figure 9. First seat–person system model.

3.4.1. *One-degree-of-freedom model*

The response of the one-degree-of-freedom model (i.e., Figure 9) is given by

$$m_1\ddot{x}_1 + K_1(x_1 - x) + C_1(\dot{x}_1 - \dot{x}) = 0, \qquad m_1\ddot{x}_1 + m\ddot{x} = K(z - x) + C(\dot{z} - \dot{x}). \quad (7, 8)$$

The seat (or foam) transfer function is

$$T(\omega) = \ddot{x}(\omega)/\ddot{z}(\omega) = (A + B\mathrm{i})/(D + E\mathrm{i}). \quad (9)$$

The transmissibility and phase of the seat response are given by

$$|T| = \sqrt{(A^2 + B^2)/(D^2 + E^2)}, \qquad \theta = a\tan(B/A) - a\tan(E/D), \quad (10, 11)$$

where

$$A = KK_1 - (m_1K + CC_1)\omega^2, \qquad B = (C_1K + CK_1)\omega - m_1C\omega^3,$$

$$D = (K - (m + m_1)\omega^2)K_1 + (mm_1\omega^2 - Km_1 - CC_1)\omega^2,$$

$$E = (KC_1 + K_1C - (m_1C + mC_1 + m_1C_1)\omega^2)\omega.$$

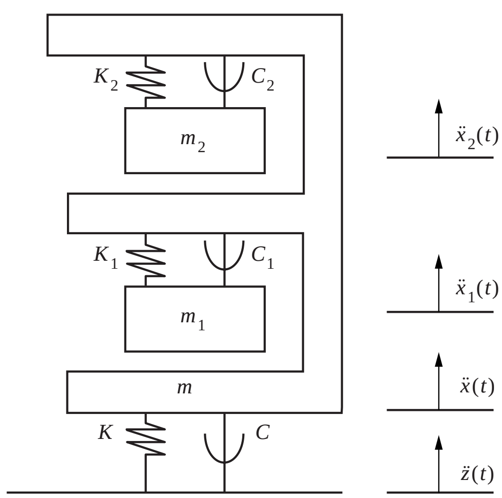

Figure 10. Second seat–person system model.

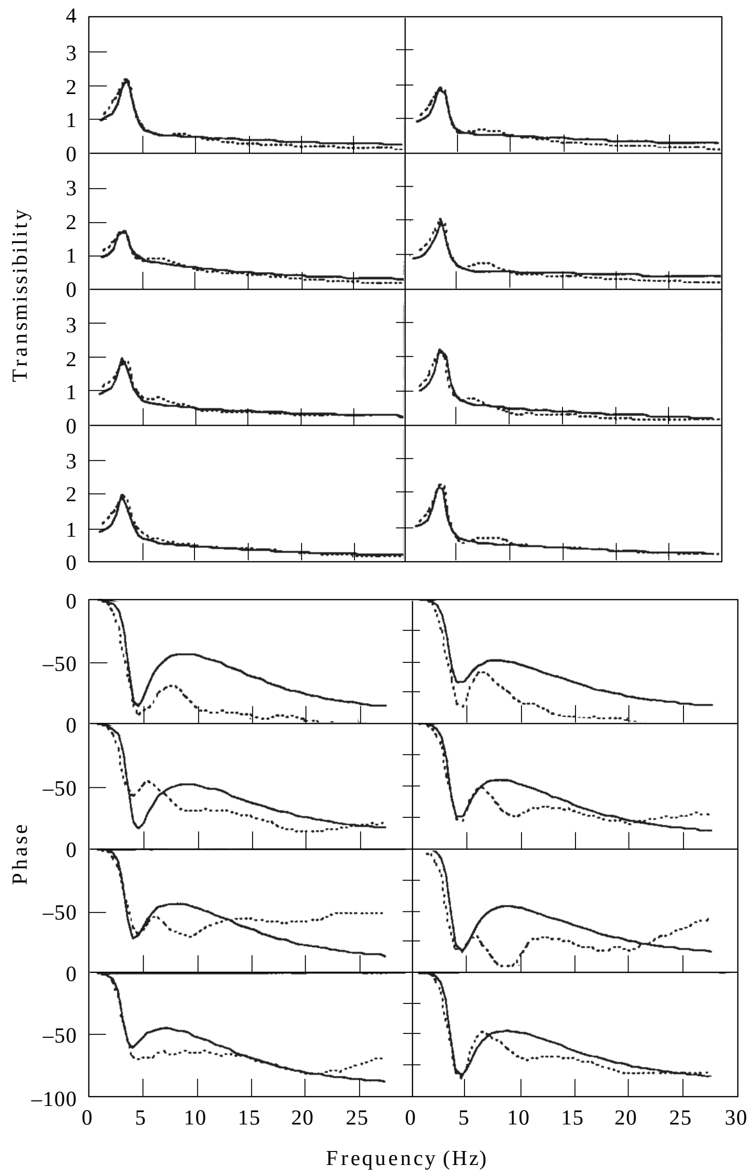

Figure 11. Comparison of measured and predicted foam transmissibility and phase when using single-degree-of-freedom model for eight different male subjects. - - - , Measured transmissibility; —, predicted transmissibility; – – – , measured phase; ——, predicted phase.

The parameters describing the mechanical impedance of the human body and the seat or foam were obtained from experimental data by curve fitting, as described above. However, for this single-degree-of-freedom model, the mass was changed according to the real weight of each subject: the value of $(m + m_1)$ was made equal to the assumed sitting weight of each subject (i.e., 75% of the subject's total weight). The values of K_1 and C_1 were not changed as there was no basis for deciding how these depend on subject mass. Equations (10) and (11) were then employed to predict the seat and foam transmissibility for each subject. The predicted transmissibilities are compared with the transmissibilities measured with the eight subjects seated on the seat and foam in Figures 11 and 12. It can be seen that the transmissibility at resonance and at higher frequencies is generally well predicted by the model. However, the single-degree-of-freedom model fails to predict the second seat resonance apparent at about 7 or 8 Hz for most subjects.

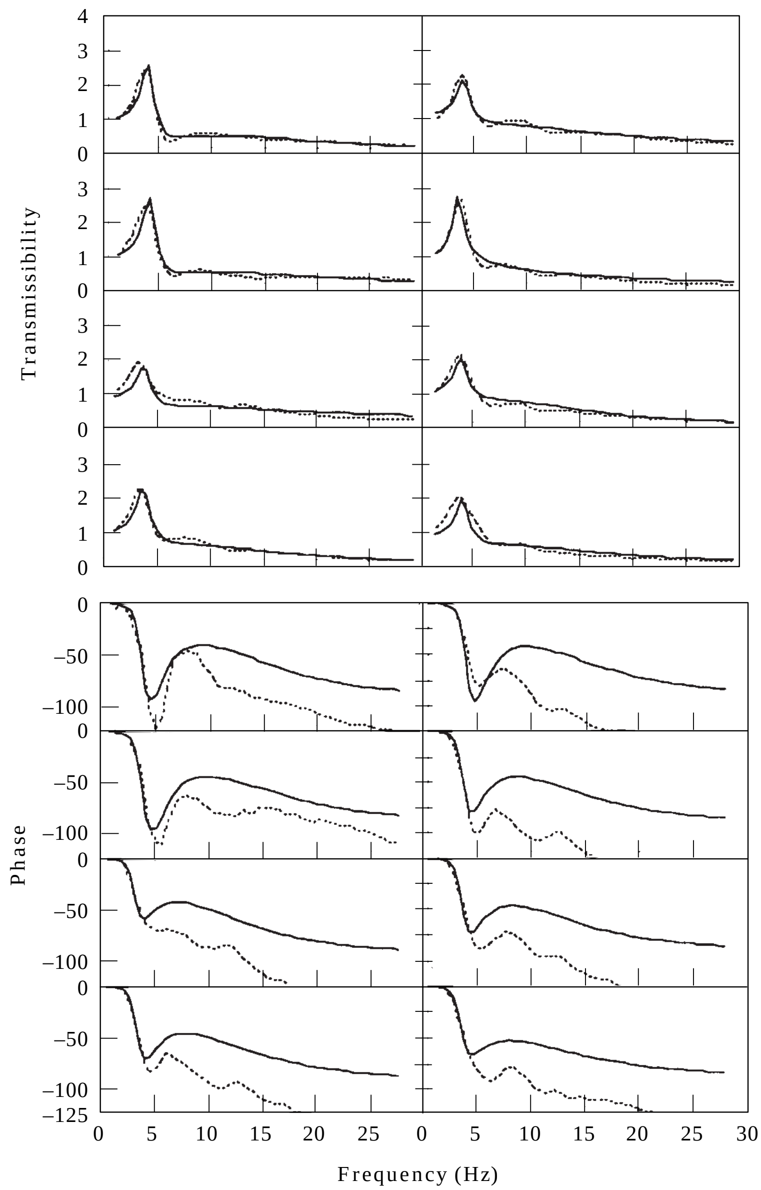

Figure 12. Comparison of measured and predicted seat transmissibility and phase when using single-degree-of-freedom model for eight different male subjects. - - - , Measured transmissibility; —, predicted transmissibility; – – – , measured phase; ——, predicted phase.

3.4.2. *Two-degree-of-freedom model*

The response of the two-degree-of-freedom model of the person combined with the seat is given by

$$m_1\ddot{x}_1 + K_1(x_1 - x) + C_1(\dot{x}_1 - \dot{x}) = 0, \qquad m_2\ddot{x}_2 + K_2(x_2 - x) + C_2(\dot{x}_2 - \dot{x}) = 0, \qquad (12, 13)$$

$$m\ddot{x} + m_1\ddot{x}_1 + m_2\ddot{x}_2 = K(z - x) + C(\dot{z} - \dot{x}). \qquad (14)$$

The seat transfer function is given by:

$$T(\omega) = (F + G\mathrm{i})/\{(H + L) + (M + N)\mathrm{i}\}, \qquad (15)$$

and the seat transmissibility and phase are given by

$$|T| = \sqrt{\frac{F^2 + G^2}{(H + L)^2 + (M + N)^2}}, \qquad \theta = a\tan\frac{G}{F} - a\tan\frac{M + N}{H + L}, \qquad (16, 17)$$

where

$$F = KP_1 - CP_2\omega, \qquad G = KP_2 - CP_1\omega, \qquad H = P_1P_5 - P_2C\omega - m_1K_1P_3\omega^2,$$

$$L = m_1C_1C_2\omega^4 - (m_2K_2P_4\omega^2 - m_2C_1C_2\omega^4),$$

$$M = P_2P_5 + CP_1\omega - (m_1C_1P_3 + m_1C_2k_1)\omega^3,$$

$$N = -(m_2C_2P_4\omega^3 + m_2K_2C_1\omega^3), \qquad P_1 = m_1m_2\omega^4 + K_1K_2 - (m_1K_2 + m_2K_1 + C_1C_2)\omega^2,$$

$$P_2 = (C_1K_2 + C_2K_1)\omega - (m_1C_2 + m_2C_1)\omega^3, \qquad P_3 = K_2 - m_2\omega^2,$$

$$P_4 = K_1 - m_1\omega^2, \qquad P_5 = K - m\omega^2.$$

With this model, the mass was not adjusted to the weight of each subject: the masses of m, m_1 and m_2 were those derived from the previous study (see Table 2 and reference [7]). The predicted gains and phases of the transmissibilities are compared with the measured transmissibilities using human subjects in Figures 13 and 14. It can be seen that the two-degree-of-freedom model of the human body predicted a second resonance in the seat and that in many cases this provides a better fit to the measured seat transmissibility than was obtained with the one-degree-of-freedom model. Although the predicted phase is closer to the measured phase with the two-degree-of-freedom model than with the single-degree-of-freedom model, the prediction is less good than for the prediction of the modulus.

Over the group of subjects as a whole, there was an encouraging correspondence between the measured and predicted values (see Figures 15 and 16). When using the two-degree-of-freedom model for the foam, the mean predicted values fell within the range of the gain and almost within the range of the phase values measured for the eight subjects over the frequency range 1·25–25 Hz. With the seat, the predictions were not so accurate but they still fell within the range of measured values of gain and phase over much of the frequency range.

4. DISCUSSION

The different methods of determining the dynamic characteristics of the seats gave different values (see Table 1). The use of a sandbag to load a seat is probably inappropriate as it must have a large volume and, probably, a greater contact area with a seat than normal subjects. As a result, an excessively large area of the seat (including the edges of the seat) influence the measured dynamic properties.

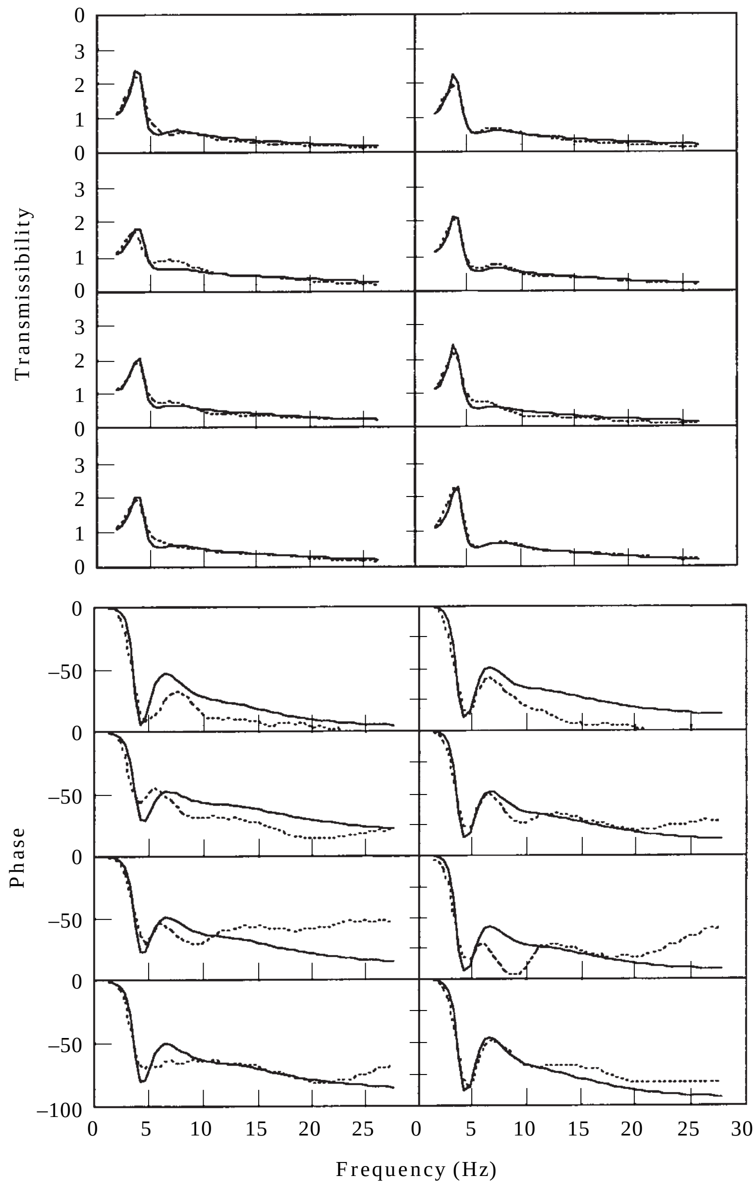

Figure 13. Comparison of measured and predicted foam transmissibility and phase when using two-degree-of-freedom model for eight different male subjects. - - - , Measured transmissibility; —, predicted transmissibility; – – – , measured phase; ——, predicted phase.

The dynamic response of the seat calculated with a rigid mass gave similar results to those obtained with the indenter. However, the indenter is preferable as it provides a more controlled condition: a mass tends to rotate and move when placed on a seat and exposed to vibration.

The dynamic responses of both the human body and some seats are non-linear. These non-linearities result in different seat transmissibilities with different vibration spectra. It will be necessary to quantify the non-linearity of both the human body and the seat material if predictions of seat transmissibility are to be calculated for different vibration magnitudes and different spectra.

From the measurements of the seat and foam stiffness at different pre-loads it can be seen that the stiffness increases appreciably with increases in the load. This may partially explain why measurements of seat transmissibility show only small changes in resonance frequency with subjects of different mass (see, e.g., reference [13]).

The methods shown here appear to allow useful predictions of seat trasmissibility from measurements of the dynamic properties of the seat material. This should allow the selection of optimum materials, and the generation of optimum shapes of materials, so as to maximize the attenuation of vibration to seat occupants. It should be possible to devise a test rig in which the SEAT value (see reference [1]) is produced from

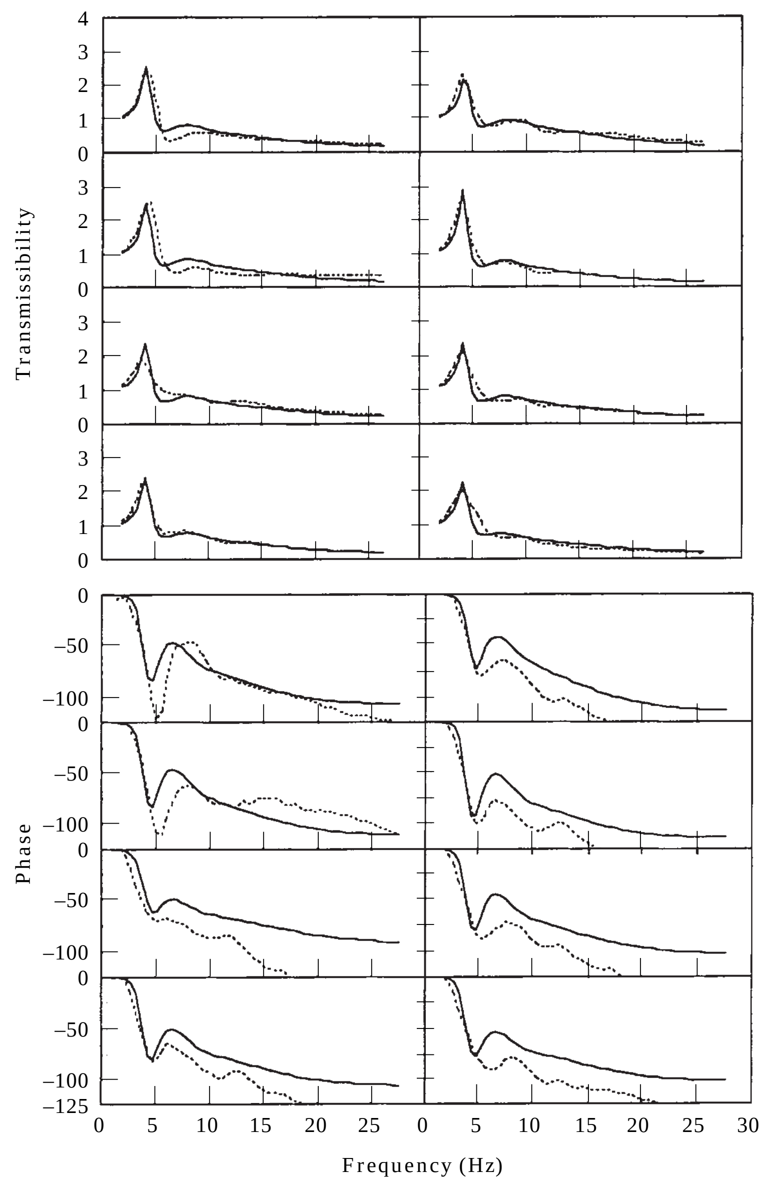

Figure 14. Comparison of measured and predicted seat transmissibility and phase when using two-degree-of-freedom model for eight different male subjects. - - - , Measured transmissibility; —, predicted transmissibility; – – – , measured phase; ——, predicted phase.

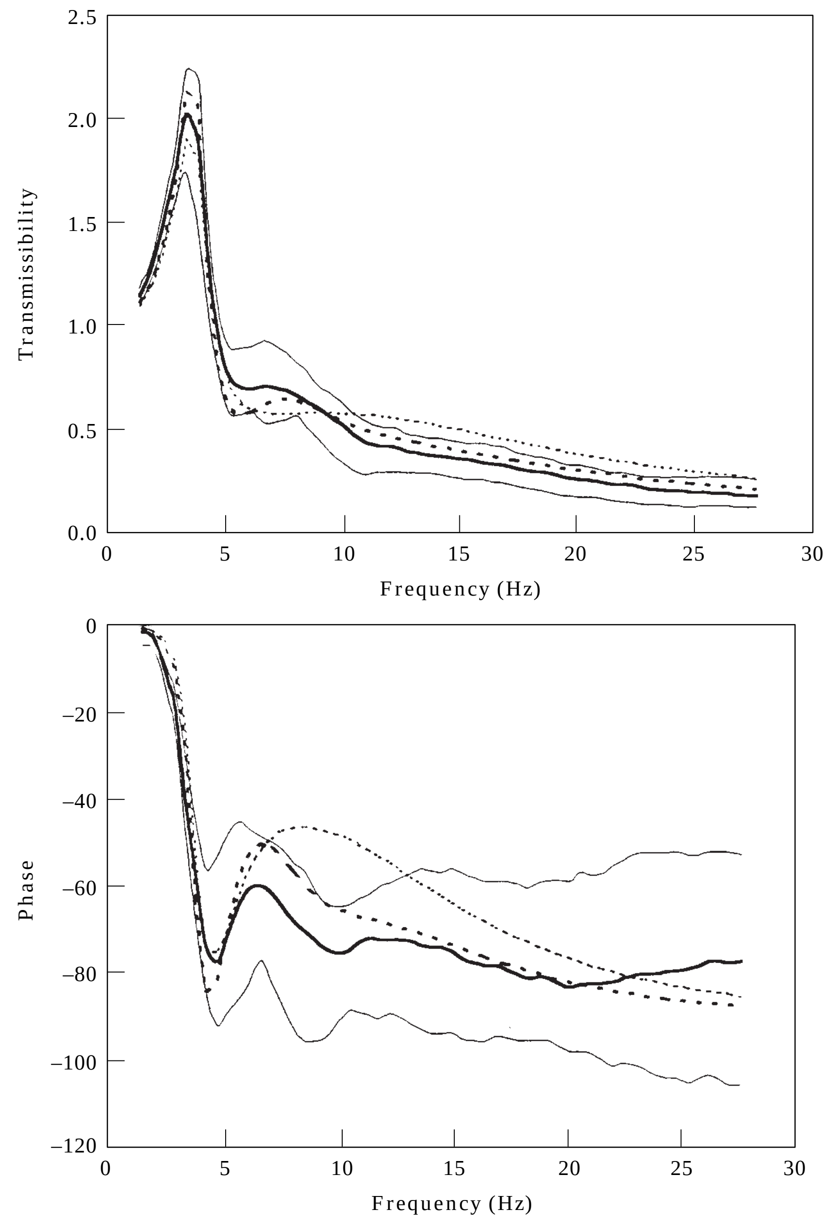

Figure 15. Comparison of measured and predicted foam transmissibility and phase. ——, Mean experimental data; —, range of experimental data; - - - -, single-degree-of-freedom model mean data; — — —, two-degree-of-freedom model mean data.

measurements of the dynamic properties of a material and the known spectrum of vibration in a vehicle.

The method requires some further development to identify the importance of the shape of the indenter. Although the SIT-BAR used here gave good results, it may be necessary to investigate alternative shapes which more closely represent the shape of the human buttocks. There is also a need to consider the influence of seat inclination on measured dynamic stiffness. With advancing understanding of the role of the non-linearity of human

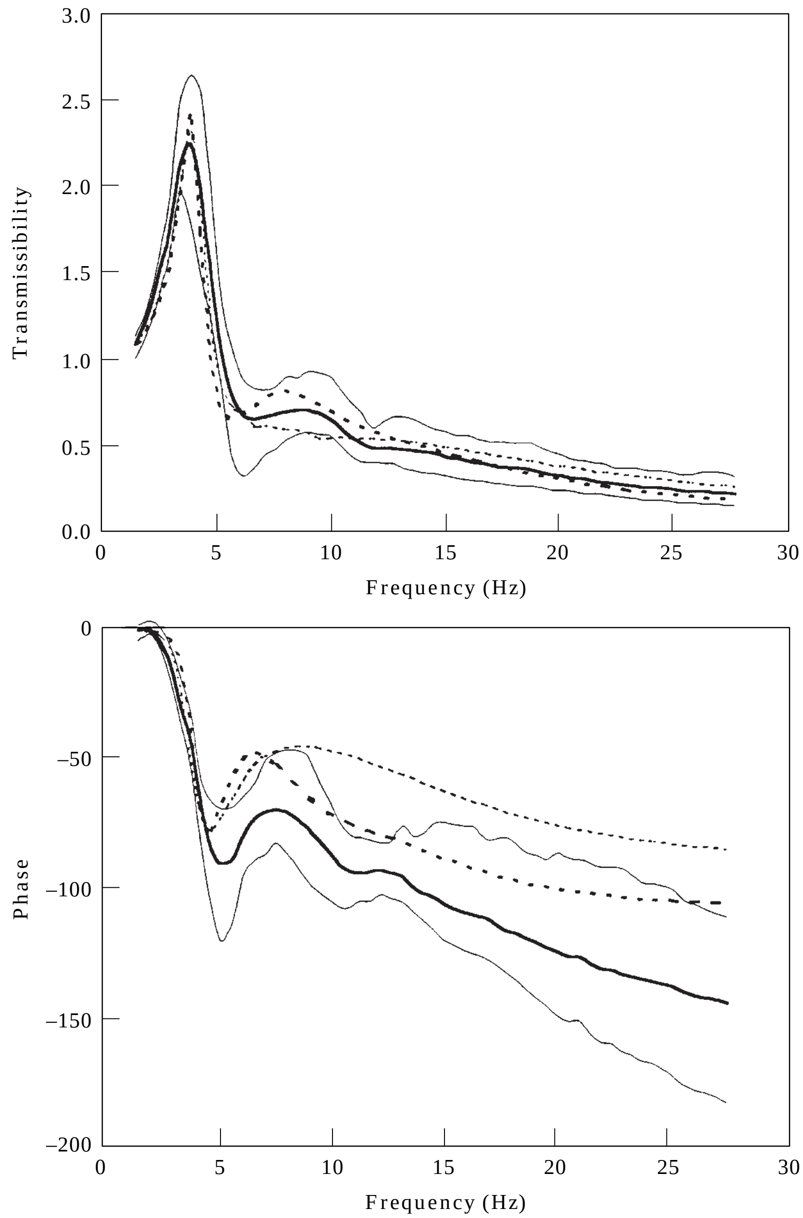

Figure 16. Comparison of measured and predicted seat transmissibility and phase. ——, Mean experimental data; —, range of experimental data; - - - -, single-degree-of-freedom model mean data; – – –, two-degree-of-freedom model mean data.

mechanical impedance, it may also be possible to develop the method to allow for differing magnitudes of vibration and the consequent non-linearity of seat transmissibility.

5. CONCLUSION

Two alternative models of the seat–person system have been investigated. A single-degree-of-freedom model can adequately reflect the dynamic characteristics of the human body at low frequencies and can be used to predict seat transmissibility at the seat resonance, usually seen around 3–5 Hz. However, a two-degree-of-freedom model provides better predictions of seat transmissibility: it predicts the second resonance, often seen in measurements of seat transmissibility around 8 Hz, and may give useful predictions of seat transmissibility at frequencies up to 25 Hz.

The encouraging results obtained from the prediction method suggest that it should allow the prediction of SEAT values for seats used in specific vibration environments.

The application of an indenter to obtain the dynamic characteristics of seats appears to provide useful data. The stiffness and damping coefficient of the seat and foam sample used in this experiment increased with increasing load.

REFERENCES

1. M. J. Griffin 1990 *Handbook of human vibration*. London: Academic Press.
2. British Standards Institution 1989 *British Standard*, *BS* 7085. British standard guide to safety aspects of experiments in which people are exposed to mechanical vibration and shock.
3. C. W. Suggs, C. F. Abrams and L. F. Stikeleather 1969 *Ergonomics* **12**, 79–90. Application of a damped spring–mass human vibration simulator in vibration testing of vehicle seats.
4. J. Matthews 1967 *Engineering Symposium of the Institution of Agricultural Engineers*, 12 September, National College of Agricultural Engineering, Silsoe, Bedford. Progress in the application of ergonomics to agricultural engineering.
5. N. J. Mansfield and M. J. Griffin 1996 in *Inter-noise* '96, *Proceedings of* 25*th Anniversary Congress*, *Liverpool*, Book 4. Institute of Acoustics. Vehicle seat dynamics measured with an anthropodynamic dummy and human subjects.
6. T. E. Fairley and M. J. Griffin 1986 *Society of Automotive Engineers International Congress and Exposition*, *Detroit SAE Paper* 860046. A test method for the prediction of seat transmissibility.
7. L. Wei and M. J. Griffin 1998 *Journal of Sound and Vibration* **212,** 855–874. Mathematical models for the mechanical impedance of the seated human body exposed to vertical vibration.
8. E. M. Whitham and M. J. Griffin 1977 *Society of Automotive Engineers*, *SAE Paper* 770253, *International Automotive Engineering Congress and Exposition*, *Detroit*. Measuring vibration on soft seats.
9. International Organization for Standardization 1992 *International Standard*, *ISO* 10326-1. Mechanical vibration—laboratory method for evaluating vehicle seat vibration—Part 1: basic requirements.
10. C. Dierckx 1995 in *Oxford Science Publications*. Oxford: Clarendon Press. Curve and surface fitting with splines.
11. T. E. Fairley and M. J. Griffin 1989 *Journal of Biomechanics*, **22**, 81–94. the apparent mass of the seated human body: vertical vibration.
12. International Organization for Standardization 1981 *International Standard*, *ISO* 5982. Vibration and shock—mechanical driving point impedance of the human body.
13. C. Corbridge, M J. Griffin and P. Harborough 1989 *Proceedings of the Institution of Mechanical Engineers* **203**, 57–64. Seat dynamics and passenger comfort.