Journal of Sound and Vibration (2002) 253(1), 295–311
doi:10.1006/jsvi.2001.4261, available online at http://www.idealibrary.com on IDEAL®

EVALUATING THE VIBRATION ISOLATION OF SOFT SEAT CUSHIONS USING AN ACTIVE ANTHROPODYNAMIC DUMMY

C. H. Lewis and M. J. Griffin

Human Factors Research Unit, Institute of Sound and Vibration Research, University of Southampton, Highfield, Southampton SO17 1BJ, Hampshire, England. E-mail: mjg@isvr.soton.ac.uk

(*Accepted* 19 *October* 2001)

Seat test standards require human subjects to be used for measuring the vibration isolation of vehicle seats. Anthropodynamic dummies, based on passive mass–spring–damper systems, have been developed for testing seats but their performance has been limited at low excitation magnitudes by non-linear phenomena, such as friction in the mechanical components that provide damping. The use of an electrodynamic actuator to generate damping forces, controlled by feedback from acceleration and force transducers, may help to overcome these limitations and provide additional benefits. The transmissibilities of five foam cushions have been measured using an actively controlled anthropodynamic dummy, in which damping and spring forces were supplied by an electrodynamic actuator. The dummy could be set up to approximate alternative single-degree-of-freedom and two-degree-of-freedom apparent mass models of the seated human body by varying motion feedback parameters. Cushion transmissibilities were also measured with nine human subjects, having an average seated weight similar to the dummy. At frequencies greater than 4 Hz, mean cushion transmissibilities measured with subjects were in closer agreement with the transmissibilities obtained with a two degree-of-freedom dummy than with a single degree-of-freedom dummy. However, at frequencies between 2 and 4 Hz, cushion transmissibilities obtained with the two-degree-of-freedom dummy showed consistently larger differences from mean transmissibilities with subjects than single-degree-of-freedom dummies, indicating a need for further development of human apparent mass models to account for the effects of magnitude and spectral content of the input motion. Vertical vibration isolation efficiencies (SEAT values) of the five foams were measured with four input motions, including three motions measured in a car. The SEAT values obtained using the active dummy were highly correlated with the median SEAT values obtained with the nine human subjects, with the two-degree-of-freedom apparent mass dummy giving the highest agreement.

1. INTRODUCTION

It is desirable that vehicle seats isolate vehicle operators and passengers from vibration and shock. The optimization of vehicle seats therefore involves the measurement of their vibration isolation.

The transmission of vibration through a seat depends on the dynamic properties of the seat and the dynamic response of the body supported by the seat. The dynamic response of the human body is complex, and differs from that of a rigid mass of the same weight, so current standards for measuring seat transmissibility require the use of human subjects (e.g., reference [1]). However, the use of human subjects can be inconvenient and costly:

0022-460X/02/$35.00

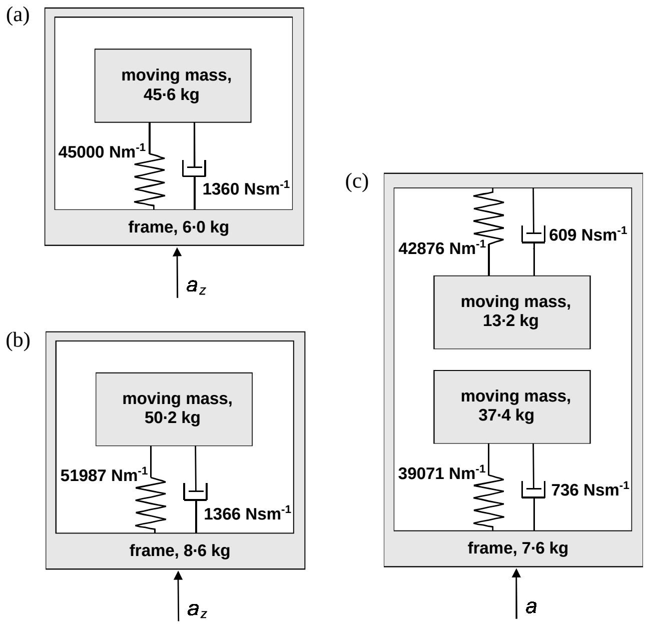

Figure 1. Mechanical models representing the apparent mass of seated subjects: (a) single-degree-of-freedom model (60 subjects) after Fairley and Griffin [3]; (b) single-degree-of-freedom model (24 men) after Wei and Griffin [4]; (c) two-degree-of-freedom model (24 men) after Wei and Griffin [4].

laboratory seat tests with human subjects require the use of specially designed simulators and experimental procedures so as to minimize the risk of injury or impairment to health [2].

1.1. APPARENT MASS MODELS OF THE HUMAN BODY

The dynamic response of the human body can be characterized by driving-point frequency response functions, such as mechanical impedance or apparent mass. These characteristics differ between individuals and also vary within the same individual, depending on sitting posture and vibration magnitude [3]. These differences between and within subjects introduce variability in the results of seat tests and make it necessary to conduct measurements with a range of subjects so as to obtain representative results.

Fairley and Griffin [3] showed that the mean normalized apparent mass of 60 subjects, seated on a hard flat seat and vibrated with broadband (0·25–20 Hz) motion at a magnitude of $1{\cdot}0\,\mathrm{m/s^2}$ r.m.s., could be represented by an ideal single-degree-of-freedom mass–spring–damper system with a natural frequency of 5 Hz (Figure 1(a)). The theoretical apparent mass of the single-degree-of-freedom model was within $\pm$ 1 standard deviation of the mean normalized apparent mass at most frequencies between 0 and 20 Hz. Wei and Griffin [4] reanalyzed the apparent mass data gathered by Fairley and Griffin [3], fitting the parameters of both a single-degree-of-freedom model (Figure 1(b)) and a two-degree-of-freedom (Figure 1(c)) models to the 60 subjects and, separately, for the men (24 subjects), the women (24 subjects) and the children (12 subjects) within the group. These studies suggest it should be possible to replace human subjects in a seat test by either a one-degree-of-freedom mechanical dummy or a two-degree-of-freedom mechanical dummy. The use of a mechanical dummy, with a fixed apparent mass characteristic, would provide standardized load conditions and circumvent the need for safety precautions and repeated tests with several human subjects.

1.2. ANTHROPODYNAMIC SEAT TEST DUMMIES

Studies with prototype passive mechanical dummies have shown that they can give similar measurements of vertical seat isolation to those obtained with human subjects on laboratory simulators [5–10] and in an automobile [11]. However, a mechanical dummy suitable for measuring seat transmissibility in laboratory conditions and in vehicles (on and off-road), would need to be capable of representing the driving point frequency response of human subjects over a wide range of vibration magnitudes. Mechanical suspension components, such as dampers, tend to have limitations that modify their dynamic performance when the excitation magnitude is lower, or higher, than an optimum operating range and result in non-linearities in mechanical dummies [12]. These mechanical non-linearities differ from the large non-linearities that occur in the driving point apparent mass of human subjects.

The use of an electrodynamic actuator to generate damping forces, controlled by feedback from acceleration and force transducers, can provide a mechanism with low mechanical friction, as well as making it possible to change the response of a dummy by switching feedback parameters. Since the resonance frequency and damping of the human body varies systematically with changes in vibration magnitude, it may be desirable to vary the response of a seat test dummy for different applications, such as for cars with low vibration magnitudes and for off-road vehicles with high vibration magnitudes. In addition, the requirements of some current seat test standards to obtain measurements with subjects having a range of weights might be achieved with an active dummy in which there was compensation for variations in mass without changes to spring and damper components.

Preliminary measurements with a prototype single-degree-of-freedom active dummy, in which the damping force and part of the spring force were provided by a power amplifier and electrodynamic actuator, have been previously reported [13]. The driving signals for the spring and damper forces were derived from the displacement and velocity of the spring mass relative to the dummy frame. The spring stiffness and viscous damping of the dummy could be varied by changing, respectively, the displacement feedback gain and velocity feedback gain. The vertical transmissibilities of five foam cushions, with different dynamic stiffnesses, were measured while they were loaded by the prototype dummy. The transmissibility of each cushion was also measured while loaded with a human subject. Figure 2 compares transmissibilities measured with the subject and with the dummy having a natural frequency and damping as in the Fairley and Griffin [3] model. The peak transmissibilities measured with the dummy, and the frequencies at which they occurred, followed the corresponding values for the human subject. The measurements with the dummy overestimated the seat transmissibility with the subject between about 4 and 6 Hz, and slightly underestimated the seat transmissibility with the subject at higher frequencies. The phase lag in the transmissibilities was greater with the dummy than with the subject at all frequencies above 4 Hz.

A re-analysis of the apparent mass data upon which the Fairley and Griffin model was based has shown that the use of a two-degree-of-freedom model provides a better fit to the phase of the apparent mass at frequencies greater than 8 Hz and an improved fit to the modulus of the apparent mass at frequencies around 5 Hz [4]. The addition of a second degree-of-freedom to the dummy may therefore bring the shapes of the resulting cushion

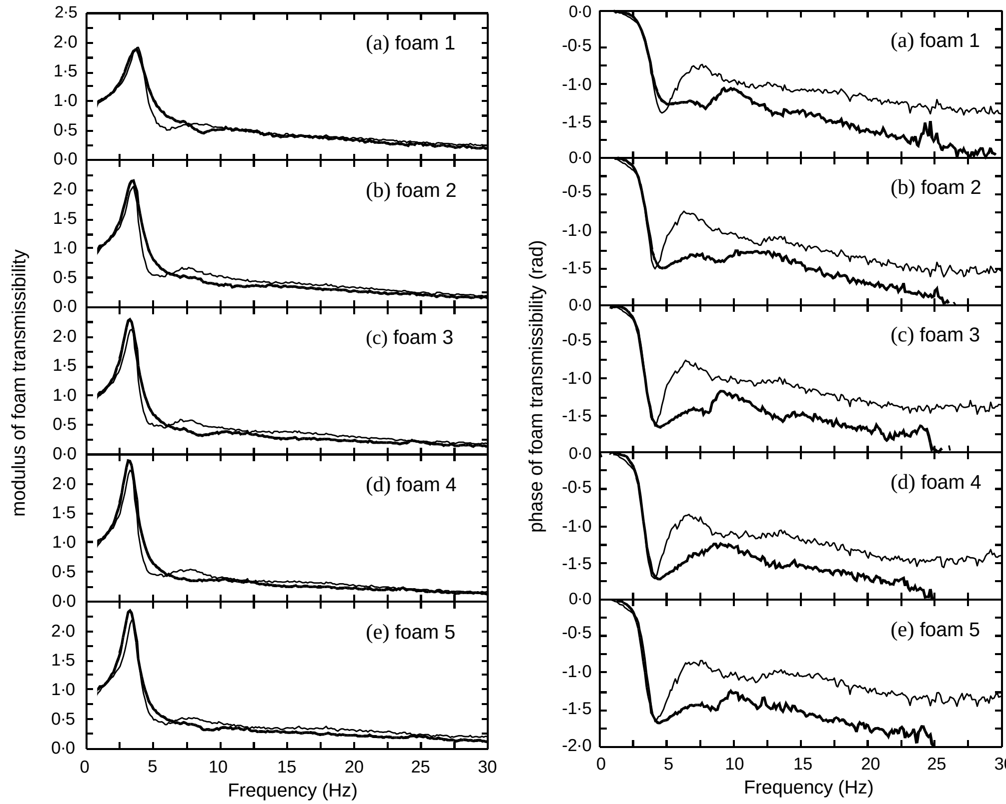

Figure 2. Measured transmissibilities of five different foams with a single-degree-of-freedom active dummy and with a human subject (after Lewis [13]). Measured with 1–30 Hz broadband acceleration at 1·0 m/s^2 r.m.s. ——— dummy; —— subject.

transmissibility curves closer to those measured with human subjects. The mechanical complexity associated with adding an additional mass–spring–damper system is likely to increase costs and calibration problems and decrease reliability, however, active control may provide a means for approximating a two-degree-of-freedom response without additional components.

In this study, the transmissibility of foam cushions was measured with nine human subjects and compared with the transmissibility measured using a modified prototype active anthropodynamic dummy. The dummy was modified so as to approximate to both a one-degree-of-freedom apparent mass model and a two-degree-of-freedom apparent mass model of the seated human body. The cushion transmissibilities were measured in the laboratory using motions recorded in vehicles and synthetic signals.

2. MATERIALS AND METHODS

2.1. ACTIVE DUMMY TEST RIG

The active dummy comprised a single moving mass, which was constrained to move in the vertical direction relative to a rigid frame by linear ball bushings running on steel shafts

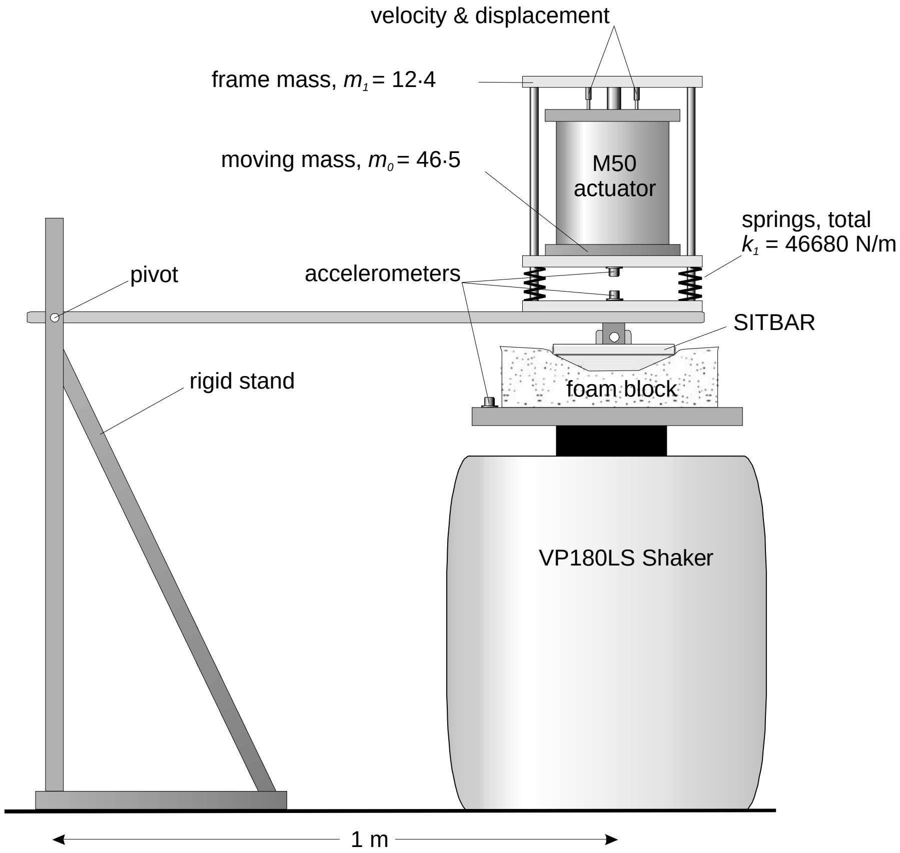

Figure 3. Elevation of the active dummy and test rig.

(Figure 3). Most of the moving mass was provided by the permanent magnet of an electrodynamic actuator (Gearing and Watson model M50). The moving part of the actuator was fixed to the top of the dummy frame. The armature of the actuator was immobile with respect to the frame. A linear variable differential transformer (LVDT: RDP electronics type D2/200A) displacement transducer and a linear velocity transducer (LVT: Trans-Tek type 0101-0000) were fixed between the frame and the moving mass so as to provide motion feedback signals to drive the actuator (Figure 4). Accelerometers with a DC response (Setra 141A: range $\pm 2g$) were fixed to the moving mass and to the frame.

The dummy was supported on the test cushions by a SIT-BAR shaped seat indenter [14] (Figure 5), and constrained to move in the vertical axis by a light swinging arm, which pivoted from a rigid stand (see Figure 3). The cushion was supported by a rigid plate that was attached to a Derritron VP180LS electrodynamic shaker.

The driving point force, f_0, at the base of the dummy is given by:

$$f_0(t) = m_0\ddot{x}_0(t) + m_1\ddot{x}_1(t). \tag{1}$$

The total force acting on mass m_1 is

$$m_1\ddot{x}_1(t) = -f_A(t) - k_1 u_1(t), \tag{2}$$

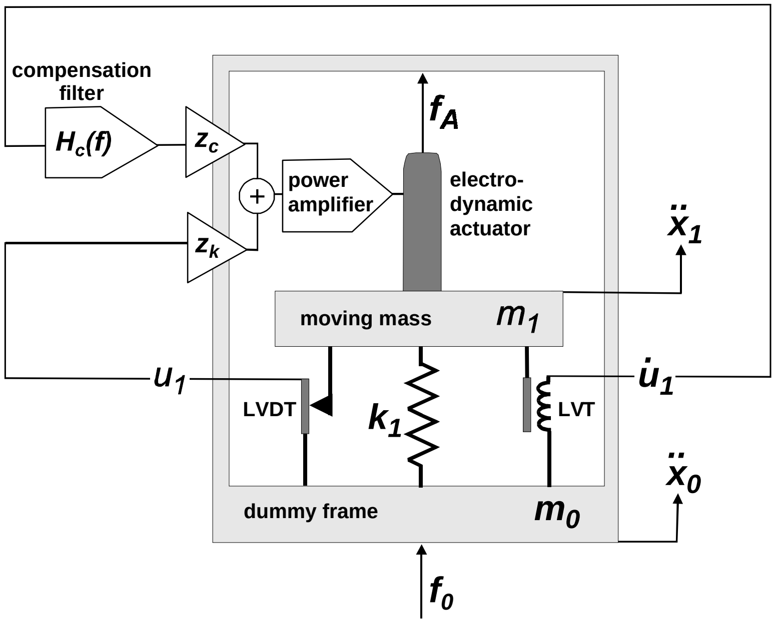

Figure 4. Schematic diagram of the active anthropodynamic dummy.

where f_A is the force developed by the electrodynamic actuator, which is derived from

$$f_A(t) = z_c\dot{u}_1(t) + z_k u_1(t) \tag{3}$$

and $u_1 = x_1 - x_0$.

For the present study, the control system was modified to approximate a two-degree-of-freedom mechanical system by the addition of an all-pass filter, $H_c(\omega)$, in the velocity feedback (see Figure 4) where

$$H_c(\omega) = \frac{-\omega^2 + \mathrm{j}\omega/q_{z2} + \omega_2^2}{-\omega^2 + \mathrm{j}\omega/q_{p2} + \omega_2^2}. \tag{4}$$

The frequency and magnitude of the second resonance in the apparent mass are determined by ω_2 and q_{p2}/q_{z2} respectively.

The vertical apparent mass, $M(\omega)$, of the dummy shown in Figure 4 is then given by

$$M(\omega) = \frac{F_0(\omega)}{-\omega^2 X_0(\omega)} = m_0 + m_1\left(\frac{\mathrm{j}\omega H_c(\omega)z_c + z_k + k_1}{-\omega^2 m_1 + \mathrm{j}\omega H_c(\omega)z_c + z_k + k_1}\right) \tag{5}$$

where $F_0(\omega)$ and $X_0(\omega)$ are Fourier transforms of $f_0(t)$ and $x_0(t)$ respectively.

Three settings of the dummy were used in the experiment. The first setting corresponded to the same one-degree-of-freedom response as in a previous study [13]. With the velocity feedback filter omitted, the values of z_k and z_c were adjusted to give the same dummy natural frequency, f_n, and damping ratio, ζ, as in the Fairley and Griffin [3] model according to the following relationships:

$$f_n = \frac{1}{2\pi}\sqrt{\frac{z_k + k_1}{m_1}} = 5{\cdot}0 \text{ Hz}, \qquad \zeta = \frac{z_c}{2\sqrt{(z_k + k_1)\,m_1}} = 0{\cdot}475. \tag{6, 7}$$

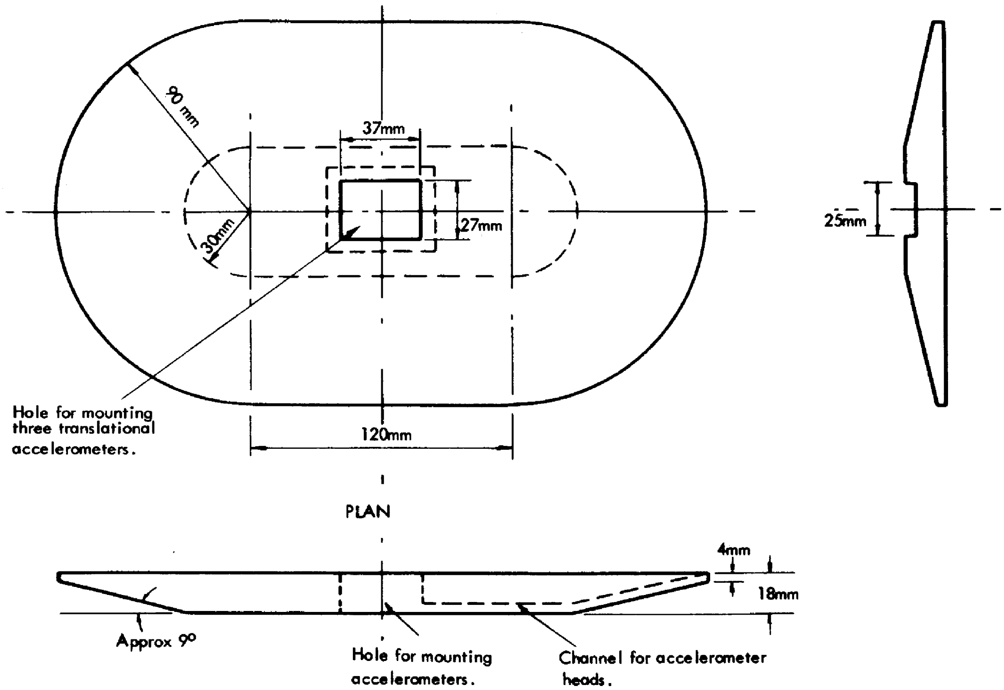

Figure 5. Design of the SIT-BAR (Seat Interface for Transducers indicating Body Acceleration Received) seat indenter, from Whitham and Griffin [14].

Due to mechanical constraints, there were some differences between the masses of the dummy and those of the Fairley and Griffin model (m_1 was 46·5 kg and m_0 was 12·4 kg compared to 45·6 kg and 6·0 kg in the model).

The other two settings of the active dummy approximated the Wei and Griffin [4] one- and two-degree-of-freedom model for 24 male subjects. The average seated mass of the 24 males was close to the 59 kg total mass of the prototype dummy, unlike the Fairley and Griffin [3] response for 60 men, women and children, which has an average seated mass of 52 kg.

The parameters of equation (5) were optimized (apart from the masses, which were constrained to the actual masses of the dummy) to fit the apparent mass curves corresponding to the Wei and Griffin [4] models at frequencies up to 30 Hz. The parameters were optimized to minimise the mean-square error between the apparent mass moduli using the Nelder–Mead simplex search procedure provided by MATLAB software (version 5.3). The fitted parameter values are shown in Table 1.

2.2. TEST CUSHIONS

Five test cushions were used in the study, consisting of identically shaped blocks of different polyurethane foams. The dynamic stiffness of each foam block was measured using the method described by Fairley and Griffin [15], using a preload force of 600 N and a 0·25–30 Hz broadband stimulus at 1·0 m/s^2 r.m.s. The dynamic stiffness of a cushion is a complex function of frequency which, over small displacements, is equivalent to

$$S(\omega) = k_0(\omega) + \mathrm{j}\omega c_0(\omega) \tag{8}$$

TABLE 1

Dummy parameters used in the experiment

Description	Parameter	Same f_n and ζ as Fairley and Griffin [3] single-degree-of-freedom model	Optimized to Wei and Griffin [4] single-degree-of-freedom model	Optimized to Wei and Griffin [4] two-degrees-of-freedom model
Moving mass (kg)	m_0	46·5	46·5	46·5
Frame mass (kg)	m_1	12·35	12·35	12·35
Equivalent stiffness (N/m)	$z_k + k_1$	45 894	48 123	61 814
Equivalent viscous damping (N s/m^2)	z_c	1387	1197	820
Natural frequency (Hz)	f_n	5·00	5·12	5·80
Damping ratio	ζ	0·475	0·400	0·242
Second resonance (Hz)	$\omega_2/2\pi$	—	—	10·0
Selectivity of zero pair	q_z	—	—	0·415
Selectivity of pole pair	q_p	—	—	0·733

where $k_0(\omega)$ is the equivalent spring stiffness and $c_0(\omega)$ is the equivalent viscous damping provided by the foam at frequency ω. The real and imaginary parts of the vertical dynamic stiffness of each test cushion are compared in Figure 6.

2.3. MEASUREMENT OF VIBRATION ISOLATION

The cushions were supported on a Derritron VP180LS electrodynamic shaker, having a maximum displacement of $\pm$ 25 mm. Four different input motions were generated. One signal comprised broadband motion with an approximately flat spectrum between 2 and 30 Hz. The other three signals were recorded at the seat rail of a small car on different road surfaces. The duration of each signal was 60 s. Figure 7 shows the acceleration power spectral densities of the four signals measured on the shaker platform. The recorded signals were high-pass filtered to attenuate the signals below 2 Hz because of the displacement limitations of the shaker, and low-pass filtered at 30 Hz, the upper design target of the dummy. Characteristics of the four signals are summarized in Table 2.

The driving signal for the shaker was generated, and the accelerations of the shaker platform, dummy frame and moving mass were digitized simultaneously at 256 samples/s using an *HVLab* data acquisition and analysis system (version 3.81). The acceleration signals were low-pass filtered (8 pole Butterworth) at 60 Hz to remove high-frequency components that could cause aliasing. The vertical transmissibility of the foam cushion, $H_0(\omega)$, was calculated from

$$H_0(\omega) = \frac{G_{p0}(\omega)}{G_{pp}(\omega)}, \tag{9}$$

where $G_{p0}(\omega)$ is the cross spectral density of the accelerations on the shaker platform and the dummy frame, and $G_{pp}(\omega)$ is the power-spectral density of the acceleration on the shaker platform. Power and cross-spectral estimates were calculated from overlapping Fourier transforms, using a Hamming spectral window, with a resolution of 0·125 Hz and 32 degrees-of-freedom.

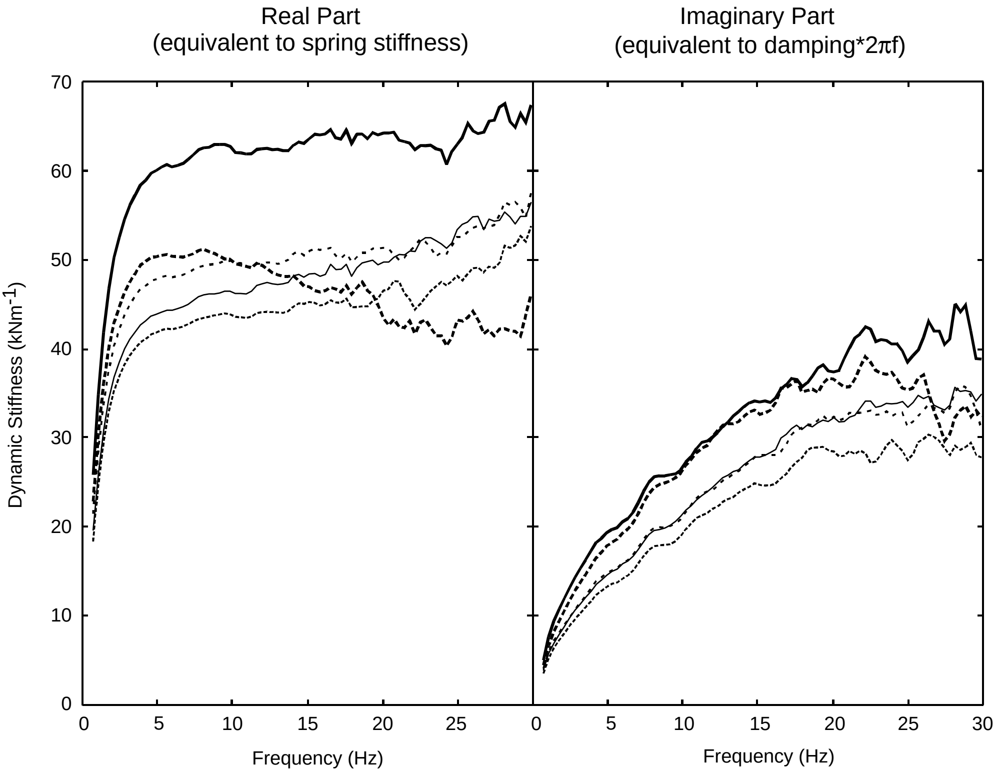

Figure 6. Dynamic stiffness of five different foams measured with a preload force of 600 N. Resolution = 0·35 Hz. ——, foam 1; – – – –, foam 2; ——, foam 3; - - - -, foam 4; ----- foam 5.

The usefulness of a seat vibration isolation characteristic depends on both the input spectrum and the transmissibility, since the seat only needs to provide isolation at frequencies that are present in the input. Seat isolation performance was indicated by the SEAT value, which can be calculated from frequency-weighted r.m.s. accelerations, or vibration dose values:

$$SEAT_{r.m.s.} = \frac{W_b \text{ weighted r. m. s. acceleration on cushion}}{W_b \text{ weighted r. m. s. acceleration under cushion}}, \tag{10}$$

$$SEAT_{VDV} = \frac{W_b \text{ weighted } VDV \text{ on cushion}}{W_b \text{ weighted } VDV \text{ under cushion}}, \tag{11}$$

where

$$VDV = \left(\int a_w^4(t)\, \mathrm{d}t \right)^{1/4}. \tag{12}$$

W_b is the frequency weighting defined for evaluation of vertical vibration on the seat surface in BS 6841:1987 [16] and $a_w(t)$ is the frequency-weighted acceleration. Current standards [16, 17] recommend that r.m.s. accelerations are used to evaluate ride comfort unless the crest factor is high, in which case the use of the vibration dose value, VDV, which gives more weight to the peak values of an acceleration time history, is preferred.

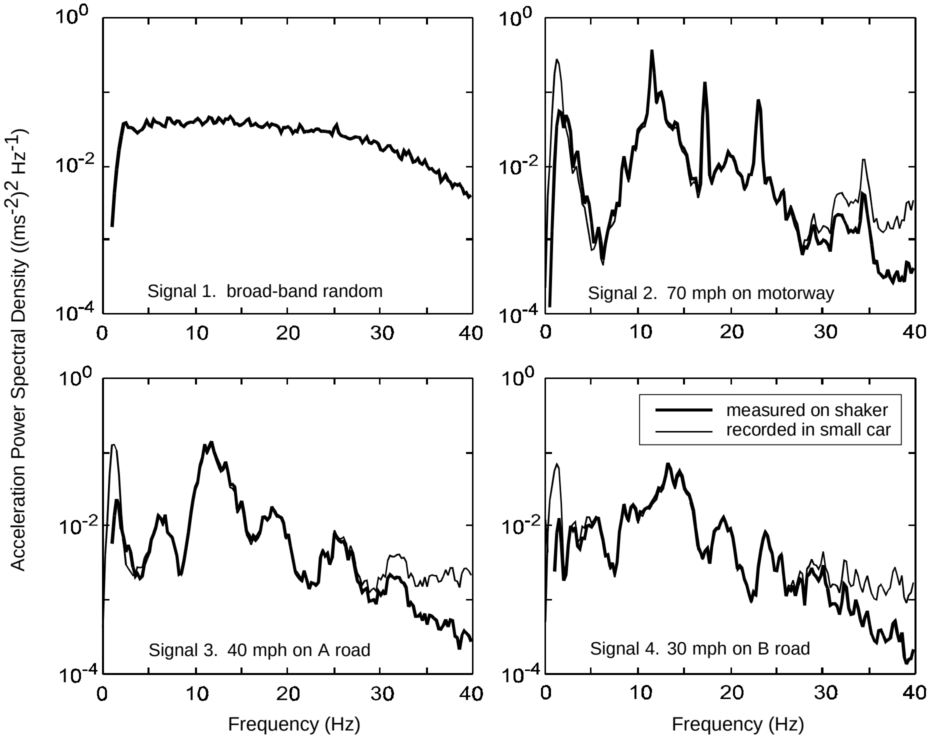

Figure 7. Acceleration power-spectral densities of the four input signals used in the study.

TABLE 2

Characteristics of the four vibration signals used in the study. The duration of each signal was 60 *s and measurements were made with a bandwidth of* 60 *Hz.* W_b *is the frequency weighting defined for evaluation of vertical vibration on the seat surface in BS* 6841:1987 [16]

Signal	Unweighted r.m.s. acceleration (m/s^2 r.m.s.)	W_b weighted r.m.s. acceleration (m/s^2 r.m.s.)	Crest factor of W_b weighted acceleration
1. Broadband random	1·00	0·78	3·7
2. 70 mph, motorway	0·82	0·65	5·2
3. 40 mph, A road	0·71	0·61	6·7
4. 30 mph, B road	0·59	0·49	8·5

The vertical apparent mass of the dummy, $M(\omega)$, is equivalent to

$$M(\omega) = \frac{F_0(\omega)}{\ddot{X}_0(\omega)}, \tag{13}$$

where $F_0(\omega)$ and $\ddot{X}_0(\omega)$ are Fourier Transforms of the driving point force, $f_0(t)$, and the acceleration, $\ddot{x}_0(t)$, on the dummy frame. It is evident from equations (1) and (13) that, if the masses are known, the apparent mass can be estimated from the accelerations on the frame

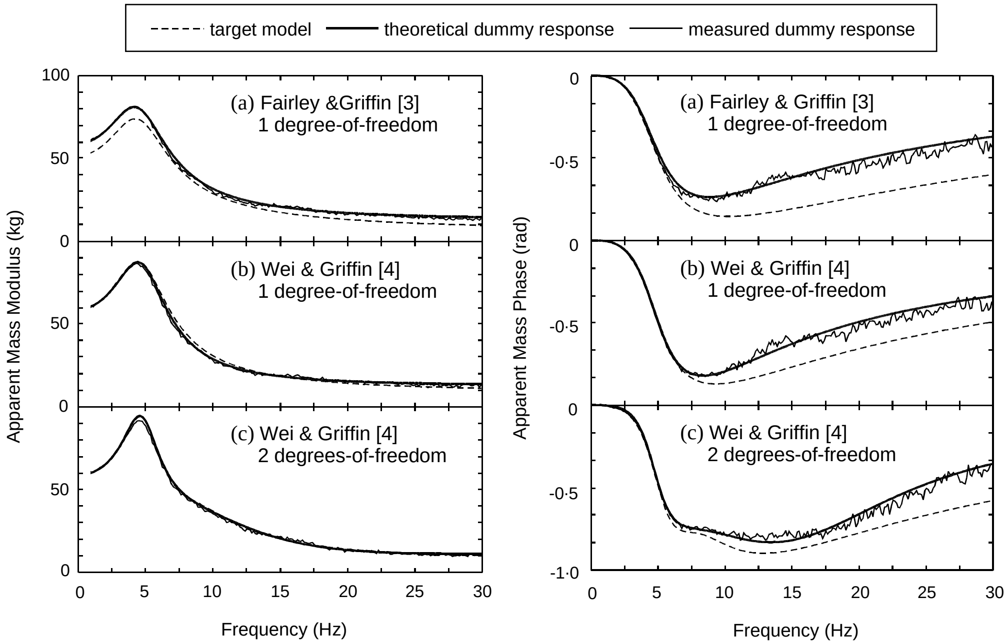

Figure 8. Modulus and phase of three published apparent mass models, compared with corresponding dummy responses (resolution 0·125 Hz). ——, foam 1; ----, foam 2; ——, foam 3; ----, foam 4; ----- foam 5.

and the moving mass:

$$M(\omega) = m_0 + m_1 \frac{\ddot{X}_1(\omega)}{\ddot{X}_0(\omega)} = m_0 + m_1 \frac{G_{01}(\omega)}{G_{00}(\omega)}, \tag{14}$$

where $G_{01}(\omega)$ is the cross-spectral density of the accelerations measured on the dummy frame and the moving mass, and $G_{00}(\omega)$ is the power-spectral density of the acceleration measured on the frame.

3. PROCEDURE

The transmissibility of each of the five foams (see Figure 6) was measured with all three dummy settings shown in Table 1, as well as with the dummy suspension locked out to provide a 59 kg rigid mass. The base of the cushion was excited by the broadband acceleration signal (see Figure 7). The vertical transmissibility of the foam cushion was calculated from the accelerations on the cushion base and cushion surface using equation (10). The vertical apparent mass of the dummy was also estimated using equation (14).

Figure 8 compares the measured apparent masses of the alternative active dummies with their theoretical apparent masses (calculated from the active dummy settings shown in Table 1) and the target apparent masses (of the original mechanical models shown in Figure 1). The apparent masses were measured indirectly during the cushion transmissibility measurements, from the accelerations of the moving mass and the frame mass, using equation (14). The measurements shown in Figure 8 were made on foam 3, but the same responses were obtained on the other foams. The theoretical and measured apparent mass of dummy (a) was higher than the Fairley and Griffin [3] target model due to the high frame mass. The theoretical and measured responses of dummies (b) and (c) closely

TABLE 3

Standing and seated weights of the nine subjects used in the study

Subject	Standing height (m)	Standing weight (kg)	Seated weight (kg)
1	1·73	89	70
2	1·83	86	63
3	1·85	82	67
4	1·75	66	54
5	1·76	77	55
6	1·85	85	58
7	1·74	86	65
8	1·70	73	53
9	1·80	87	65
Mean	1·78	81	61

followed the moduli of the target models at frequencies between 2 and 30 Hz. The parameters of these models were optimized according to the apparent mass modulus; the phase responses departed from the ideal models above 5 Hz, but followed a similar pattern. The phase departures were primarily caused by the high frame mass of the prototype dummy compared with the original models. This might be reduced in a production version by the re-design of the structure and the use of lighter materials.

The transmissibility of each of the five test cushions was also measured with nine male human subjects (see Table 3). The subjects sat in an erect posture on the foam block, which was supported on the VP180LS shaker platform, as for the measurements using the dummy. The feet of the subjects were supported by a stationary footrest. The vertical acceleration on the cushion surface was measured using a rigid SIT-BAR [14] (see Figure 4), placed between subjects and the foam block. A rigid SIT-BAR was used to standardize the contact conditions between dummy and subjects, so as to ensure that measured differences were only due to differences in apparent mass and not confounded with contact conditions. The SIT-BAR has been shown in other studies to give similar seat transmissibilities to those obtained using a standard flexible seat pad [1].

The vibration isolation of each of the five foams was calculated, using equations (10) and (11), for all four input signals defined in Table 2 and Figure 7. The $SEAT_{r.m.s.}$ and $SEAT_{VDV}$ values were calculated for all three dummy settings in Table 1, for the rigid mass, and for each of the nine human subjects.

4. RESULTS

4.1. TRANSMISSIBILITY MEASUREMENTS

Figure 9 compares the means of foam transmissibilities measured with the subjects with transmissibilities measured using the dummy. Data are shown for each of the four dummy settings and for foams 1 and 4 (the most stiff and the least stiff of the five foams at low frequencies: see Figure 6).

With the dummy suspension locked, to provide a rigid mass, the transmissibility peak was higher by a factor of more than two, and occurred at a higher frequency, than with the subjects. At frequencies above 10 Hz the cushion provided more attenuation when loaded with the mass than when loaded with subjects.

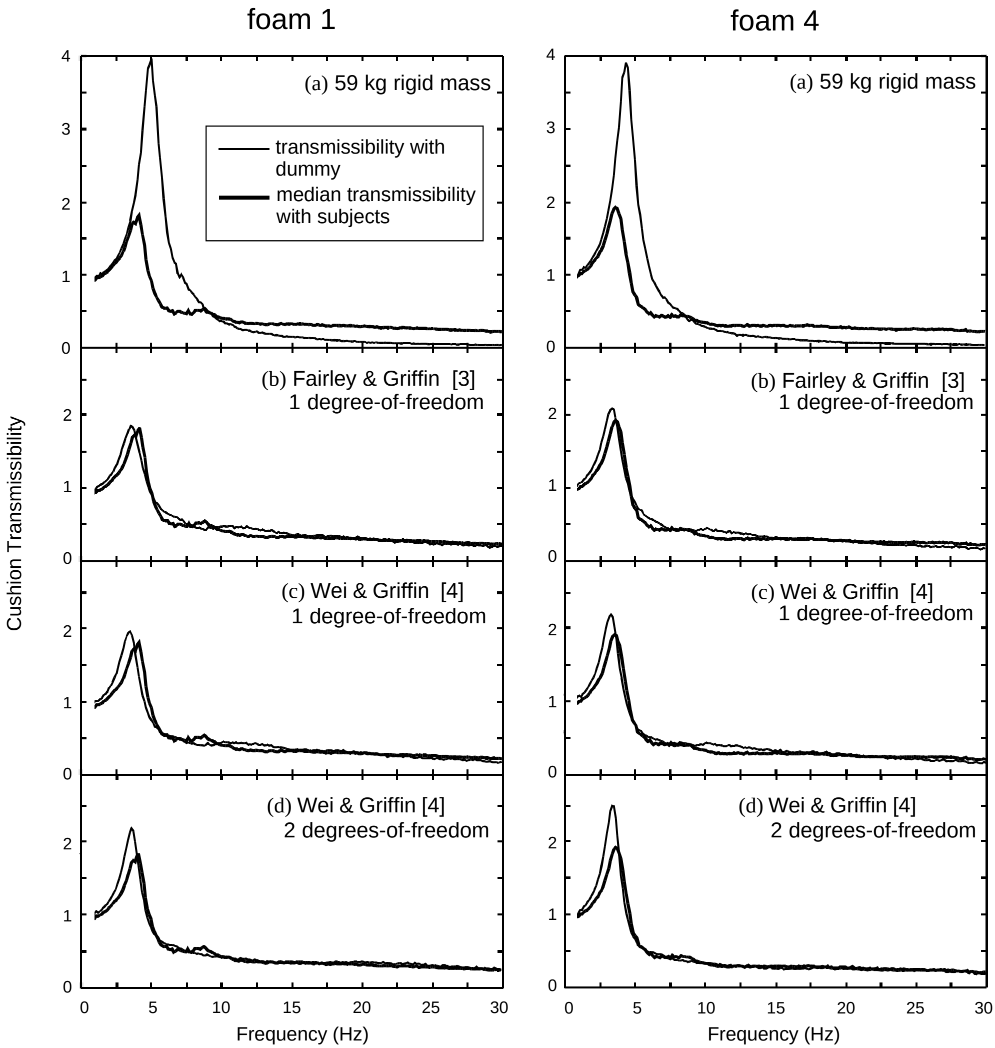

Figure 9. Mean cushion transmissibility measured with nine subjects compared with cushion transmissibilities measured with a rigid mass, and with the three dummy configurations shown in Figure 8. Broadband random excitation at 1·0 m/s^2 r.m.s. (signal 1 in Figure 7). Resolution = 0·125 Hz.

With each of the three active dummy settings, the cushion resonance occurred at a slightly lower frequency than with the subjects, and the transmissibility at resonance was higher than with subjects. This was particularly the case for the two-degree-of-freedom dummy. However, between 4 and 30 Hz, the cushion transmissibility measured with the two-degree-of-freedom dummy was closest to the median values measured with the subjects. The single-degree-of-freedom responses were close to the subject data over much of the frequency range, but there were some departures at frequencies around 7 Hz, around 12 Hz, and above 25 Hz.

The frequency of the main cushion resonance was about 5% higher with foam 1 than with foam 4, but the differences in seat transmissibility between the dummies and the subjects were similar for both foams.

TABLE 4

Proportion of frequency points, between 2 and 30 Hz, at which the foam transmissibility with the dummy, measured with broadband acceleration (signal 1 in Table 2), was within the 95% confidence interval for the mean transmissibility with the subjects

Dummy setting	Foam 1	Foam 2	Foam 3	Foam 4	Foam 5
59 kg rigid mass	8%	8%	7%	4%	2%
One degree-of-freedom [3]	48%	47%	47%	44%	30%
One degree-of-freedom [4]	46%	72%	72%	67%	54%
Two degree-of-freedom [4]	83%	87%	87%	87%	77%

The data suggest that it may be desirable to adjust the frequency and damping of the first resonance of the two-degree-of-freedom model so as to improve the agreement with data from subjects at frequencies between 2 and 4 Hz. The models used to establish the dummy parameters were based on data from subjects exposed to 1 m/s^2 r.m.s. broadband (0–20 Hz) acceleration on a hard, flat seat. The apparent mass of human subjects varies with input vibration magnitude, and the difference in vibration magnitude on the seat may have contributed to the differences seen in Figure 9. The broadband input vibration (at 1·0 m/s^2 r.m.s., unweighted) in the current study was modified by the response of the cushion so the parameters of the apparent mass model may not be optimum for the motion that occurred on the cushion surface. Active control makes it feasible to account for systematic changes in human response by varying the parameters of the mechanical model based on appropriate motion feedback. However, further research is needed to determine suitable relationships between optimum dummy parameters and motion characteristics.

Ninety-five percent confidence intervals were computed for the means of the transmissibilities measured with the subjects, using the Student's t distribution. The confidence intervals were calculated for each of the five foams, at 0·125 Hz intervals between 2 and 30 Hz. The transmissibilities with each of the four dummies were then compared with the confidence intervals to determine the frequencies at which there was no statistically significant difference between the cushion transmissibility measured with a dummy and the mean transmissibility measured with subjects (i.e., frequencies at which a dummy provided a statistically satisfactory estimate of the average transmissibility obtained with subjects). Table 4 shows the proportions of the frequency points at which the transmissibility with each dummy was within the 95% confidence interval for the mean transmissibility with subjects. It can be seen that, with each of the five test foams, the rigid mass provided estimates of the transmissibility within the 95% confidence interval for less than 10% of the frequencies, while the two-degree-of-freedom dummy setting provided estimates within the 95% confidence interval over more than 75% of the frequency range.

4.2. SEAT VALUES

Figure 10 shows the isolation efficiency of the five foam cushions, as indicated by SEAT values. The SEAT value takes into account the spectrum of the input motion, giving an indication of the overall severity of the motion on the seat surface compared with that at the seat base. SEAT values were calculated for all 20 combinations of four input signals and five foams, with each of the four dummy settings. For each dummy setting, the SEAT values measured with the dummy are plotted against the means of the SEAT values measured with

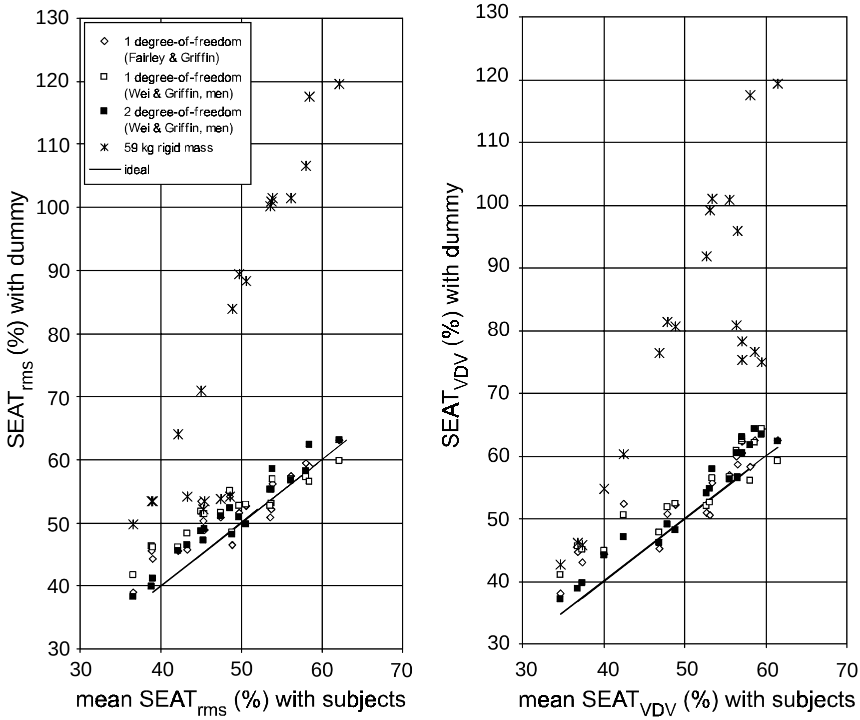

Figure 10. $SEAT_{r.m.s.}$ and $SEAT_{VDV}$ values measured with the dummy, plotted against mean values for nine subjects, for all 20 combinations of four input signals and five foam cushions.

TABLE 5

Correlation (Pearson r), intercept and slope of the linear regression line between SEAT values with dummy and the mean SEAT value with subjects, over all combinations of five foams and four input signals

Dummy setting	r	Intercept (%)	Slope	r	Intercept (%)	Slope
59 kg rigid mass	0·91	– 79	3·22	0·81	– 30	2·16
One degree-of-freedom [3]	0·92	14	0·77	0·93	10	0·85
One degree-of-freedom [4]	0·93	22	0·61	0·92	16	0·75
Two degree-of-freedom [4]	0·97	4	0·97	0·97	1	1·03

the subjects. With the rigid mass, the SEAT values are all higher, and more scattered, than with subjects. The three active dummy settings all appear to give useful estimates of the vibration isolation provided by the cushions. Pearson product-moment correlation coefficients between the dummy data and the mean subject data showed that there was a highly significant ($p < 0{\cdot}01$) association between the SEAT values with all dummies (including the rigid mass) and the mean SEAT values for the subjects (Table 5), but the two-degree-of-freedom dummy gave the closest correlation with the subject data, both when SEAT values were calculated from r.m.s. acceleration and when they were calculated from vibration dose values (see equations 10–12).

Calculations were also made of the intercept and slope of the least-squares regression line between the mean SEAT value with subjects and SEAT values with the dummy. A slope close to unity, with an intercept close to zero and a high correlation coefficient, indicates that SEAT values with the dummy were close estimates of mean SEAT values with subjects over the whole range of investigated vibration inputs and foam dynamic stiffness. The values shown in Table 4 indicate that the two-degree-of-freedom dummy gave the closest agreement with the subject data, with a slope of 0·97 for $SEAT_{r.m.s.}$ values and a slope of 1·03 for $SEAT_{VDV}$ values.

5. CONCLUSIONS

Active control offers the ability to vary spring and damper forces in a single-degree-of-freedom seat test dummy as a function of frequency, making it possible to generate an apparent mass characteristic that departs from that of a single degree-of-freedom system without the complexity of adding more moving parts.

Measurements of the effective vibration isolation provided by foam cushions with the active dummy were highly correlated with median measurements from nine human subjects. A dummy approximating a two-degree-of-freedom apparent mass characteristic gave a higher agreement with the median data from nine subjects than a single-degree-of-freedom dummy. This was consistent with the two-degree-of-freedom dummy providing the best estimate of seat transmissibility, with a better fit to the subject data, at frequencies between 4 and 30 Hz.

At frequencies between 2 and 4 Hz, cushion transmissibilities obtained with the two-degree-of-freedom dummy showed consistently larger differences from the mean transmissibilities measured with subjects, compared with single-degree-of-freedom dummies. This indicates a need for further investigation and optimization of apparent mass models of the body, accounting for the effects of the magnitude and spectral content of the input motion.

A mechanical dummy with a suitable apparent mass characteristic may provide a standard measurement condition that avoids the need for human subjects in seat testing and eliminates the random error caused by inter-subject variability. If an appropriate relationship can be defined between changes in human dynamic response and motion characteristics, active control should also make it possible to produce an adaptive dummy that will simulate the driving point response of human subjects over a wide range of different input motion magnitudes.

ACKNOWLEDGMENTS

This research was supported by the Ford Motor Company. The authors would particularly like to thank Dr Bill Pielemeier and Mr Alan Brunning for their suggestions and kind assistance.

REFERENCES

1. INTERNATIONAL ORGANIZATION FOR STANDARDIZATION 1992 *International Standard* 10326-1:1992(*E*). Mechanical Vibration—laboratory method for evaluating seat vibration—part 1: basic requirements.
2. INTERNATIONAL ORGANIZATION FOR STANDARDIZATION 1998 *International Standard EN ISO* 13090-1:1998. Mechanical vibration and shock—guide to the safety of tests and experiments in

which people are exposed to vibration and shock—part 1: mechanical vibration and repeated shock.

3. T. E. FAIRLEY and M. J. GRIFFIN 1989 *Journal of Biomechanics* **22**, 81–94. The apparent mass of the seated human body: vertical vibration.
4. L. WEI and M. J. GRIFFIN 1998 *Journal of Sound and Vibration* **212**, 855–874. Mathematical models for the apparent mass of the seated human body exposed to vertical vibration.
5. C. W. SUGGS, C. F. ABRAMS and L. F. STRIKELEATHER 1969 *Ergonomics* **12**, 79–90. Application of a damped spring–mass human vibration simulator in vibration testing of vehicle seats.
6. R. W. TOMLINSON and D. J. KYLE 1970 *Departmental note DN/TE/*037/1445, *Tractor and Machine Performance Division*, *National Institute Of Agricultural Engineering*, *Wrest Park*, *Silsoe*, *Bedford*. The development of a dynamic model of the seated human operator.
7. D. R. HUSTON, C. C. JOHNSON and X. D. ZHAO 1998 *Journal of Sound and Vibration* **214**, 195-200. A human analog for testing vibration attenuating seating.
8. B. RICHTER and S. WERDIN 1999 *Summary of research project F*1687, *Federal Institute of Occupational Safety and Health*, *Dresden*, *Germany*. Design review and realization of a mechanical vibration model representing the sitting human.
9. A. CULLMAN and H. P. WÖLFEL 2001 *Clinical Biomechanics* **16** (Suppl. 1), S64–S72. Design of an active vibration dummy of sitting man.
10. M. TOWARD 2000 *Presented at the U.K. Group Meeting on Human Response to Vibration held at the ISVR*, *University of Southampton*, *England*, 13–15 *September*. Use of an anthropodynamic dummy to measure seat dynamics.
11. N. J. MANSFIELD and M. J. GRIFFIN 1996 *Inter-noise* '96, *Proceedings of* 25*th Anniversary Congress*, *Liverpool*, Book 4, 1725–1730. Institute of Acoustics. Vehicle seat dynamics measured with an anthropodynamic dummy and human subjects.
12. C. H. LEWIS 1998 *Presented at the U.K. Group Meeting on Human Response to Vibration held at the Health and Safety Executive*, *Buxton*, *Derbyshire*, *England*, 16–18 *Septembe*r. The implementation of an improved anthropodynamic dummy for testing the vibration isolation of vehicle seats.
13. C. H. LEWIS 2000 *Presented at the U.K. Group Meeting on Human Response to Vibration held at the ISVR*, *University of Southampton*, *England*, 13–15 *September*. Evaluating the vibration isolation of soft seats using an active anthropodynamic dummy.
14. E. M. WHITHAM and M. J. GRIFFIN 1977 *SAE Paper* 770253, *Society of Automotive Engineers*. Measuring vibration on soft seats.
15. T. E. FAIRLEY and M. J. GRIFFIN 1986 *SAE Paper* 860046, *Society of Automotive Engineers*. A test method for the prediction of seat transmissibility.
16. BRITISH STANDARDS INSTITUTION 1987 *British Standard BS* 6841. Measurement and evaluation of human exposure to whole-body mechanical vibration and repeated shock.
17. INTERNATIONAL ORGANIZATION FOR STANDARDIZATION 1997 *International Standard* 2631-1:1997. Mechanical vibration and shock—evaluation of human exposure to whole-body vibration—part 1: general requirements.